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Technical Assessment of Petroleum Road Fuel Tankers
Summary Report (all Work Packages)

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In partnership with:

TWI   HEALTH & SAFETY LABORATORY

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Contents amendment record

This report has been amended and issued as follows:

Version	Date	Description	Editor	Technical Referee
1.0	25/09/14	Initial full draft for DfT review	BR	RC/ME
2.0	08/10/14	2nd full draft, incorporating client and peer review comments to date	BR	RC/ME
3.0	08/11/14	New analyses (WP1, 2 & 3) incorporated	BR	RC/ME
4.0	28/11/14	Further revisions, final text	BR	RC/ME

Contents

Executive Summary

Vehicles used to transport large quantities of dangerous goods, including petroleum products, must meet the requirements of the European Agreement on the Carriage of Dangerous Goods by Road (ADR). Following examination, certain petroleum road fuel tankers have been found not to be fully compliant with the provisions of Chapter 6.8 of ADR. Amongst other things, the tanks were not radiographed during initial inspections, and those radiographed since are seen to exhibit extensive 'lack of fusion' indications in the circumferential welds that join the shell of the tank to the extrusion bands.

Following an initial assessment[1], which showed that these circumferential welds might rupture under rollover and ADR load conditions, the Department for Transport (DfT) commissioned further research to assess the safety of these non-compliant tankers and explore opportunities arising for better regulation. The scope of the research did not extend to other issues relevant to the non-compliant tankers, such as manway flanges and filling/offtake nozzles. The research consisted of three work packages:

- WP1 – Full scale testing and associated modelling, led by HSL.
- WP2 – Detailed Fracture and Fatigue Engineering Critical Assessment (ECA), led by TWI Ltd.
- WP3 – Accident data and regulatory implications, and production of an overall summary report of the research, led by TRL Ltd.

This report pulls together and summarises all the results, drawing out the key conclusions for a non-technical, policy and legislative audience. Full technical details are provided in specific reports from each of the Work Packages.

Industry contacts indicate that petroleum road fuel tankers used in the UK are typically six-axle articulated vehicles travelling some 220,000 km each year. The UK road fuel tanker fleet is estimated to be around 1,000 – 1,500 vehicles, of which at the outset of the research, there were estimated to be 220 of the non-compliant tankers.

A wide range of evidence points to rollover as a contributory factor in major fuel spillage incidents. Rollovers appear to be rare, and trends suggest they, and major spills from them, have become rarer over recent years, probably through preventative technologies and other interventions. These cannot, however, prevent all rollovers, e.g. those from a driver losing concentration or making sudden (evasive) steering inputs. When a rollover results in major spillage, a combination of overturning and sliding is usually involved, with tank rupture caused by puncturing impacts with road-side objects, as the tanker slides on its side. No evidence has been found to indicate that failures of circumferential welds have played any significant role in real-world fuel spillage incidents, although none of the non-compliant tankers are known to have been involved in such incidents.

Statistics from casualty accidents involving FL-registered six-axle articulated vehicles in GB (i.e. those with tractor units licensed to pull semitrailers carrying flammable liquids) were combined with evidence on damage-only incident frequencies. If the likelihood of involvement and spillage in a collision was identical for all such tankers, on average a collision of one of the 220 non-compliant tankers involving spillage could be expected to occur once every three years or so, and a rollover collision once every 2 - 3 years.

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[1] 'Short-term Fitness for Service Assessment of [non-compliant] Road Tankers, TWI (Draft) Report 23437/1/13, September 2013 and 'Project 23437 Contract Amendment: Additional FEA for assessment of [non-compliant] road tankers, TWI (Draft) Report 23437/2/13, October 2013.

It is possible that factors such as:
- the low centre of gravity of fuel tankers (compared to other types of tanker);
- a greater focus on road safety within the petroleum industry; and
- greater investments in safety technologies amongst the major oil company fleets

lead to accident, rollover and spillage frequencies for petroleum road fuel tankers somewhat lower than those experienced by other types of FL-registered articulated vehicles. This hypothesis could not be fully tested during the research. The safety record of petroleum tankers may well be better, on average, than these other tankers, but no amount of safety interventions can be certain to eliminate every road safety risk. If only those rollover cases known to have involved an articulated petroleum road fuel tanker over recent years are considered, the (lower bound) rollover collision frequency expectation for the 220 non-compliant tankers becomes once every 4 - 5 years on average (and the lower bound risk of rollover per tanker per year is estimated to be 0.1%). In probability terms, if the 120 non-compliant tankers thought (at the time of drafting this report) to still be in use on UK roads are likely to overturn at this same (lower bound) overall average frequency as all petroleum road fuel tankers, then there is estimated to be a 51% chance of at least one of them overturning in the next 6 years, and a 66% probability when a period of 9 years is considered.

Some stakeholders have suggested that the rollover risks for non-compliant tankers are even lower than these lower bound estimates, perhaps by a factor of two. Using their estimates means, however, that while the absolute rollover probabilities may be somewhat lower than those indicated above, they are of the same order of magnitude (30% and 42% probabilities for the 6 and 9 year periods respectively).

A topple test has been developed and shown to be both repeatable and a reasonable representation of real-world rollovers in terms of the velocity of impact of the tank with the ground. Two non-compliant, eight-banded tankers, one from 2008 and another from 2011, were subjected to this topple test, laden with water uniformly distributed across all compartments to give the same overall mass as if carrying fuel (70% by volume as water is denser than fuel)[2]. The performance of both tankers was similar, despite some differences in their detailed design and construction. These differences include the profile of the extrusion bands that join sections of the shell together, how the bulkheads/baffles have been welded to the extrusion bands, the presence and location of internal fillet welds and the extent of lack of fusion indications in the circumferential welds.

In both tests the tanks leaked from impact damage to an end bulkhead where it joined the extrusion band (which formed the end of the main tank shell). One leaked at the rear bulkhead and the other at the front, both at the top of the impact area. All of the compartments in the 2008 tank lost their internal integrity (although the leaks between them were very slow) and in the 2011 tank, internal integrity was lost between two pairs of compartments but not between all of them. Neither test showed any obvious external visual indication of failure of the circumferential welds resulting in leakage, but on closer inspection, an apparent through-wall crack along the rear circumferential weld at the top of the impact zone of the 2008 tanker was observed. In the context of inspecting the damage sustained by the tankers used in the testing, the manufacturer of the non-compliant tankers indicated that the damage around the joints between the extrusion band and the bulkhead/baffles for both tankers is consistent with real-world rollovers.

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[2] Testing with fuel was deemed infeasible, mainly, but not solely, due to safety and environmental concerns.

A finite-element (FE) model of the non-compliant tankers used in the topple tests, including both structural and fluid elements, was developed. Overall, the predicted structural deflections, fluid pressures, and significant bending moments and stresses agreed reasonably well with those measured in the tests. Further, the model predicted that the highest strains occurred in the bulkheads near the top and bottom of the deformed impact area on the tanker, at plastic levels which might lead to failure, as was found in the tests. The validated model was also used to assess the likely performance of the non-compliant tankers when loaded with fuel oil, with one compartment empty, and with petrol, in all compartments (both are industry practice). For all cases modelled, impact-related deformation of the bulkheads had a larger effect on the stresses within the tank structure than the effects of fluid pressure. The fuel oil and petrol modelling suggests that fuel loads may lead to more severe effects, such as higher deformations, than for the water loads tested, especially if a compartment is empty.

The highest bending stresses, derived by the fuel oil model, were found to be 254 MPa acting near the circumferential welds, together with much lower estimated membrane stresses. The limiting effect of the plastic deformations in the tank shell means that other loading scenarios would be unlikely to give significantly different results. In contrast, a 2-bar pressure impulse simulation (conventionally used as a design load case for rollover) generated bending stresses no higher than 150 MPa at the same location.

All welded structures contain imperfections of some kind. Good workmanship rules typically provide acceptance criteria based on quality control, but to assess fitness for service of the circumferential welds a more comprehensive approach has been used based on Engineering Critical Assessment (ECA) techniques. An ECA is an analysis, based on fracture mechanics principles, of whether a given flaw in a given welded joint is safe (from brittle fracture, fatigue or plastic collapse) under specified loading conditions.

Detailed examinations have revealed that lack of fusion indications may correspond to crack-like defects in the circumferential welds, so the ECA was used to determine the maximum acceptable defect size (also known as the critical crack size) under various loading conditions, such as normal operational conditions and during rollover events.

To determine the safe operating life of a component containing a defect, two main factors are required: the primary loads and the fatigue stresses. The primary loads allow for the determination of the critical defect size, and the fatigue stresses allow for the determination of how long it will take a sub-critical defect to grow by fatigue and become critical. The fatigue stresses under "normal operational conditions" were obtained from on-road tests with an instrumented tanker, scaled to represent the 220,000 km annual mileage on a mixture of different road types. The primary loads under "normal operational conditions" were defined to be the ADR design load cases, which the tanker should expect to encounter at any point during its operational life. The two rollover loading conditions were based on outputs from the fuel oil rollover model and, for completeness, the 2-bar pressure-impulse simulation.

The ECA related to the safe operating life of the circumferential welds found that:

- Providing an initial defect is present, the fatigue data (for a 2010 ten-banded tanker) identified the cradle positions above the fifth wheel coupling and above the front of the rear longitudinal support members as most susceptible to fatigue crack growth.
- Under normal operating conditions, the minimum critical defect height is greater than 2.0 mm and may be 4.0 mm or more. Variation in this defect height will depend on:

- the presence of an internal fillet weld between the toe of the extrusion band and the inner surface of the shell;
- the magnitude of misalignment between the shell and extrusion band; and
- the size of the weld cap.

- Assuming an initial defect size of 2 x 100 mm (i.e. a 2 mm deep by 100 mm long surface-breaking flaw) based on observations that such a flaw would not be unexpected, the fatigue life of the joint (i.e. the time required to grow the 2 x 100 mm defect to a critical size) is greater than 20 years when the said internal fillet weld is present and continuous (or suitable if intermittent).

- When a continuous (or suitable intermittent) internal fillet weld is not present, the fatigue life of the joint is influenced significantly by the misalignment and weld cap geometry. For this case, a parametric study involving over 300 simulations was used to derive a quadratic relationship between the fatigue life (assuming an initial 2 x 100 mm flaw) and a geometry parameter that incorporates the weld cap height and misalignment. This allows a conservative estimate of the fatigue life of a joint (without the said internal fillet weld) to be easily determined from a look-up table (derived from the quadratic relationship) using measurements of misalignment and weld cap height, which can be taken relatively quickly with a profile or laser gauge.

The ECA of the circumferential welds related to the rollover conditions found that:

- For the rollover case derived from the topple tests and associated FE modelling with fuel oil, and allowing for some ductile tearing to occur, the critical defect height in an "average" weld geometry is 1.1 mm when no internal fillet weld is present. Here the "average" weld geometry relates to measurements from the 2011 tanker and may, therefore, not be truly representative of all non-compliant tanker joints.

- This critical defect height agrees well with the experimental observation of the through-wall rupture of a circumferential weld resulting from a lack of fusion defect 1.0 mm deep and longer than 230 mm in a section taken from the impacted side of the 2008 tanker, paying due regard to the specific geometry of the weld involved. Less severe embedded flaws observed in the 2011 tanker have not been analysed to a similar degree but are considered to be much less problematic. The difference between the tankers is thought to be due to a change in the welding process during manufacture, which occurred in mid-2010.

- Considering the rollover load case derived from the pressure-impulse simulation, and allowing for some ductile tearing to occur, the critical defect height in the "average" weld geometry is 2.5 mm when no internal fillet weld is present.

- When a well-made and suitable internal fillet weld is present, the integrity of the tank in a rollover is not governed by the quality of the circumferential weld, but by the strength of the parent metal of the tank shell or other factors such as the bulkhead to extrusion band joint(s), which were seen to fail in the topple tests.

No evidence of fatigue crack growth was observed in the samples taken and examined from the non-compliant tankers assessed in this research.

A review of existing regulations identified two performance-based test procedures that may at least form starting points for the development of enhanced requirements for road fuel tankers, to further reduce the risks of major spillage in complex, but realistic, rollover events. These are the static rollover test used in UN(ECE) Regulation No. 66 (which is similar to the topple test used in this research) and the front pillar pendulum impactor test used in UN(ECE) Regulation No. 29.

1 Introduction

1.1 Project Background

Vehicles used to transport large quantities of dangerous goods, including petroleum products, must meet the requirements of the European Agreement on the Carriage of Dangerous Goods by Road (ADR)[3]. Following examination, certain petroleum road fuel tankers have been found not to be fully compliant with the provisions of Chapter 6.8 of ADR. Amongst other things, the tanks were not radiographed during initial inspections, and those radiographed since are seen to exhibit extensive 'lack of fusion' indications in the circumferential welds that join the shell of the tank to the extrusion bands.

Following an initial technical assessment[4] of the circumferential welds, which showed that the welds might rupture under rollover and ADR load conditions, the Department for Transport (DfT) commissioned further research to assess the safety of these non-compliant tankers relative to ADR requirements. The researchers were also tasked to explore any opportunities arising for better regulation which could improve both the safety and efficiency of all petroleum tankers. The scope of this research did not include other issues applying to the non-compliant tankers, such as manway flanges and filling/offtake nozzles.

The research consisted of three work packages:

- WP1 – Full scale testing and associated modelling, led by the Health and Safety Laboratory (HSL)

- WP2 – Detailed Fracture and Fatigue Engineering Critical Assessment (ECA), led by TWI Ltd (TWI, formerly known as The Welding Institute).

- WP3 – Accident data and regulatory implications, and production of an overall summary report, led by TRL Ltd (TRL, the Transport Research Laboratory).

This summary report is thus a deliverable from Work Package 3, but, in partnership with leaders of the other two Work Packages, it pulls together all the results, summarises them, draws out the key conclusions, and presents the findings impartially and in a style and format suited to a non-technical, policy and legislative audience. Full technical details are provided in specific reports from each of the Work Packages.

1.2 Research Consortium

Each Work Package was led by one of three organisations with particular expertise and experience relevant to that activity:

The Health & Safety Laboratory (**HSL**) is one of the world's leading providers of health and safety solutions to industry, government and professional bodies. HSL is an independent agency of the Health and Safety Executive (HSE), and employs over 350 scientific, medical and technical specialists. Its main site in Buxton has extensive testing, analysis and modelling facilities.

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[3] *European Agreement concerning the International Carriage of Dangerous Goods by Road* (ADR)
[4] 'Short-term Fitness for Service Assessment of [non-compliant] Road Tankers, TWI (Draft) Report 23437/1/13, September 2013 and 'Project 23437 Contract Amendment: Additional FEA for assessment of [non-compliant] road tankers, TWI (Draft) Report 23437/2/13, October 2013.

TWI Ltd (TWI, formerly known as The Welding Institute) is one of the world's foremost independent research and technology organisations, with expertise in solving problems in all aspects of manufacturing, fabrication and whole-life integrity management technologies. Established in Cambridge, UK and with facilities across the globe, the company has a first class reputation for service through its teams of internationally respected consultants, scientists, engineers and support staff. Its core expertise is in the testing, analysis and assessment of materials and joining technologies.

TRL Ltd (the Transport Research Laboratory) provides independent and impartial world-class research, consultancy, testing and certification for all aspects of transport. TRL's vehicle safety scientists and engineers have been instrumental in improving the structural design and integrity of vehicles over several decades and it has considerable expertise in accident research studies, database analyses and policy evaluation, including many studies for the Department for Transport.

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 Work Packages. This peer review function was further aided by the involvement of a fourth organisation; Cambridge University Engineering Department (CUED).
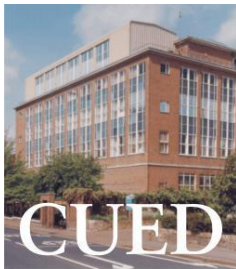
CUED creates world-leading engineering knowledge that fosters sustainability, prosperity and resilience. It shares this knowledge and transfers it to industry through publication, teaching, collaboration, licensing and entrepreneurship. CUED serves as an international hub for engineering excellence. Dr Michael Sutcliffe is a Reader in CUED's Mechanics, Materials and Design Division and contributes to the work of the Centre for Sustainable Road Freight, with specific expertise in the mechanical behaviour of materials.

1.3 Work Package structure

The major outputs from each Work Package and the dependencies between them are summarised diagrammatically in Figure 1.

The WP numbering convention follows simply the order in which the individual WP contracts were tendered. The ordering used in the diagram is more logical from the perspective of the tasks performed, and is followed in the remainder of this report.
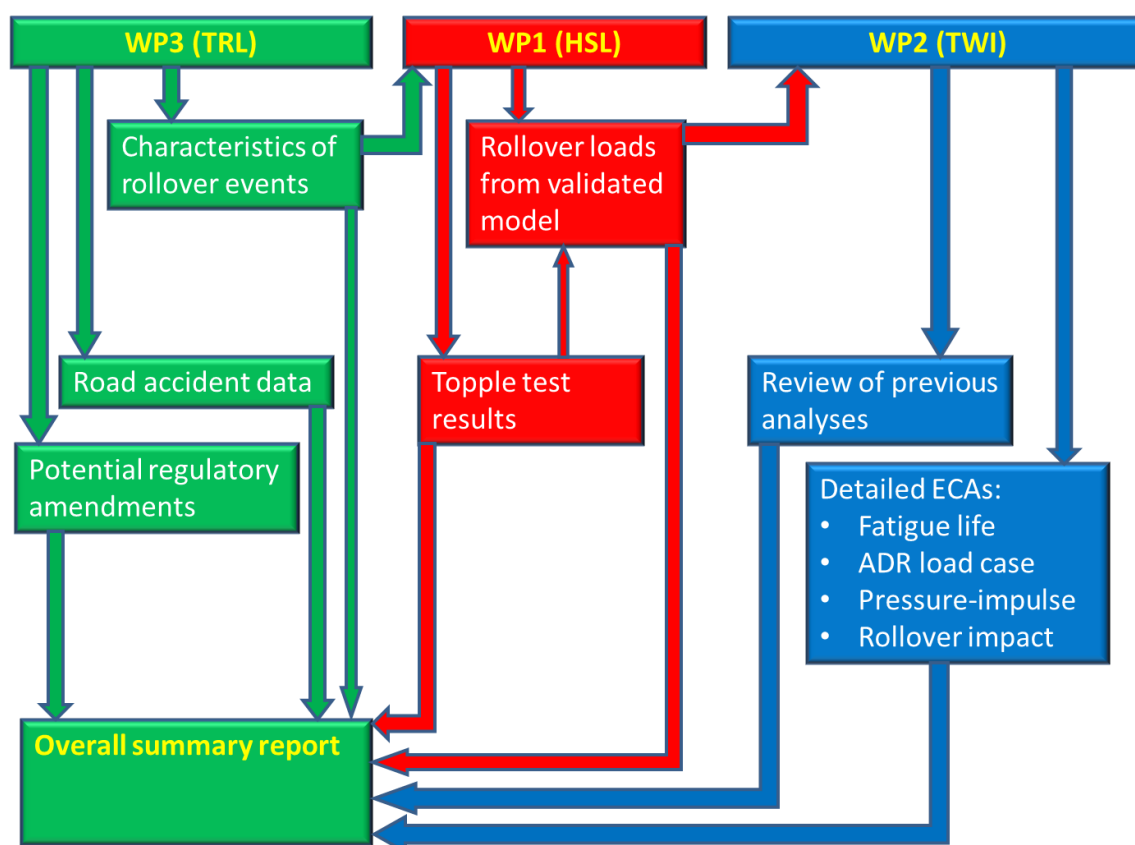
Figure 1. Work Package task structure and inter-dependencies

1.3.1 Work Package 3 (WP3) – accident data and regulatory implications

The fundamental objectives of this Work Package (aside from the production of this Summary Report and contributing to the peer review activities described above) were to:

- Determine representative rollover and collision scenarios and loading conditions.
- Identify regulatory implications and potential amendments.

Its primary relationship with the other Work Packages was to provide information/evidence to HSL (WP1) on the kinematics of real-world tanker rollover events. It also set out to provide real-world context for the other Work Packages, particularly the likely frequencies and severities of accidents involving fuel tankers.

1.3.2 Work Package 1 (WP1) – full scale testing and associated modelling

This Work Package set out to:

- Develop an independent non-proprietary structural hydrodynamic model of the non-compliant tankers, validate this model against the results of tanker tests, and report modelling findings.
- Design, construct and commission a rig to test tankers, including selecting and procuring suitable instrumentation for data gathering.
- Undertake tests on tankers, including preparing the tankers, assessing the tanker test method and results, and reporting the findings.

- Determine suitability of tankers for large scale tests and acquire tankers, as appropriate, in accordance with project objectives as specified by DfT.

- Capture collision and/or deformation data from relevant impacts, for example by laser scanning, to corroborate the modelling and tanker tests, and reconcile any inconsistencies.

Its main links to other Work Packages were first, to use information provided by WP3 to assess the relevance of the full scale test method to real-world accidents and, second, to provide load case data (from the validated model) for TWI (WP2) to use in the Engineering Critical Assessment of a rollover impact.

1.3.3 Work Package 2 (WP2) – detailed Engineering Critical Assessment

The major tasks in this Work Package were to:

- Determine the typical in-service life cycle fatigue loadings at worst case locations on the circumferential weld seam, including, when appropriate, the effects of filling and dispensing from compartments;

- Review a proprietary finite element model of the non-compliant tanker as well as the crack growth and leak-before-break and associated technical documents and studies undertaken previously by its manufacturer and TWI;

- Address deficiencies as appropriate (such as fracture toughness properties, fatigue crack growth rates and weld residual stresses, wherever possible using strength and fatigue tests of samples taken from tankers to validate the model);

- Engage with the non-compliant tanker manufacturer to solicit and incorporate views as appropriate;

- Undertake a detailed engineering critical assessment (ECA) to predict crack growth, likely fatigue life of weld seams and defect sizes under suitable loading conditions, incorporating geometric variability;

Its major relationship to the other Work Packages was with WP1 for the engineering critical assessment of the rollover impact condition.

1.4 Reporting structure

The following three Chapters of this report summarise the research methodologies and key findings from each of the Work Packages in turn; WP3 (Chapter 2), WP1 (Chapter 3) and WP2 (Chapter 4).

Chapter 5 combines the results from all the Work Packages, discusses the findings and draws a set of overall conclusions that the researchers believe will be of most relevance to policy makers and legislators.

1.5 Acknowledgements

The authors would very much like to thank the very many individuals and organisations who have contributed to and assisted this research. The fact that such a comprehensive programme of work has been carried out over such a short period of time (just ten months) is testament to their hard work, dedication, collaborative spirit and expertise.

2 WP3 – Accident data and regulatory implications

2.1 Research methods

WP3 was divided into two tasks. The aim of Task 1 (to determine representative rollover and collision loads) was to provide background intelligence on fuel tanker accidents, e.g. their frequency and how often rollovers and rear impacts lead to fuel spillage and/or tank rupturing, and in what specific circumstances.

This helped define the overall risks and informed the test and modelling work (other WPs) to ensure simulated conditions were broadly realistic.

Tank rupture was thought, at the outset of the project, likely to be rare, so the research net was cast wide and involved an international review of multiple sources:

   i.   Published international research literature (from 1995 – 2014)

   ii.   DfT statistics/records (including STATS19[5] data, ADR and RIDDOR[6] reports)

   iii.   Local news media articles

   iv.   Detailed (in-depth) truck accident databases (RAIDS)[7]

   v.   Stakeholder surveys (of tanker operators, repairers and international experts)

Further details regarding the methodologies employed under each of these headings are given in the following sections.

The results were also used to inform Task 2 (to identify regulatory implications and potential amendments), supplemented by a dedicated review of potentially relevant current legislation.

2.2 Results

2.2.1 Research literature

This task involved a search using TRL's access to various international research literature databases and the internet. Research papers and articles were identified using the search terms "tanker", "tank trucks" or "liquid cargo handling" AND "road" AND "incident", "rollover", "roll over", "collision", "spillage", or "rupture". Date limits of 1995 – 2014 were used.

In total, 116 papers and articles were identified using these search terms. No directly relevant UK published research was found. However, various studies of relevance were identified internationally, in particular from the USA and Germany but also Spain, Netherlands and China. The following paragraphs describe the most directly relevant literature in more detail.

To inform the identification of regulatory implications and the potential for amendments to current regulations, some existing regulations and international standards were also reviewed, and the results are summarised in the following section.

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[5] The database of police-reported road accidents in Great Britain

[6] Reporting of Injuries, Diseases and Dangerous Occurrences Regulations

[7] The Road Accidents In-Depth Studies database

"Tanker trucks in the current accident scene and potentials for enhanced safety", Gwehenberger & Langweider, 2002 (Germany)

This analysis draws heavily on the earlier THESEUS project (discussed below). The main risk leading to hazardous material spillage was reported to be single vehicle accidents with rollover and rear/side impacts with other HGVs.

The report highlights the role of local and "global" (distributed) loads – for local loads, material strength properties determine failure threshold, but for global loads "failure tends to occur where abrupt transitions in rigidity, for example in the bases, bracing rings or welded bracing bands, impede distortion."

The report's authors criticised the move to aluminium tanks; "a stainless steel tank with a wall thickness of 3 mm is almost twice as safe for transporting class 3 hazardous substances by road than a tank made of aluminium alloy of the conventional type" [5 mm wall thickness].

"THESEUS – Maximum possible tanker safety through experimental accident simulation", Rompe & Heuser, 1996 (Germany).

Over the course of a nine year research programme, thirty six crash tests and twelve dynamic overturn tests were performed and the results are summarised. The test parameters used were devised from an analysis of 232 road accidents of tankers (including rigid tankers, articulated tankers and drawbar trailer tankers) involving the risk of spillage of dangerous goods.

Single vehicle accidents accounted for 44% of all the accidents studied, but 71% of the spillage cases – half on bends, half on straight sections of road. The average vehicle speed before overturn was 48 km/h, with 28% occurring above 70 km/h. Rear and side impacts with other vehicles accounted for 32% and 28% respectively. The average rear impact speed was 20 km/h.

The overturn tests used an articulated tanker, driven at 50 km/h on a curve, rolling onto a smooth road surface. Without any obstructions, the overturned tankers skidded for 25-35 m before coming to rest. Tank deformations of 35-80 mm were typical, but no failures/spillages were recorded.

Rear impacts from another HGV at 25-27 km/h led to spillage in 63% of the tests, from local intrusion of impacting parts.

German HGV accident statistics. Statistisches Bundesamt, 2013. Verkehrsunfaelle - Unfaelle von Gueterkraftfahrzeugen im Strassenverkehr.

The German Federal Statistical Office publishes an annual report on road accidents of HGVs. The accident numbers of tankers transporting dangerous goods since 1999 are summarized, relating to all reported accidents severe enough to have either caused at least one casualty or enough damage to mean at least one vehicle involved had to be towed away.

The statistics are broken down by accident severity, but only as far as injury accidents and damage-only incidents. Generally speaking, injury accidents account for about two-thirds of the cases, and a similar proportion of the spillage cases. Between 1999 and 2003, 14% of accidents involved spillage, whereas in the years from 2009 to 2012, that frequency had fallen to between 0% and 5% (2% on average). No other data is contained in the report on types of accidents or the volume of spilt goods, but these data are useful for basic incident frequency and risk analyses.

"Rollover stability of tank trucks, test and calculation requirements based on ECE 111 regulation", Martin et al, 2010 (Spain).

This report starts by reviewing Spanish accident statistics (from ADR incident reports) and then describes two new approaches to improving the accuracy of the static roll stability calculation required by ECE Regulation No. 111[8].

The Spanish incident report data has 43 spillage cases and 45 without. The most frequent accident types involving spillage are rollover and running off road (60%), followed by collisions with other vehicles (15%).

"Tank lorry fires involving dangerous goods", Dutch Safety Board, 2006 (Netherlands).

This report describes the reporting system in the Netherlands and finds that it is not properly fulfilled nor enforced. Official accident statistics are also reported to be deficient in that they cannot adequately record accidents involving dangerous goods.

The report then focuses on fires involving tank vehicles carrying dangerous goods, describing six such incidents in the Netherlands between 1999 and 2005. Specific risk factors identified include the size, construction and placement of vehicle fuel tanks.

"The dynamics of tank-vehicle rollover and the implications for rollover-protection devices", UMTRI, 1998 (USA) & "Cargo tank rollover force verification", FMCSA, 2006 (USA).

These reports describe track tests and computer simulations of tanker rollover accidents (the 2006 report covers track tests carried out to validate the simulations reported by UMTRI in 1998).

A wide range of roll inducing manoeuvres were simulated (126 each for five different tractor-semitrailer combinations). The simulation runs went up to the moment the tank contacted the ground. Three basic scenarios were defined; mild, 90 degree roll with slide and 180 degree rollover.

In a mild rollover, simulated roll rates at impact ranged from 100 – 150 deg/s (1.75 – 2.60 radians/s).

Where a vehicle landed on its side and then slid, impacts with vertical objects (e.g. guardrails, retaining walls or embankments) were simulated to occur at velocities (perpendicular to the road) of 20 - 40% of the initial forward speed of the vehicle.

In the more dramatic simulated events, the vehicle could become airborne and roll rapidly enough so that the impact with the ground was with the roof of the tank. Downward velocities ranging from 1.8 – 9 m/s were simulated.

The 2006 tests showed generally good correlation with the 1998 simulations.

"Hazardous materials serious crash analysis: phase 2", FMCSA, 2005 (USA).

For this study, 1,629 crashes were analysed (from 2002). 914 involved class 3.0 products (flammable liquid hazardous materials), and 20% resulted in a spillage.

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[8] UNIFORM PROVISIONS CONCERNING THE APPROVAL OF TANK VEHICLES OF CATEGORIES N AND O WITH REGARD TO ROLLOVER STABILITY - applies to the rollover stability of tank vehicles of category N2, N3, O3 and O4 intended for the carriage of dangerous goods as defined in the ADR agreements.

Rollover was found to significantly increase the probability of a spill, and the more load carried, the greater the likelihood of a rollover.

Rollover, loss of control and run off-road were closely associated with Class 3.0 spills.

"Cargo Tank Roll Stability Study", FMCSA, 2007 (USA).

Crash statistics (all tanker accidents not just hazmat) were used to evaluate four complementary mitigation strategies; driver training, electronic stability aids, new vehicle designs and highway design.

The authors found an average of 1,265 cargo tank rollovers per annum (in 2002). Run-off road with a tripped rollover was the most common scenario (47%) identified, with un-tripped rollovers accounting for a further 14%.

An evasive manoeuvre by the driver was a factor in 5-10% of rollovers. Driver error was a factor in 75%.

"Simulation of Nonlinear Dynamics of Liquid Filled Fuel Tanker Shell Structure subjected to Rollover Collision with Validation", Park, Gilmore & Singer, 1998 (USA).

This report is mainly concerned with the modelling of a tank structure, but does describe some earlier (1983) accident data and highlights that "puncture of tank shell is a major source of spillage both in crash and in rollover accidents".

"A survey on hazardous materials accidents during road transport in China from 2000 to 2008", Yang et al, 2010 (China).

This report describes an analysis of 322 on-road hazmat accidents in China. Driver errors leading to collisions with other vehicles or improper emergency responses accounted for 60% of the cases examined. 85% of the cases involved release of hazardous material, usually (64%) without subsequent fire/explosion or gas cloud. Fires occurred in 10% of cases, and explosions in 3%.

2.2.2 *Current regulations and standards*

In preparation for possible future regulatory amendment proposals, the main goal of this task was to establish whether certain aspects of the existing regulatory environment (relevant to heavy goods vehicles and/or the carriage of dangerous goods by road) could be of potential use in enhancing the regulations affecting petroleum road fuel tankers. For example, if these existing regulatory mechanisms already represent realistic heavy vehicle impact and rollover scenarios, adapting them for ADR might offer an easier implementation path than developing an entirely bespoke set of procedures.

2.2.2.1 *Current ADR requirements*

ADR[9] section 6.8 contains "requirements for the construction, equipment, type approval, inspections and tests, and marking of fixed tanks (tank-vehicles), demountable tanks and tank-containers and tank swap bodies, with shells made of metallic materials, and battery-vehicles and multiple element gas containers (MEGCs)". It therefore provides the

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[9] European Agreement Concerning the International Carriage of Dangerous Goods by Road, applicable as from 1 January 2013.

main technical requirements concerning the design, construction and testing of the petroleum road fuel tankers of interest to this study.

These requirements include some non-mandatory provisions for "protection of the tank against damage through lateral impact or overturning", but they are purely design-based, rather than requiring any performance-based testing. The design requirements are provided in section 6.8.2.1.20. For shells with a circular or elliptical cross-section having a maximum radius of curvature of 2 m (as would be the case for the tank vehicles of interest), there is "protection against damage" when:

"*the shell is equipped with strengthening members comprising partitions, surge-plates or external or internal rings, so placed that at least one of the following conditions is met:*

- *Distance between two adjacent strengthening elements of not more than 1.75 m.*

- *Volume between two partitions or surge-plates of not more than 7,500 l.*

The vertical cross-section of a ring, with the associated coupling, shall have a section modulus of at least 10 cm^3.

External rings shall not have projecting edges with a radius of less than 2.5 mm.

Partitions and surge-plates shall be dished, with a depth of dish of not less than 10 cm, or shall be corrugated, profiled or otherwise reinforced to give equivalent strength. The area of the surge-plate shall be at least 70% of the cross-sectional area of the tank in which the surge-plate is fitted.

The thickness of the partitions and surge-plates shall in no case be less than that of the shell."

There are separate requirements for thicknesses of shell material, but if the above damage protection requirements are met, a reduced shell thickness is permitted. Mild steel shells with a diameter exceeding 1.80 m, for example, need only be at least 4 mm thick if they meet the damage protection requirements, whereas in the absence of such protective measures they must be at least 6 mm thick. Equivalent thicknesses apply to all other materials, e.g. stainless steel or aluminium alloy.

There are also some requirements (6.8.2.1.28) for the protection against damage caused by overturning of fittings mounted on the upper part of the tank. Again, these are purely design-based, however:

"*This protection may take the form of strengthening rings, protective canopies or transverse or longitudinal members so shaped that effective protection is given.*" No guidance is provided on what constitutes "effective" in this context.

2.2.2.2 Current UN(ECE) requirements

UN(ECE) Regulation No. 29 concerns the protection of the occupants of the cab of a commercial vehicle. It contains performance-based test requirements for the strength of the cab in various simulated impact and rollover conditions. N_3 vehicles and N_2 vehicles > 7.5t must pass three separate tests:

- Test A – frontal impact

- Test B – frontal pillar impact

- Test C – roof strength

Test A (Figure 2) involves a large (1,500 kg) flat, rectangular metal pendulum (2.5 m wide and 0.8 m high) being swung into the front of the cab.
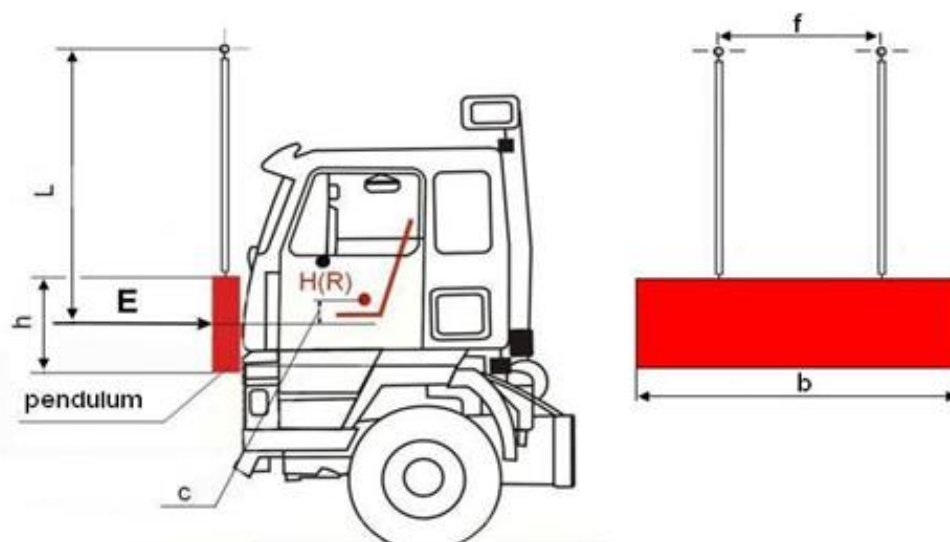

Figure 2. Frontal impact test (Test A, Regulation 29)

These conditions are intended to represent an HGV colliding with the rear of another heavy vehicle, striking the main structural parts of that vehicle within the chassis and/or load platform. The impact energy is broadly equivalent to an impact at around 6 km/h for a 40 t vehicle, so it is not particularly severe but it does help to ensure that the driver (with seat belt) is able to remain in the cab and not be hit by intruding structural components in such an impact.

For such a test to be useful as a performance-based regulatory mechanism for fuel tankers, it would need to represent a rear impact in to the back of the tank (from a following heavy goods vehicle). However, in such a scenario, the main structural components of the impacting vehicle would normally be somewhat lower than the structure of the tank trailer. Only the relatively 'soft' upper frontal structure (in the windscreen area) would tend to interact with the tank itself. A large rigid and heavy metal pendulum would not properly represent such an impact. That said, such a pendulum impact might be relevant to assess the integrity of the area at the rear of the tanker below the tank (at a height designed to be similar to the chassis members of the impacting vehicle), where pipework and other components potentially carrying fuel might be located.

Test B (Figure 3) is a relatively recent introduction to Regulation 29. It involves a cylindrical pendulum weighing at least 1,000 kg, with a diameter of 550 – 650 mm and length of at least 2.5 m. It is impacted horizontally, parallel to the median longitudinal plane of the vehicle such that its centre of gravity at impact is midway between the lower and upper windscreen frame. This test is intended to replicate an overturned vehicle sliding on its side into a secondary (frontal) impact with a tree. As with Test A, an occupant survival space needs to be maintained for the test to be passed.
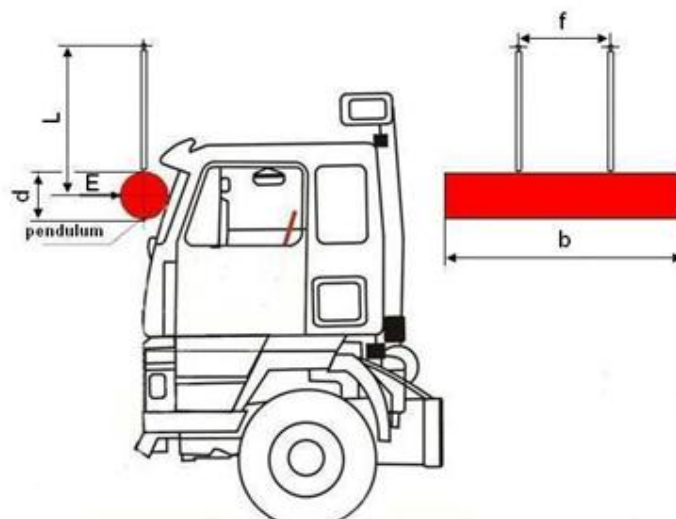
Figure 3. Front pillar test (Test B, Regulation 29)

While such a frontal impact would be unlikely to be relevant to the tank of a road fuel tanker in an overturn (by ninety degrees), dropping a similarly sized "pendulum" onto the roof of a tank could potentially simulate a tanker overturning and sliding out of the curve into a tree. The Regulation 29 impact energy could be achieved by dropping the 1,000 kg pendulum/cylinder from a height of 3 m above the roof of the tank.

Test C (Figure 4) combines a dynamic impact to the upper side of the cab with a static roof strength test.
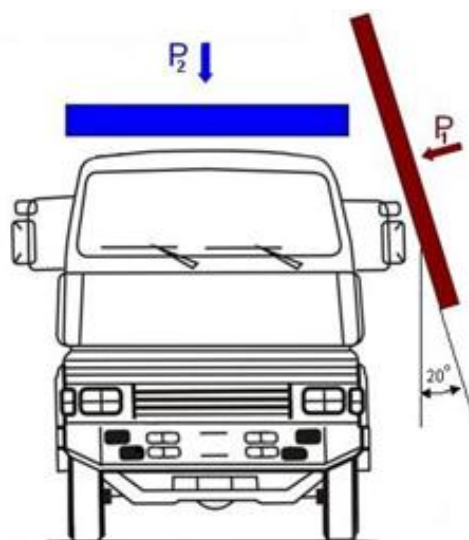
Figure 4. Roof strength (Test C, Regulation 29)

The dynamic impact preloads the structure via a 1,500 kg rigid, flat, rectangular impactor. The static load then simulates much of the weight of the cab itself acting on the (fully overturned) roof. The static load is 98 kN (equivalent to 10 tonnes) or less and corresponds to the maximum authorised mass of the front axle or axles of the vehicle. As with Tests A and B, an occupant survival space needs to be maintained for the test to be passed. The load is applied via a rectangular and flat device larger than the roof of the cab (i.e. the load is distributed over the whole of the roof area).

For a road fuel tanker, an equivalent test would need to impose the load over the whole of the uppermost part of the tank and be around 235 kN to simulate the combined maximum weight of the tank semitrailer (24 tonnes for a tri-axle).

UN(ECE) Regulation No. 66 concerns the strength of the superstructure of large passenger carrying vehicles (single deck buses and coaches). The requirements involve ensuring a survival space is provided in the event of a partial overturn. The exact rollover condition is shown in the figure below, and involves a simple topple but into a rigid ditch set 800 mm below the level of the road surface.

The Regulation contains various alternative (equivalent) approval test methods, including a rollover test on body sections which are representative of the complete vehicle, quasi-static loading tests, quasi-static calculations based on the results of component tests and computer simulation via dynamic calculations.
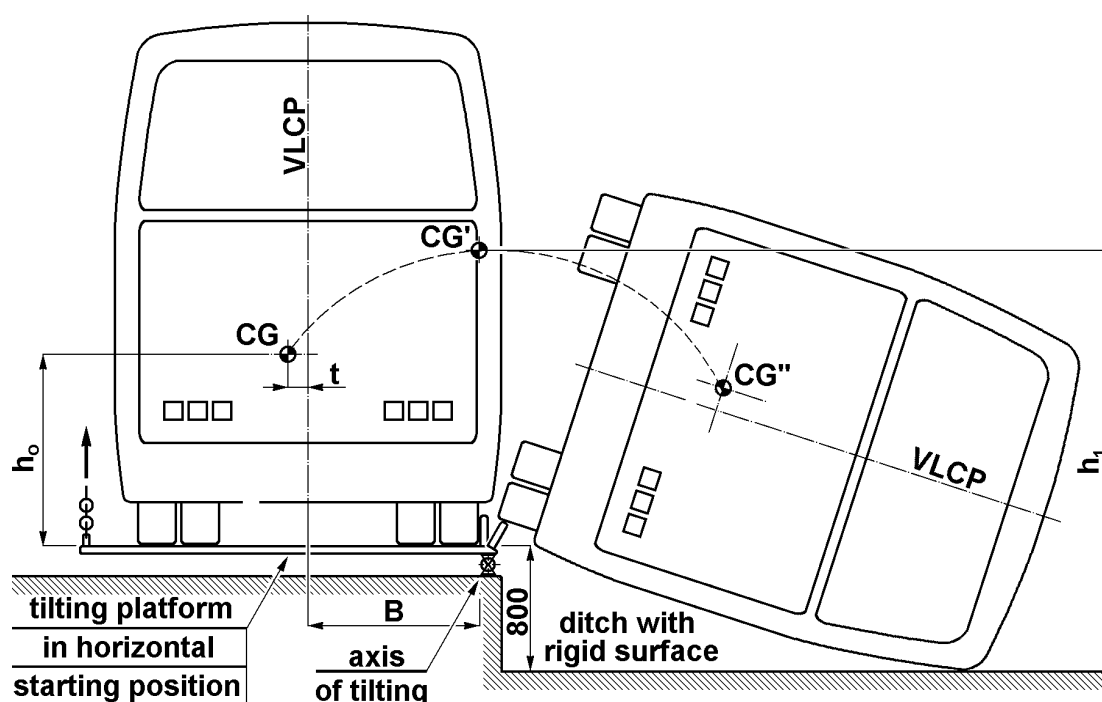
Figure 5. Rollover test specification, Regulation 66

Although the survival space criteria would obviously not be relevant to the tank of a road fuel tanker, the basic test method may well represent a rollover situation, and of somewhat greater severity than a simple topple onto a level road surface. The test does not, however, involve any forward or sideways motion of the overturned vehicle so does not replicate any post rollover sliding along the ground or subsequent impacts.

2.2.3   Analysis of accident data and incident reports

Data was obtained from several different sources to aid the analysis. This included STATS19 data, BBC news reports, ADR incident reports, RIDDOR reports and two specialist, in-depth HGV accident databases.

STATS19 provides data from all police reported road accidents involving personal injury in Great Britain. Using data held by the Driver & Vehicle Standards Agency (DVSA), it

was possible to identify injury accidents involving tankers licensed to carry flammable liquids, and discover the circumstances of each accident.

BBC news articles from between 2009 and 2014 were also reviewed to gain an approximation of the frequency of rollover and spillage accidents of tankers in injury and non-injury accidents. However, data collected from this source was treated carefully due to the tendency of the media to mostly report particularly exceptional accidents such as events involving fatalities, spillages or road closures creating severe delays. The data was compared with ADR and RIDDOR reports provided by DfT and the Health and Safety Executive (HSE) in order to establish any levels of underreporting by operators of accidents involving dangerous goods vehicles.

ADR incident reports must be produced by operators and supplied to DfT whenever a vehicle carrying dangerous goods is involved in a serious accident where the load is released or has a high probability of being released or causes injury. In addition to the ADR report, RIDDOR requires employers to report and keep records of certain 'dangerous occurrences'. Incidents of interest involving flammable liquid tankers needed to be reported under RIDDOR up until October 2013. Since then, HSE advise that amended rules mean only ADR reports are required, unless there is an unintentional spillage of 500 kg or more. It was expected that BBC news reports would exist for many of the officially reported incidents as they are serious enough to be considered newsworthy. Incidents involving personal injury will also be recorded in STATS19.

Road Accident In-Depth Studies (RAIDS) is an in-depth investigation study that will provide a unique source of data on accident causation and consequences, contributing evidence to underpin the development and implementation of measures to reduce risk and mitigate injuries. RAIDS incorporates the historic data from previous in-depth accident studies, including the Heavy Vehicle Crash Injury Study (HVCIS) and Truck Crash Injury Study (TCIS). Tanker accidents within each database were examined.

The following section describes a review and analysis of this collision and incident data, with the objective of identifying representative rollover and collision loads for petroleum road fuel tankers. The relationship between these data sources can be seen in Figure 6.
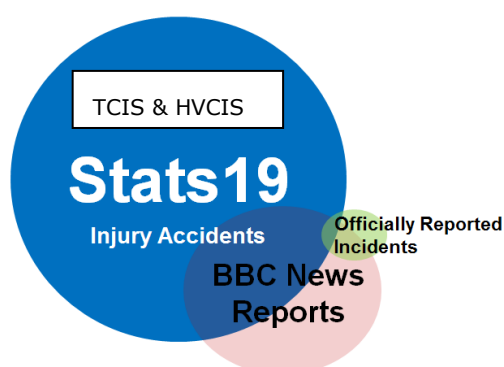
Figure 6. Sources of information analysed

2.2.3.1   *Detailed STATS19 data from accidents involving FL-registered vehicles*

The analysis was limited to articulated vehicles with a 3+3 wheel plan above 7.5 tonnes mgw (in the following referred to simply as '6-axle artics') because this is the most commonly used type to transport petroleum. A sub-group of these 6-axle artics are certified under ADR Regulations to carry flammable liquids (FL), i.e. are registered as 'FL

vehicles'. Analysis of vehicle certification data by DVSA provided the vehicle stock numbers for both groups as detailed in Table 1 (more recent data could not be obtained). Note that FL certification under ADR Regulations needs to be obtained for both the tractor unit and the trailer, but only the number of tractor units could be obtained from DVSA data. Because an FL trailer must be towed by an FL tractor unit, a ratio of one tractor unit to one trailer was assumed for the following analysis.

Table 1. Number of registered vehicles (years 2007-2012)

Year	6-axle artic vehicle stock	Of which, FL vehicle stock
2007	77,489	2,363
2008	76,622	3,037
2009	74,359	3,270
2010	76,201	3,626
2011	78,481	3,924
2012	80,389	4,236
Average	**77,256.8**	**3,409.3**

It was possible to identify FL vehicles involved in collisions by linking the STATS19 data[10] to the DVSA data through the vehicle registration mark of the FL tractor unit. These were identified by the DfT for the years 2007 to 2013. For this seven year period, 370 six-axle articulated FL vehicles were identified in the collision data, involved in 369 collisions[11]. The following sections provide an overview of the collision involvement rate to provide an estimate of the frequency of potential spillage incidents involving non-compliant FL tankers, and a comparison of the collision typology of both groups in order to reveal potential indications of higher risk collisions of FL vehicles.

Of particular interest for this study is the collision involvement rate of FL vehicles compared to all 6-axle artics. The numbers provided in Table 2 are annual averages for the years 2007 to 2012 (vehicle fleet data for 2013 was not available). It can be seen that the annual involvement rate of FL registered vehicles in injury collisions is almost 20% lower than the rate of all 6-axle artics. This trend is even more marked for rollovers where the involvement rate is 43% lower. The reasons for the lower involvement rate cannot be derived from this analysis; however, it can be assumed that better driver education and commonplace fitment with Electronic Stability Control (ESC) contribute to the reduced rates. ESC is particularly effective in preventing some rollover collisions.

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[10] Data are Crown Copyright and are reproduced with permission of the Department for Transport.

[11] "Collisions" means accidents or incidents rather than individual collisions within a single accident or incident

Table 2. Average annual collisions and collision involvement rate (average numbers for 2007-2012)

	FL vehicles		6-axle artics	
	Annual collisions	Annual collisions per 100 registered vehicles	Annual collisions	Annual collisions per 100 registered vehicles
All injury collisions (STATS 19)	54.0	1.58	1,519.5	1.97
of which:				
Rollover collisions	3.7	0.11	145.2	0.19
Non-rollover collisions	50.3	1.48	1,374.3	1.78

In the seven year period from 2007 to 2013, a total of 25 rollover collisions involving FL vehicles were recorded in STATS19. An analysis of the rollover collisions on a yearly basis indicates that the rollover involvement rate for all 6-axle artics reduced by half between 2007 and 2012 and appears to show a downward trend over time. The involvement rate of FL vehicles, however, does not show a marked downward (or upward) trend over the years. The fluctuations between years are naturally higher due to the low absolute case numbers (varying between 0 and 7 cases in any one year), which is why average numbers from the period 2007-2012 will be used as best estimates for the subsequent calculations related to FL vehicles.

In addition to these recorded injury collisions, an unknown number of damage-only collisions occurred. Official data from Germany (described in the literature review) indicates that for every two injury collisions involving an FL tanker there was another one involving damage only (severe enough for a vehicle being towed away from the scene). Based on an analysis of news reported collisions (described later in this Chapter), a higher ratio of one severe damage-only collision for each injury collision is possible. Using these ratios as a guide, a full estimate for GB would thus be that there are around 81 to 108 FL vehicle collisions each year severe enough to cause an injury or tow-away damage (around 2.38 to 3.17 per 100 registered FL vehicles) of which 5 to 7 involve the FL vehicle rolling over (around 0.16 to 0.22 per 100 registered FL vehicles).

As reported as part of the literature review, over recent years in Germany, between 0% and 5% of severe collisions there have involved spillage of load. It could therefore be anticipated that there might be up to 5 spillage incidents involving 6-axle FL vehicles >7.5 tonnes mgw per annum in GB, if the situation in GB is similar to that in Germany.

Table 3 provides an overview of the types of objects hit by FL vehicles and the comparator group in collisions. Objects hit off-carriageway by an overturned tanker present a risk of rupturing the tank, thus increasing the risk of load spillage. 60% of the FL vehicle rollover collisions involved hitting an object off-carriageway, i.e. a total 15 cases in the 7-year period analysed. This proportion is similar to the comparator group. Less than 4% of the FL vehicles that did not overturn hit such an object.

Table 3. Objects hit off carriageway in collisions, 2007 - 2013

Object hit off carriageway	FL articulated vehicle				6-axle artic			
	Rollover		No rollover		Rollover		No rollover	
No off carriageway object hit	10	40.0%	332	96.2%	437	43.5%	9,409	95.0%
Sign / post / pole / tree	4	16.0%	5	1.4%	172	17.1%	129	1.3%
Crash barrier / wall or fence	6	24.0%	4	1.2%	221	22.0%	204	2.1%
Entered ditch	4	16.0%	2	0.6%	64	6.4%	48	0.5%
Other permanent object	1	4.0%	2	0.6%	110	11.0%	110	1.1%
Total vehicles	**25**	**100.0%**	**345**	**100.0%**	**1,004**	**100.0%**	**9,900**	**100.0%**

The proportion of collisions happening at roundabouts is markedly higher for rollover than non-rollover cases (Table 4) and this proportion is similar for FL vehicles and all six-axle artics.

Table 4. Breakdown by detail on junction situation, 2007 - 2013

Junction detail	Coll. involving FL vehicle				Coll. involving 6-axle artic			
	Rollover		No rollover		Rollover		No rollover	
Roundabout / mini roundabout	9	36.0%	31	9.0%	361	36.6%	985	10.3%
Slip road	2	8.0%	21	6.1%	44	4.5%	511	5.3%
Private drive or entrance	0	0.0%	12	3.5%	3	0.3%	293	3.1%
Other junction	1	4.0%	62	18.0%	79	8.0%	1,777	18.6%
Not at or within 20m of a junction	13	52.0%	218	63.4%	498	50.6%	5,988	62.7%
Total collisions	**25**	**100.0%**	**344**	**100.0%**	**985**	**100.0%**	**9,554**	**100.0%**

2.2.3.2 *Officially reported incidents (ADR and RIDDOR) and traffic logs*

RIDDOR reports classed under 'tanker incident' or 'release, escape of substances' from the years 2011 – 2013 were provided by the HSE. Further information on ADR tanker incidents was provided by DfT from traffic logs and ADR incident reports from 2005 – 2013. The traffic logs record all incidents on the major road network where a traffic incident involved a dangerous goods vehicle. The ADR incident reports are (or at least should be) submitted to DfT by vehicle operators whenever their vehicles are involved in a serious accident or collision.

Under the terms of ADR section 1.8.5, a report has to be filed by the loader, filler, carrier or consignee

> *"if a serious accident or incident takes place during loading, filling, carriage or unloading of dangerous goods on the territory of a Contracting Party (…)".*

A 'serious' accident in this context is further defined as being:

> *"if dangerous goods were released or if there was an imminent risk of loss of product, if personal injury, material or environmental damage occurred, or if the authorities were involved and one or more of the following criteria has/have been met:*

> *Personal injury means an occurrence in which death or injury directly relating to the dangerous goods carried has occurred, and where the injury*

> *(a) Requires intensive medical treatment;*

> *(b) Requires a stay in hospital of at least one day; or*

> *(c) Results in the inability to work for at least three consecutive days.*

> *Loss of product of Class 3.0 flammable liquids means the release of dangerous goods in quantities of 1,000 kg / 1,000 litres or more".*

Incidents, even rollovers, quite rarely lead to loss of load in these quantities or lead to injuries of the threshold severity levels and relating to the dangerous good carried (i.e. injuries sustained from the traffic accident alone are not considered in this instance). Crucially, the ADR regulation goes on to state:

> *"The loss of product criterion also applies if there was an imminent risk of loss of product in the above-mentioned quantities. As a rule, this has to be assumed if, owing to structural damage, the means of containment is no longer suitable for further carriage or if, for any other reason, a sufficient level of safety is no longer ensured (e.g. owing to distortion of tanks or containers, overturning of a tank or fire in the immediate vicinity)."*

This implies that any incident of overturning warrants a completion of an incident report, even more so if the tank structure has become distorted.

In total, 15 officially reported incidents from the period 2005 – 2013 have been identified as being relevant to this study, i.e. involved a flammable liquid tanker. The name of the tank manufacturer is not provided in ADR or RIDDOR incident reports. To focus the analysis on articulated vehicles, rigid tankers were excluded where the vehicle type could be identified from the reports. Generally speaking, the ADR and RIDDOR reports relate to more severe incidents than those described in the traffic logs.

Eight of the 15 accidents (53%) involved a spilt load, of which 6 cases were major spills, i.e. >1,000 litres, 1 case was a minor spill and 1 case unknown. The major spills were associated with overturning in 5 out of 6 cases and a side impact (impact with the jib of a mobile crane) in 1 out of 6 cases. 3 of the 6 cases were with aluminium tanks, the other three were of unknown material. The prevalence of aluminium tanks in the FL vehicle fleet is not known.

Seven of the 15 accidents (47%) did *not* involve spilt load, although 1 of these accidents led to minor loss of fuel from the tanker vehicle's running fuel tank, but no fire or explosion was reported. 2 out of 7 non-spill cases involved overturning. The remaining non-spill cases involved a mixture of rear, side and frontal impacts and vehicles catching fire (but without subsequent loss of product or explosion).

In total, 9 accidents involved overturning of which 7 led to load spillage (5 of which were major spills >1,000 litres, 1 minor spill, and 1 unknown). 2 of these 5 cases were with aluminium tanks (rest unknown). One case involving an overturn but not leading to spillage was with a steel tank (1 unknown).

2.2.3.3   Local news reports

A search for local news reports involving tanker incidents which occurred in the UK between 2009 and 2014 was carried out using the BBC news website. The main aim of the search was to identify the magnitude of potential underreporting of ADR incidents by comparing the results from news reports with official ADR reports provided by DfT. Initially, only reports involving the spillage of flammable liquids were recorded. However, due to the lack of results this produced, all news stories involving the words 'tanker' and 'accident' were then included in the search. This identified 59 incidents, involving a variety of vehicles (both rigid and articulated) and loads (foodstuffs, chemicals and fuel).

Table 5 provides a summary of the 59 identified UK news reports on tanker incidents.

Table 5. Summary of incidents reported by local news reports.

	Spillage	Flammable liquid	Injury	Collision	Tanker overturn	of which	Led to spillage
Number of cases	25	34	31	37	22		14
% of cases	42%	56%	53%	63%	37%		64%

It would be unrealistic to expect that all incidents involving tankers on UK roads would be identified using this approach. News reports were mainly produced for tanker incidents which were notable for a particular reason. This may be because:

- The tanker shed its load during the incident, particularly if large quantities were spilled or the load was a dangerous substance and posed a threat to the public.
- The accident caused roads to be closed and caused severe congestion or delays.
- The accident had a high severity including fatalities or injuries.

42% of the reported cases were found to be spillage incidents, with 80% of those cases involving flammable liquids. There may be a bias towards reporting events involving flammable liquids due to the more serious consequences of spilling this type of load; increased presence of emergency services and higher chance of road closures.

A tanker overturned in 37% of the news reported incidents. Of these incidents, 64% were then reported to have spilled their load. This implies that up to seven out of ten rollover events might lead to spillage, although the news reports might be biased towards more severe cases.

Casualties occurred in 53% of incidents reported in the news articles. This implies that for every 100 injury tanker accidents there could be 89 non-injury accidents. However, all values estimated using this data have a high degree of uncertainty due to the bias in reporting exceptional cases and potential exaggeration by the media.

In terms of accidents specifically involving articulated petroleum road fuel tankers, the news reports show there were a minimum of 6, possibly 7, overturning incidents over the last four complete calendar years. One involving a gas oil tanker in 2011, 2 aviation fuel, 1 diesel and 1 petrol tanker in 2012, and 1 ethanol[12] and 1 (possible) in 2013, where the exact load was not stated but one report into the incident described it as "fuel". Up to the time of drafting this report, there had been no incidents, so far, reported in 2014 and there were none in 2010.

2.2.3.4   In-depth truck accident databases

The Heavy Vehicle Crash Injury Study (HVCIS) is a collection of data coded by TRL from a sample of police fatal collision reports for collisions between 1995 and 2008 that involved at least one commercial vehicle.

The database contains 75 collisions involving articulated tankers, and a closer examination showed that 10 of these (13.3%) involved overturning. The circumstances of the overturning collisions were:

- Four rollovers occurred on bends where excessive speed was a factor.

- Three rollovers occurred prior to impact after corrective or evasive steering/braking.

- One rollover occurred turning left at a roundabout.

- Two rollovers were as a result of the tanker leaving the carriageway.

The load being carried at the time of the collision was known in nine cases (90%). Two were carrying fuel and three others were carrying loads that could be classified as dangerous goods under ADR.

- A chemical spill was recorded for 1 of 9 cases (where load was known). This suggests that chemical spills from articulated tankers are rare in fatal accidents.

- Fire was recorded for two of the ten cases. Although the cause of fire was not specified, one of the vehicles was a fuel tanker where the impact involved a large vertical drop and compartment rupture was likely. The second collision involved an overturned tanker being struck by an on-coming HGV and therefore tank rupture was also possible.

The descriptions of the remaining 65 collisions were reviewed to identify any other cases that may be of interest. There were five collisions where a tanker was impacted from the rear by another large vehicle, although in 2 cases there was another vehicle in-between the rearmost vehicle and the tanker:

- One tanker was unladen at the time of the collision.

- Of the remaining four collisions, all of them were carrying loads that may be classified as dangerous under ADR. There was one collision where a chemical spill was recorded, although this was not fuel.

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[12] An industry stakeholder confirmed that tankers used to transport ethanol were of the same basic design as those used to carry road fuels, hence its inclusion here.

- The collision resulting in chemical spillage was an impact with a closing speed of 53 mile/h, although a small car was in between the two HGVs involved in the collision which is likely to have affected the interaction between the two HGVs.

The Truck Crash Injury Study (TCIS) is a sample of collision data coded by VOSA examiners from vehicle inspections between 1995 and 2009. The database contains data relating to collisions of all severities from non-injury to fatal.

There were ten cases identified involving an articulated vehicle tanker. The accident severities were four fatal, two serious and four slight.

Of the ten TCIS tank trailers:

- Four of the collisions were solely overturning after failure to negotiate bend.

- Two collisions involved overturning after some kind of corrective or evasive steering/braking.

- Two collisions involved impacts to the rear of the tanker. One of these was hit by a light commercial vehicle. The second case involved an HGV colliding with the rear of a car and pushing it into the rear of the tanker, which is the same collision identified in the HVCIS data.

Where the loading status of the trailer was known, eight of nine trailers were laden:

- There were four hazardous loads; three flammable and one explosive.

- One chemical spill was recorded. This was the same collision involving a chemical spill as identified in the HVCIS data.

- There was one case where there was mention of damage to pipes after impact.

2.2.4   Stakeholder surveys

2.2.4.1   UK industry

Initially, key contacts in the UK fuel tanker industry, from operators and repair organisations, were identified by participants in the various work packages and the Department for Transport. An email questionnaire was distributed to all of them (to six individual companies and one trade body).

Respondents did not generally provide precise accident involvement statistics to a common definition. Rates provided varied from 1.5 – 5 per million vehicle kilometres, but these usually included mainly very minor incidents (e.g. during low speed manoeuvring or cracked wing mirrors). Where figures were provided for more serious incidents such as overturns and spillages, frequencies were generally very low; typically historical rollover frequencies of 1 in every 150 – 400 vehicles per year and major spillage resulting from 20-25% of those.

With regard to accident typology, respondents reported on the one hand rear and side impacts while being parked as well as low speed manoeuvring collisions, all of which were low severity accidents. On the other hand single vehicle roll-overs were mentioned, which were more severe and resulted in fuel spillage in 1 out of 4 cases. The amount of spilt fuel was limited; however, one stakeholder mentioned roll-overs as an accident type that was frequently associated with fuel loss. Other respondents mentioned limited fuel

spillage (up to 50 litres) from impacts affecting valves and pipework or general malfunction of valves.

The vehicle types in use varied between the respondents without a clear trend becoming obvious. Reported vehicles were articulated vehicles with three or six axles (plated at between 26 and 44 tonnes) as well as two- or three-axle rigid vehicles (from 3.5 tonnes to 26 tonnes). Respondents reported use of predominantly aluminium tanks for fuel; however, a small number of stainless steel tanks were also in use.

Where known, all the tankers were reported as being fitted with some form of vehicle stability function (e.g. ESC- Electronic Stability Control) and the general lack of rollovers was thought to be at least in part related to that fact.

Tanker operations were reported to be most often at full load (by weight not volume, 38,000 to 41,000 litres typically) or empty, though some journeys at partial loads were also made (e.g. individual compartments emptied at separate delivery locations).

In the latter stages of the project, further feedback was provided by a stakeholder on the numbers of road fuel tankers in use in the UK and on their perceived safety relative to other types of FL-registered vehicles. This feedback suggests that fuel tankers constitute about 25% to 30% of all six-axle articulated FL-registered vehicles, which would equate to around 1,000 – 1,500 vehicles on UK roads.

The stakeholders also suggested that petroleum road fuel tankers are relatively more safe than other FL-vehicles, in that they are more likely to be fitted with modern braking systems and anti-rollover technology, as well as having tank designs that provide lower centres of gravity than some other tankers (e.g. circular). A major oil company provided data, however, confirming that they had experienced one rollover of a road fuel tanker within their UK fleet in recent years, and one other case involving a vehicle working under contract to them. They also confirmed that, globally, their vehicles have been involved in at least 20 rollover incidents per annum over recent years.

2.2.4.2   International experts

In a later phase of the project, an amended version of the questionnaire was distributed among 50 international experts in tank safety via email. The scope of the questions was extended so as to acquire data on potential under-reporting of accidents. Despite a sufficient timescale and an email reminder, the number or responses was very limited.

The accident involvement rates reported per distance travelled ranged from 0.25 to 1.0 accidents per million vehicle kilometres. The lower-end of this range might be subject to under-reporting of incidents. Data from Finland suggests an accident involvement rate of 3.4 accidents per 100 registered vehicles per annum, based on reported accidents only.

No numbers were provided by stakeholders that would allow distinguishing between severe (e.g. injurious) and damage-only accidents. Data from Spain gave an indication of the frequency of spillage incidents of dangerous goods in the country. Between 2002 and 2011, on average, 36.6 cases of load spillage from dangerous goods tankers occurred annually in Spain. Figures on the relevant fleet size could not be obtained.

The accident types that were reported as being most common were front and rear impacts involving other vehicles and single vehicle rollovers. Loss of load occurred in 23% of reported accidents in Finland, i.e. in a total of 3 out of 13 cases, none of which led to fire or explosions. Two of these incidents resulted in leakage from damaged

manhole or other covers (c. 1000 litres); one minor spillage (c. 50 litres) resulted from damage during the rescue operation. Minor leakages from pumps, pipework and hoses were reported by a UK stakeholder.

No ADR or RIDDOR reports relevant for the scope of this study were submitted by stakeholders. However, it can be inferred from the replies that differing interpretations of ADR reporting requirements exist (e.g. as to whether a tanker overturn requires an ADR report in any case or not).

Under-reporting of incidents was seen as a potential problem in Finland, although no official evidence of under-reporting existed. For Spain, no under-reporting was expected by stakeholders because incident reports have to be filed by two independent entities (the operator of the tanker and traffic police), which can be used for cross-comparisons. In order to improve reporting levels, the focus was put on user-friendliness in general and aligning or combining the reports required by ADR and RIDDOR.

No evidence has been found to indicate that failures of circumferential welds have played any significant role in real-world fuel spillage incidents, although none of the non-compliant tankers are known to have been involved in such incidents.

3 WP1 – Full scale testing and associated modelling

3.1 Research methods

WP1 consisted of five tasks (in addition to cross-consortium peer review activities):

Task 1. Develop an independent non-proprietary structural hydrodynamic model of the non-compliant tankers, validate this model against the results of tanker tests, and report modelling findings.

Task 2. Design, construct and commission a rig to test tankers, including selecting and procuring suitable instrumentation for data gathering.

Task 3. Undertake tests on tankers, including preparing the tankers, assessing the tanker test method and results, and reporting the findings.

Task 4. Determine suitability of tankers for full scale tests and acquire tankers, as appropriate, in accordance with project objectives as specified by DfT.

Task 5. Capture collision and/or deformation data from relevant impacts, for example by laser scanning, to corroborate the modelling and tanker tests, and reconcile any inconsistencies.

The following section summarises the findings of the work undertaken in WP1, including in relation to its key objective, to provide load case data for WP2 to use in the Engineering Critical Assessment of a rollover impact.

For simplicity and reader convenience, Tasks 4 and 5 are combined as "the assessment and supply of tankers", which met the objectives:

- Facilitate the selection, inspection and procurement of tankers to be used by HSL and other consortium members in the delivery of the project.
- Capture data, including physical samples if needed, from damaged tankers where these data or samples may be beneficial to the project.

Tasks 2 and 3 are combined as "tanker topple tests" which met the objectives:

- Design, construct and commission a test rig for tankers which offers a reliable and repeatable method to provide experimental data for use in both improving the understanding of tanker (rollover) impact behaviour and validating HSL's Finite Element (FE) modelling.
- Prepare and test tankers to provide experimental data for use in both improving the understanding of tanker impact behaviour and validating HSL's FE modelling.

Task 1 is titled "modelling to provide load case data for rollover", with the objectives:

- Create and validate a structural hydrodynamic model of the non-compliant tankers tested under rollover conditions.
- Apply the validated finite element model to a real-world fuel load representative of real-world conditions and consider the model outputs.

The tankers considered in this research were of "banded" construction - the tanker shell was constructed in short sections, and these were joined using an extrusion band between shell sections. Two circumferential welds joined each extrusion to two shell sections. Bulkheads and baffles were also welded to the extrusion band. In this report the term "band" is used to mean the constructed extrusion band, including the circumferential welds.

3.2 Full scale testing

3.2.1 Assessment and supply of tankers

The primary criteria for tanker selection, for both the topple tests (WP1) and the fatigue data collection activities (in WP2) were that:

- tankers should be representative of non-compliant tankers in UK service; and

- the condition of the tanker's circumferential welds, based on radiography.

The circumferential welds of ten 8- and 10-banded non-compliant tankers manufactured between 2007 and 2011 were radiographed (four prior to the project). The results of the radiography both informed the selection of tankers for the full scale (topple) and road tests and provided information on the condition of the welds in a range of tankers manufactured over a five year period. Two 8-banded 6-compartment tankers, one manufactured in 2008 and the other in 2011, were selected for topple tests. One 10-banded 6-compartment tanker (2010) was selected for road tests to gather fatigue data in WP2. The radiography for the 2011 tanker showed the highest proportion of lack of fusion indications in the welds, whilst the 2010 and 2008 tankers showed the lowest.

The tankers selected for test were all fully ADR inspected and, where necessary, remedial work (other than to the circumferential welds) was conducted to ensure that the tankers satisfied the test requirements, and were roadworthy and loadworthy. In addition, the tankers selected for topple test were subject to a second radiography examination, and to internal surveys of the fillet welds. The 2011 tanker was subject to an additional internal survey of circumferential weld misalignment, and an external laser scan survey of the circumferential weld caps.

General design and construction differences between 8- and 10-banded tankers which were relevant to the research have been established. Specific design and construction differences between the 2008 and 2011 tankers were found in the extrusion profiles, the bulkhead (or baffle) welding to the extrusion bands and the fillet welds.

The basic configuration of an 8-banded, 6-compartment non-compliant tanker is given in Figure 7. Only compartment C1 contains a baffle. In a 10-banded non-compliant tanker, three compartments - C1, C2 and C4 - contain a baffle, as shown in Figure 8.
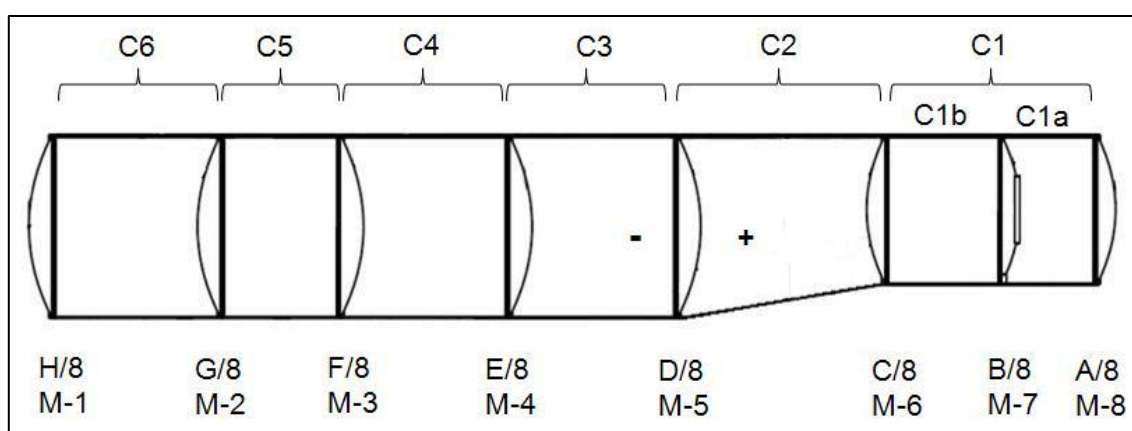

Figure 7. **8-band, 6-compartment non-compliant tank - bulkheads and baffles**
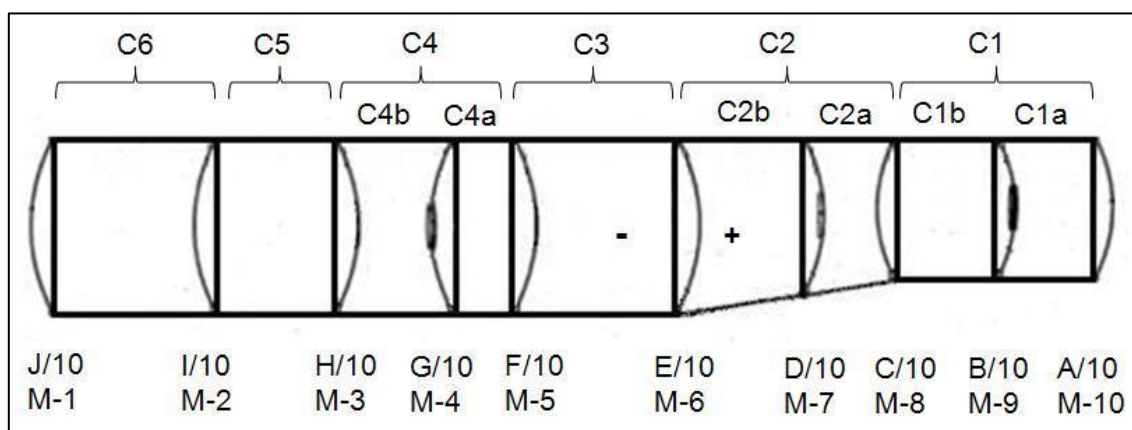
Figure 8. **10-band, 6-compartment non-compliant tank - bulkheads and baffles**

A suitable 8-compartment 40,000 litre petroleum road tanker of aluminium construction in roadworthy and loadworthy condition was sourced for a proof of concept topple test. Note that the tanker used for the proof of concept test was of stuffed construction - the tanker shell was one single construction, and the bulkheads/baffles were fitted inside and welded to the inner wall of this shell.

Two (impact) damaged non-compliant tankers, one (2010) with rear damage and another (2009) with front damage, were laser scanned to provide dimensional information on the damage, and physical samples were taken from both tankers for use in WP2.

3.2.2   Tanker topple tests - procedures

3.2.2.1   Topple test approach

After considering various approaches, and discussing the proposed test method with the research consortium, HSL developed a topple test to roll over a suitably instrumented tanker in a controlled and repeatable way. This was a uniform longitudinal dynamic impact of the tanker side with the surface of a prepared test pad, resulting from the lateral rotation of the tanker around the axis formed by the outer edge of the tanker's road wheels. The tanker was placed on a ramp and was tilted quasi-statically to the point where its centre of gravity was above the axis of rotation, as illustrated in Figure 9. Once in this position, the stability of the tanker was sufficiently compromised such that only a small additional impetus was required to induce the tanker to topple. The tanker was filled with water to represent the fuel load: petroleum, diesel or fuel oil was not practical for environmental and safety reasons (as well as cost).

Information on the tanker's dimensions, geometry and centre of gravity was used to calculate the approximate angle at which the tanker would become unstable. The ramps were manufactured to provide an initial tilt angle several degrees less than the angle required for topple, to allow an appropriate safety margin. These ramps were secured to a concrete test pad and a plate steel landing pad was used to provide a more robust and repeatable impact area (than the concrete).
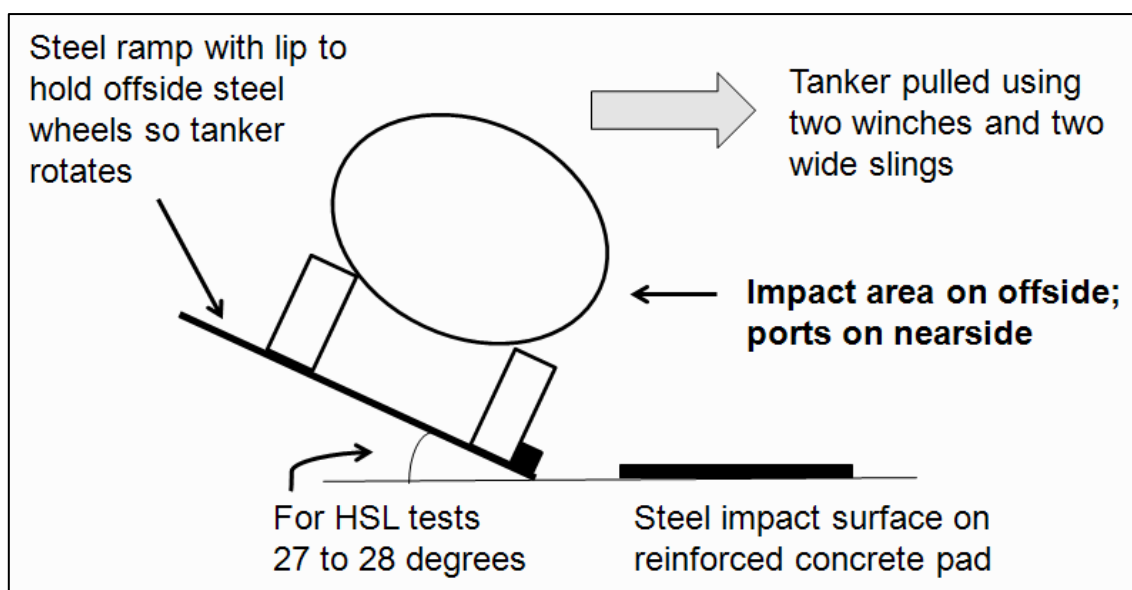
Figure 9. The key features of the HSL tanker topple test

Impact was on the offside of the tanker to avoid damage to the filling ports on the nearside of the tanker. Once in position on the ramps, and prepared for test, the tanker was filled with the required volume of water evenly distributed across all the individual compartments. The tanker was then toppled sideways, pivoting around the outer edge of its offside wheels. To eliminate the risk of the offside tyres coming off the wheel rims during the test, and to avoid variability from uncontrolled shear movement in these tyres during the topple, the offside wheels were replaced with dimensionally similar rectangular steel supports ('steel wheels').

The tanker was not tested with a tractor unit to avoid uncontrolled variations between tests caused by tractor unit rotation and possible failure of the kingpin due to unconventional loading. Instead, a steel frame, known as the 5th wheel assembly, was fitted to the tanker at the front (kingpin plate) to give the support normally provided by the tractor and to keep the tanker at the desired coupling height for the test. The tanker's suspension was blocked and held rigid to remove sources of uncontrolled variation, such as changes in the ride height, and to keep the tank position fixed relative to the suspension during the topple.

The tanker was rotated into the topple position using two parallel winching systems (the winches were two horizontally-positioned chain hoists) with wide slings to spread the load and prevent high stress levels on the tanker body and comb when the winch forces were applied to the slings.

Rotating the tanker into the topple position was controlled by ensuring the load on each winch line was similar, giving an 'even pull'. A 'Winch Master' controlled the operation by monitoring the load on each line and giving orders to the 'Winch Operators' to ensure that the loads on each line remained similar. As the winches began to take the weight of the tanker, the tanker rotated and pivoted around the offside steel wheels and the offside of the 5th wheel assembly. When the point of instability was reached, the winching lines slackened and the tanker toppled onto its side under the influence of gravity alone.

3.2.2.2   Tankers tested

Three tankers were tested. First, a proof of concept test was conducted on a 'guinea pig' aluminium petroleum road tanker, which met the needs for the proof of concept test rather than the full tanker specification. The aim of this test was to establish that the basic test method and data logging system were sound, so minimal test instrumentation was used on the tanker, with the full data logging system operating. All the key features of the preparation and topple test, including tanker recovery, were conducted, so that improvements or modifications to the test method could be considered.

Second and third, respectively, non-compliant tankers 8-banded 6-compartment, 2008 and 8-banded 6-compartment, 2011 were tested with full test instrumentation. The configuration of these tankers is given in Figure 7, together with the compartment and band numbering convention used.

3.2.2.3   Tanker instrumentation

The full data gathering instrumentation comprised of strain gauges (24), pressure transducers (14) and accelerometers (6), i.e. 44 channels in total. These provided data for finite element model validation and characterising the general impact behaviour.

Cables from instrumentation located inside a compartment passed out of the compartment through a specially designed baffle. This cabling then led to junction boxes which linked the tanker instrumentation to the main wiring loom connected to the data loggers. Cables from instrumentation on the outside of the tanker were also brought to these junction boxes.

Two independent data loggers were used, with each logger specific to one of compartments C1b and C4. During the tests, these loggers were synchronised with the high speed video and set to acquire data at 50,000 samples per second (50 ks/s), or one recording every 0.02 milliseconds.

3.2.2.4   Pre-test inspection of tankers

The fully-tested tankers were laser scanned 'as received', after lifting onto the ramps, after topple (on their sides) and after recovery (upright, on their wheels). This provided general dimensional data and allowed confirmation of:

- any changes caused by HSL preparation of the tanker; and
- any changes to the tanker dimensions and shape after the impact.

The internal circumferential welds in the 2008 tanker were visually inspected during preparation, and the locations of fillet welds between the extrusion band and the shell were noted. A fuller survey of the internal circumferential welds in the 2011 tanker was made, including the locations of misalignments, fillet welds between the extrusion band and the shell and other features. The external weld caps on the circumferential welds in this tanker were surveyed by laser scan to provide dimensional data on cap height and width, cap spacing and misalignment for WP2.

Once preparation of the tankers, including instrumentation, was complete, the manway lids were refitted and a pneumatic pressure test was conducted to confirm that the tankers remained fully sealed and loadworthy, with full internal and external integrity.

3.2.2.5   Tanker filling

Immediately before test, the tankers were filled with water using a calibrated water meter. The proof of concept tanker was filled to about 90% by volume, the nearest to the volume that would be used for a fuel load (95% full, 5% ullage) that could be achieved when the tanker was inclined on the ramps. This gave a mass of 37,990 kg of water. Although this was above the tanker's rated maximum load, it provided a suitably severe test of the mechanical integrity of the test system.

For the non-compliant test tankers, it was agreed by the research consortium to fill the tankers to their maximum rated load mass (31,380 kg), which was below their rated volume for fuel. Both tankers were filled with 31,376 litres of water, with each compartment filled to about 70% of its maximum capacity.

3.2.2.6   Photography and video

The full tests were recorded using thirteen video cameras ranging from standard speed (25 frames per second) to high speed (1000 frames per second), plus hi-quality stills photos and time-lapse of the preparation at the pad, tests and after-test at the pad activity. Frames from the high speed video were analysed to obtain accurate measurements of acceleration and impact velocity at the front and rear of the tanker.

3.2.2.7   After-test activity

Immediately after impact, visual examination was used to identify leaks and other impact features. The tanker was then emptied and lifted back upright (on its wheels), and a further visual examination made. For the non-compliant test tankers, a pressure test was conducted to ascertain the internal integrity of the compartments and bulkheads. Physical samples were taken from both tankers for use in WP2.

3.2.3   Tanker topple tests - results

3.2.3.1   Test rig and method

The proof of concept test was successful, with no major problems found in the topple test rig or test method. Some minor improvements in the exact sequence and detail of the test were noted and implemented during the full tanker tests. These tests were also successful, with no problems from the test rig or test method.

3.2.3.2   Instrumentation and data gathering

There was good agreement between the accelerometer data and acceleration values obtained by analysing the high speed video.

All 44 channels of instrumentation provided valid data for the 2008 tanker. For the 2011 tanker, all but three channels of instrumentation provided valid data. Signals from three of the internal strain gauges were lost when the gauges came into contact with the water during filling. Although the signals from these gauges re-appeared during the impact, with their trend after this being similar to comparable gauges, data from these three gauges could not reliably be used.

3.2.3.3   Impact behaviour

The overall impact duration was a few seconds for all the tests, with most deformation occurring in the first 100 ms. Using the high speed video:

- The proof of concept tanker was found to have impacted reasonably uniformly along its length, with front and rear hitting the ground within a few milliseconds of each other. The impact speed at the rear of the tanker was 4.25 m/s (around 2 rad/s).

- The 2008 tanker impacted with speeds of 4.50 m/s (1.82 rad/s) at the front and 4.10 m/s (1.86 rad/s) at the rear of the tanker, and the rear hitting the ground first, less than 1 ms before front of the tanker.

- The 2011 tanker impacted with speeds of 4.55 m/s (1.84 rad/s) at the front and 4.25 m/s (1.93 rad/s) at the rear of the tanker, and the rear hitting the ground first, less than 7 ms before front of the tanker.

These impact velocities lie within the range of those for mild rollovers in real accidents, where impact velocities of between 100 to 150 degrees/s, which correspond to 1.75 to 2.62 rad/s, have been reported in published research literature (as reported by WP3).

3.2.3.4   Impact and deformation data

The pressure data in both compartments were similar for both non-compliant tankers (Figure 10). Short duration pressure peaks between 2 and 7.7 bar (28.4 and 110 psi) were observed during the first 20 to 30 ms of the impact; these were above the 2-bar (28.4 psi) used in previous (pressure-impulse) rollover modelling by Bysh and Dorn, 1996. However, between around 20 and 40 ms after impact the pressures were around 2-bar, and after this the pressures reduced further.

The strain data in both compartments were similar for both tankers. Strains at the centres of the compartments were reasonably consistent between the two tankers, with more variation in the hoop strains than in the longitudinal strains. During impact, for both tankers, high speed video captured free travelling flexural waves propagating away from the impact line around the circumference of the tanker. Such waves should result in more pronounced ripples in the circumferential strain than the longitudinal strain at the centre of the compartment and, for both tankers, this was found to be the case. Strains near the circumferential welds were higher than those at the compartment centre, with some yielding and plastic deformation observed in the strain behaviour near the welds.

After the test, the offside (impact side) of both the tankers had a similar deformation shape with the impact area flattened - Figure 11 illustrates the damage to one of the non-compliant tankers. The deformation profile was similar along the length of the tankers, with the level of deformation increasing from front to rear. The deformation data, both as a reduction in tanker diameter and as the length of the flattened impact chord, were similar for both tankers. Comparison of laser scan images taken before and after the tests showed that the impact had caused permanent reduction in tanker diameter of approximately 100 mm at the rear and 82 mm at the front of the 2008 tanker; and of approximately 107 mm at the rear and 82 mm at the front of the 2011 tanker.
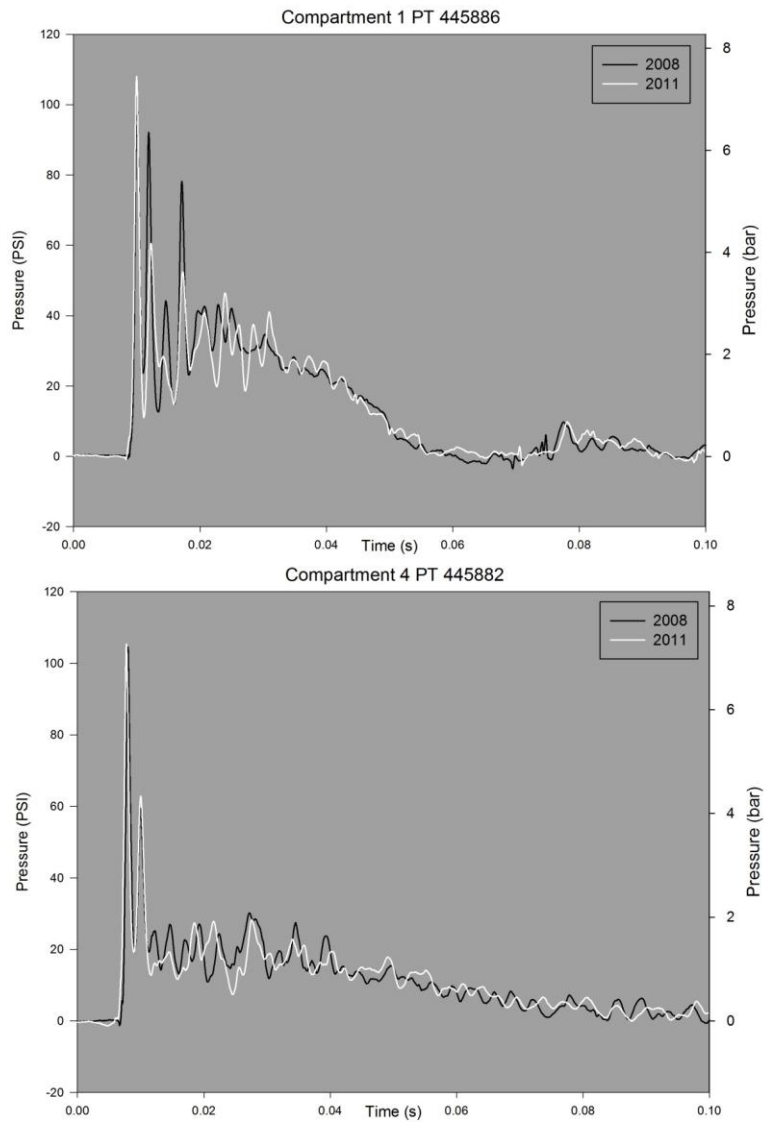
Figure 10. Comparison - pressure measured at the centre of the impact area.

Figure 11. Impact damage to a non-compliant tanker, viewed from front

3.2.3.5 Impact damage

(2008 tanker). During impact, a small amount of water was lost through the pressure relief valves. Immediately after the test, the only visible leak from the tanker was between the rear bulkhead and extrusion band at the top of the impact area. Subsequent visual inspection found a rupture within the weld between the rear bulkhead and extrusion band at the top of the impact area (Figure 12). Folding of the rear bulkhead over the weld along most of the impact area prevented assessment of the extent of this rupture across the impact area. However, there was no visible damage at the bottom of the impact area where the rear bulkhead was not folded over the weld. Before the water was pumped out of each compartment, no obvious lowering of the water level in the adjacent compartments was observed; so it was unclear if there had been any breaches between compartments. Once the tanker had been lifted back onto its wheels, pneumatic pressure tests found that all compartments had lost their internal integrity. On this tanker the bulkheads were welded to the extrusion bands on one side, the convex side of the bulkhead curvature, and not to both sides of the extruded band. The convex side of the rear bulkhead was the outside.

Figure 12. The 2008 tanker – rupture in the weld at top of impact zone

During post-mortem examination, an apparent through-wall crack along the circumferential weld at the top of the impact zone was observed. This apparent crack can be seen in Figure 13, which is taken from close examination of a photograph of the tanker after being lifted back onto its wheels.
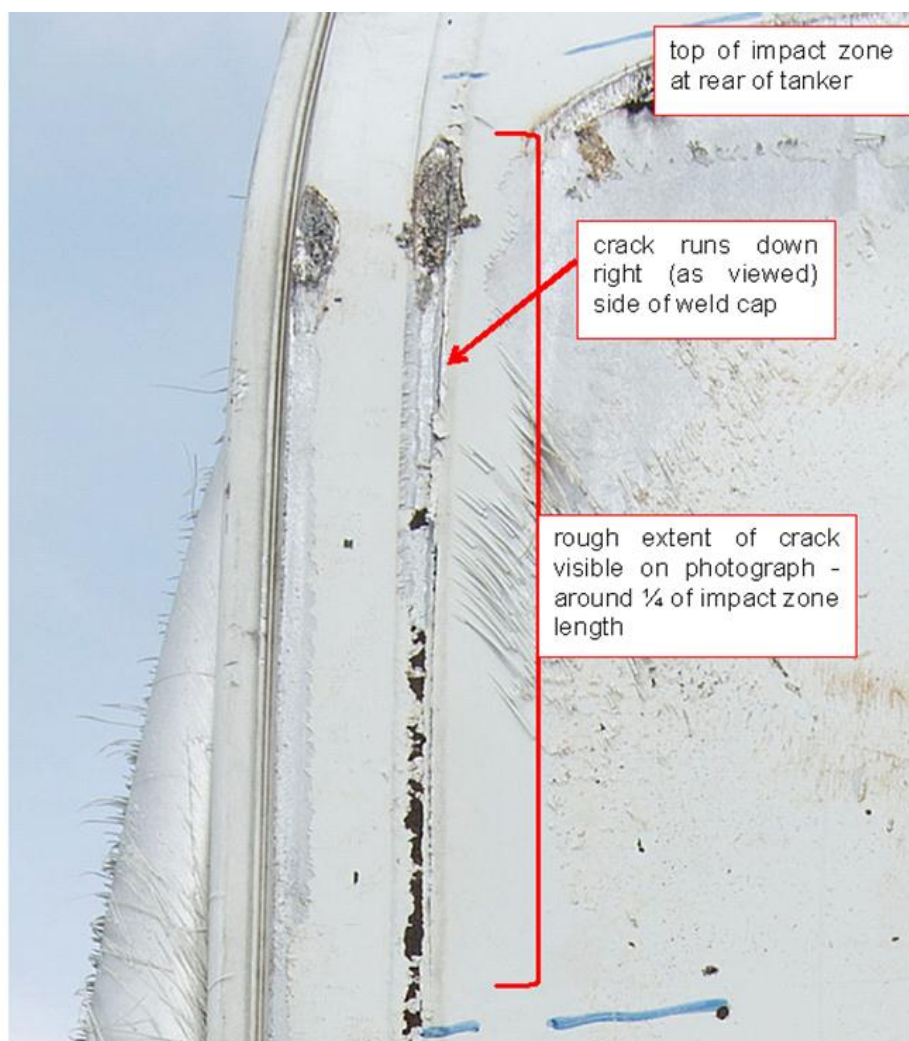
Figure 13. The 2008 tanker – apparent through-wall crack along the circumferential weld at the top of the impact zone at the rear

(2011 tanker). During impact, a similar amount of water was lost through the pressure relief valves as for the 2008 tanker. Immediately after the test, the only visible leak from the tanker was between the front bulkhead and the extrusion band at the top of the impact area. Subsequent visual inspection found a rupture in the toe of the weld between the front bulkhead and the extrusion band at the top of the impact area (Figure 14, left). Folding of the front bulkhead over the weld along most of the impact area prevented assessment of the extent of this rupture across the impact area. However, there was also a crack in the toe of the weld at the bottom of the impact area where the front bulkhead was not folded over the weld (Figure 14, right).

Before the water was pumped out of each compartment, compartment 1 had emptied through the leak at the front bulkhead, and compartment 2 had started to empty. This suggested a leak at the bulkhead between compartments 1 and 2. Similarly, when compartment 4 was emptied, the water level reduced in compartment 5, suggesting a leak at the bulkhead between compartments 4 and 5. Once the tanker had been lifted

back onto its wheels, pneumatic pressure tests confirmed that internal integrity had been lost between compartments 1 and 2, and between compartments 4 and 5, while the other bulkheads and compartments had maintained their internal integrity. On this tanker the bulkheads were welded to the extrusion bands on both sides of the bulkhead.

Physical samples covering the following areas were taken for use in WP2:

- the rear bulkhead, the rear extrusion band (H) and the rear end of the shell for the offside and nearside of the 2008 tanker; and

- the front bulkhead, the front extrusion band (A) and the front end of the shell for the offside of the 2011 tanker.

Figure 14. The 2011 tanker – rupture at toe of weld at top of impact zone (left) and crack at toe of weld at bottom of impact zone (right).

In the context of inspecting the damage sustained by the tankers used in the testing, the manufacturer of the non-compliant tankers indicated that the damage around the joints between the extrusion band and the bulkhead/baffles for both tankers is consistent with real-world rollovers.

3.2.3.6   Test uniformity and use of test data for HSL's finite element model

Overall, the test method met the objective of providing a reliable and repeatable method very successfully. The consistent impact behaviour and instrumentation data for the non-compliant tankers tested was evidence of the test method's repeatability.

The following section describes how the test data from these tankers was used to refine and validate HSL's finite element model of tanker rollover.

3.3 Modelling to provide load case data for rollover

A suitable initial finite element (FE) model of the tankers, based on HSL's topple test, was created. This model has been refined and validated against experimental data from the topple tests, and then used to simulate real-world fuel loads. The modelling software used was ANSYS Autodyn version 15.

3.3.1 Initial development of the tanker rollover model

The mass of a fuel tanker with a full load consists of the load itself (approximately 30 tonnes of fuel) and the tank (approximately 5 tonnes). Therefore, an appropriate representation of the fuel is necessary to accurately model the event. After consideration of alternative approaches, the Euler/Lagrange fluid structure interaction approach was chosen for the analysis of the tanker topple event. This approach allows the detailed geometry of the tanker to be represented using shell elements and the liquid in the tanker to be modelled.

The impact of the tanker with the ground is a highly dynamic event, with an impact velocity around 4 m/s, and is likely to result in large deformations and high levels of strain. In terms of dynamic analysis, this is relatively slow (compared to ballistic events, for example) but it is still fast enough to be suitable for an explicit analysis. The duration of the main impact which causes the majority of the deformation and stress in the tanker was approximately 100 ms.

The empty space in the tanker's compartments was modelled as a void, as opposed to assuming air or air/fuel vapour, as this approach is much more efficient in terms of solution time. It also prevents the build-up of pressures in the compartment due to the reduction in volume caused by crushing, as in reality this build-up would be prevented by the tanker's pressure relief valves.

As this model does not consider the detailed behaviour of the welds at the extrusion bands, a mesh size of between 10 mm and 20 mm was found to be appropriate for the sections of the tanker subject to the largest deformations, and very little difference in deformation values was observed with further refinement. However, when model data was compared to topple test data the mesh size was refined in some key locations.

The 2008 and 2011 tankers used different extrusion designs in the construction of the bands which join the sections of the tanker together, so geometries for both designs were created for the model of the extrusion band. These tankers also included fillet welds in different positions on their circumferential welds. Geometries for the extrusion band with and without fillet welds were created for use where appropriate.

Appropriate material properties were used in the finite element model. In particular, the properties for aluminium were based on a series of test results on plate and weld metal from another non-compliant tanker. As the vast majority of the tanker consists of parent plate material and the welds are not explicitly represented in the finite element model, only the parent metal test results were considered.

The techniques of mass scaling (adding mass to some small elements to increase the solution speed) and Euler subcycling (solving the fluid regions of the model less frequently than the solid parts) were found to offer large benefits in terms of faster solution times without significantly affecting the results obtained.

3.3.2 Refinement and validation of the tanker rollover model

3.3.2.1 Features of the refined finite element models

Finite element models of both the tested tankers were created, and the effect of differences between the tankers on model outputs considered. The finite element model included representation of the main shell, the extruded bands, the bulkheads and baffle, the comb along the top, and basic representations of the support structures at the front (fifth wheel location), the landing gear support and the rear. Simplified representations of the suspension, axles, steel wheels and manway covers were used. For simplicity, the finite element model used steel wheels on both sides of the tanker. Details omitted included smaller holes in the baffles, the sumps and pump, the guttering and vapour recovery tubes through the compartments, and any other small, non-structural attachments. The basic features of the finite element model are given in Figure 15.
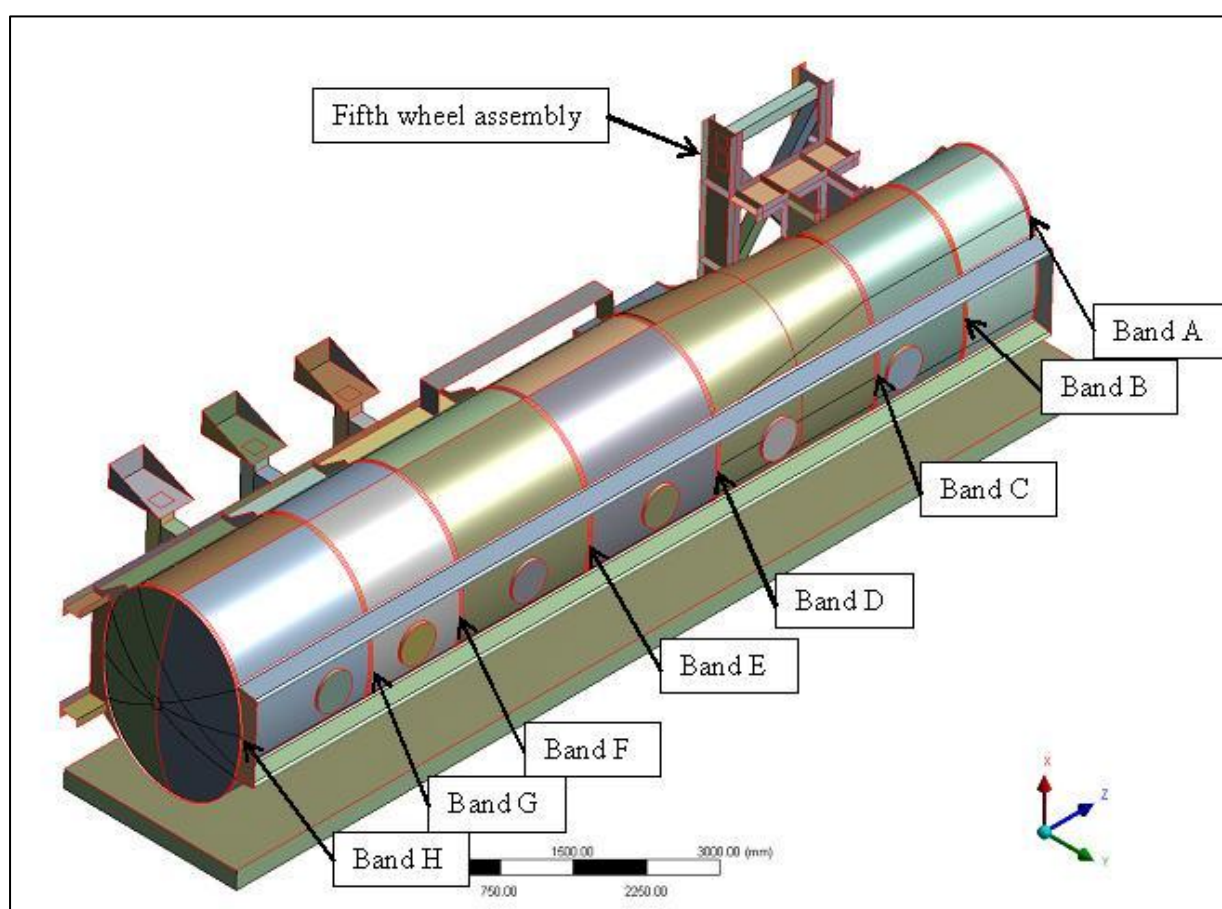
Figure 15. Overview of tanker finite element model

The key differences between the two tankers affecting the finite element model were:

- different fillet weld locations; and
- different extrusion profiles and extrusion band welding.

Figure 16 illustrates the fillet welds in the 2011 tanker. A long fillet weld runs to the bottom left corner - long welds were also used on the 2008 tanker. A short (or 'stitch') fillet weld is above the strain gauge location point - these short welds were used extensively on the 2011 tanker but infrequently on the 2008 tanker.

Strain gauge location (before attachment)
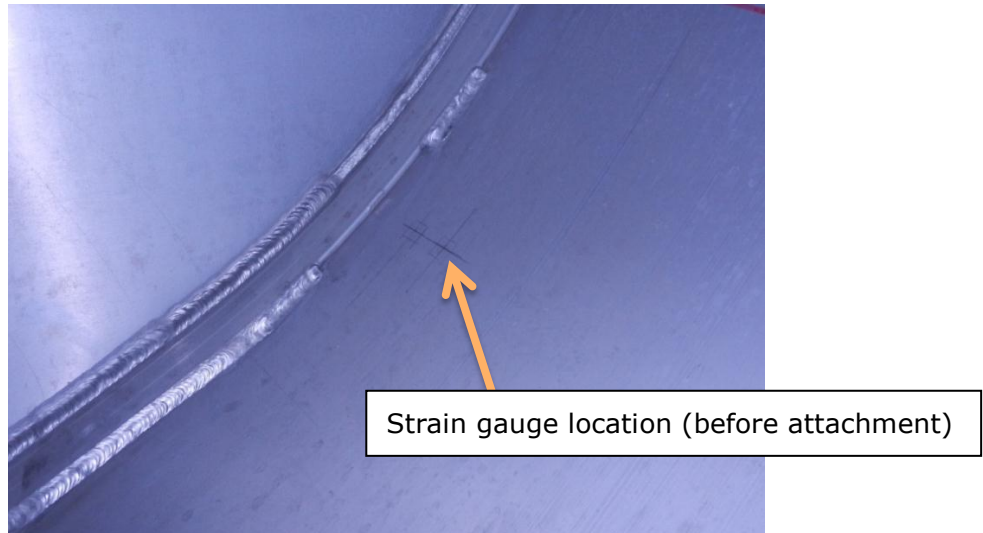
Figure 16. Example of fillet welds, 2011 tanker

Figure 17 gives the extrusion profile for the 2011 tanker, and Figure 18 gives the extrusion profile for the 2008 tanker (all dimensions are approximate). In the 2008 case, the bulkhead was only welded to the convex side of the bulkhead (or baffle), whereas in the 2011 tanker, the bulkhead was welded to both sides. In addition, the extrusion was a different shape between the two tankers. If a fillet weld was present (not shown in the Figures) it would have been at the toe of the weld between the tanker shell and the extrusion.

Block

Bulkhead

Weld location

Figure 17. Band extrusion profile for the 2008 tanker

Figure 18. Band extrusion profile for the 2011 tanker

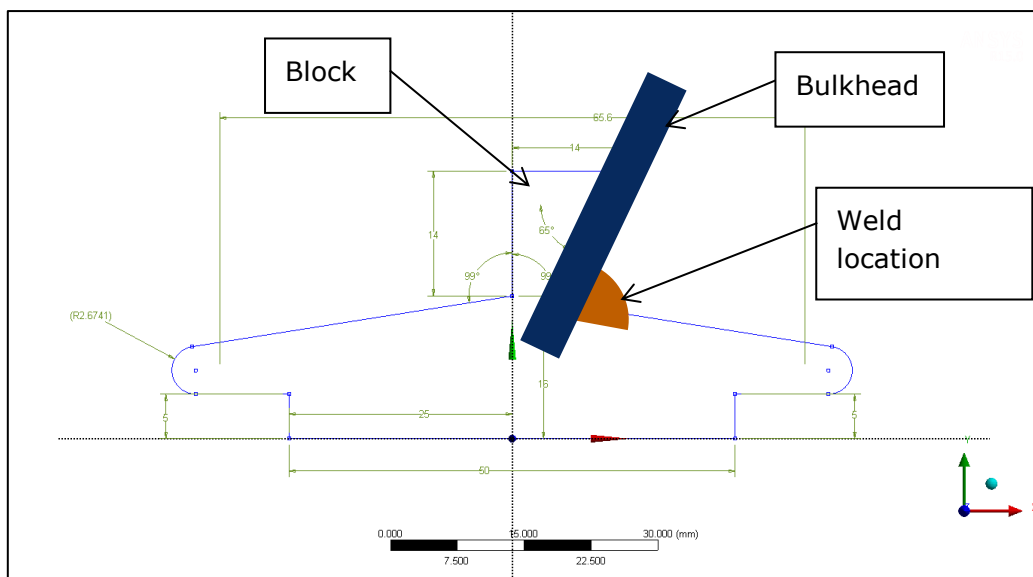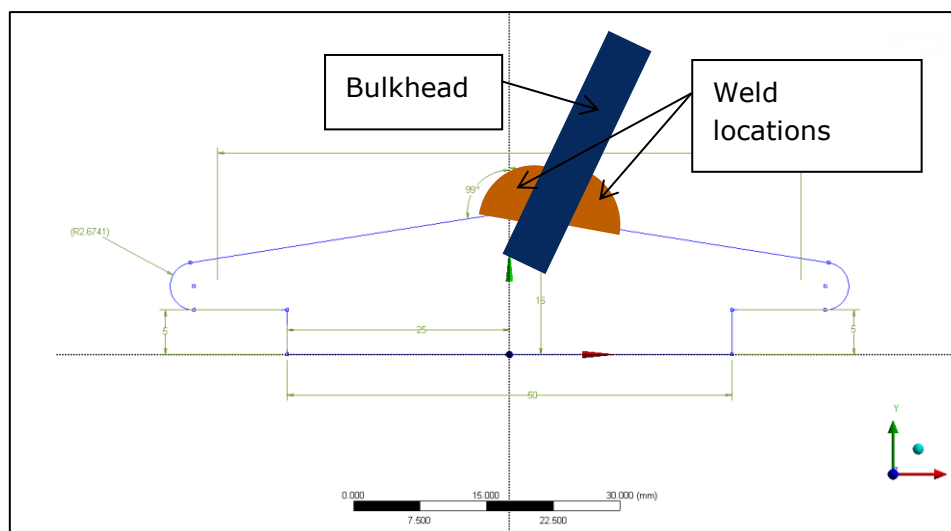There were only small differences between the finite element model results from the two tankers. This was a similar outcome to the topple test results.

Modelled variations in impact velocity did not have a major influence on the results from the finite element analysis. This was shown both by the comparison between models using water with impact velocities of 1.89 rad/s and 2.0 rad/s, and the comparison between models using fuel oil with impact velocities of 2.0 rad/s and 2.6 rad/s. In general, the deformations and significant bending moments and membrane stresses were seen to vary by up to 10% for the cases modelled.

The orientation of the bulkheads was found to have a large effect on the bending moments in the tanker shell near to the extrusion bands; in simple terms, they were higher on the convex side of the bulkheads. In more detail, the bending moments were higher in the positive direction (putting the inner surface of the tanker shell in tension) on the convex side of the bulkheads. This was probably due to the buckling of the bulkheads during impact resulting in a slight twisting of the extruded band.

Although the resolution of the finite element model was not sufficient to consider the extrusion bands and welds in detail (this detail was considered in WP2), fillet welds were found to affect behaviour near to the extrusion bands. Results from the models suggest that internal fillet welds between the extrusion band and the shell reduce the bending moment in the shell next to the bands. It would appear that the bending moments at the shell/band interface are reduced, and not just moved to the fillet weld location. More detailed examination, with more detailed modelling of the fillet weld, would be needed to confirm the extent of the benefits of the fillet weld and the effect of intermittent ('stitched'), rather than continuous, fillet welds.

3.3.2.2   Finite element model validation

As there was little difference between the 2008 and 2011 tanker finite element models, outputs from the 2011 model were compared to topple test data for that tanker. Good agreement was obtained between the modified finite element model results for deformation and the topple test results, based on laser scan data (Figure 19). All the flat lengths measured were within 15%, with less than 5% difference at most locations.
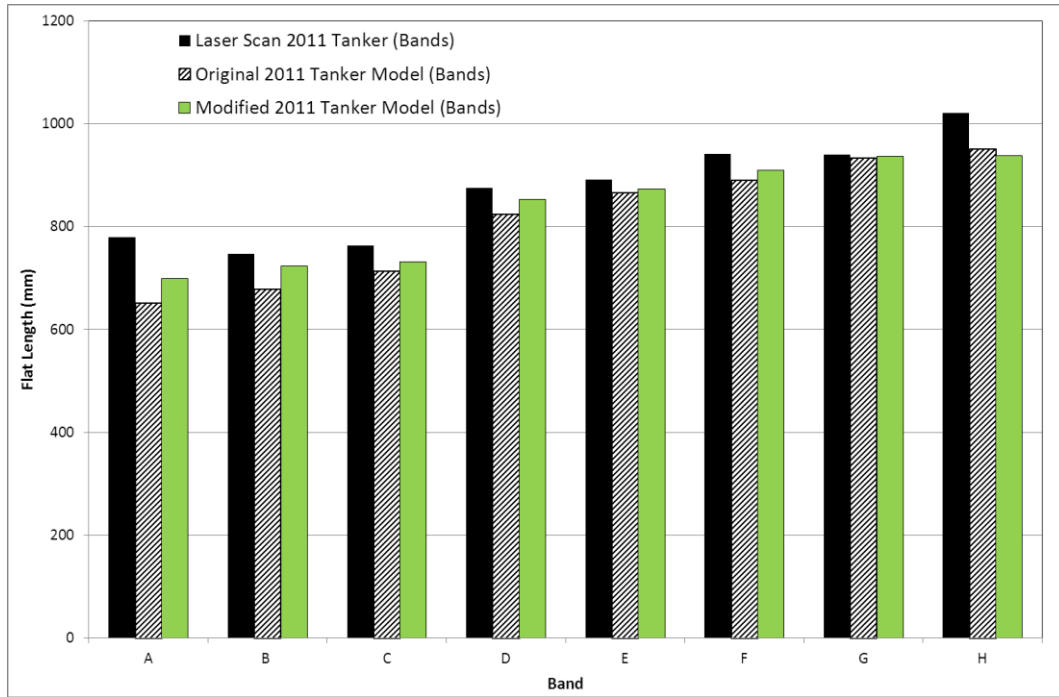
Figure 19. Comparison of flat lengths at band locations for the 2011 tanker

The bending moments near to the extrusion bands (and thus of greatest interest to the research) showed generally good agreement between the test values and the finite element values. An example is given in Figure 20. The largest difference between test and finite element model results for bending moments at these locations was 22%, with most other results within a few percent. The membrane stresses showed a larger variation between test and finite element model, but the membrane stresses were generally much lower than the bending stresses at these locations, and therefore were not as important.

Figure 20. Comparison of bending moments (per unit length) near band B for the 2011 tanker

The finite element models did not predict the bending stresses in the central regions of the compartments well. In particular, bending moments in the longitudinal direction were significantly overestimated by the finite element models, although at this location, the bending moments were very small.

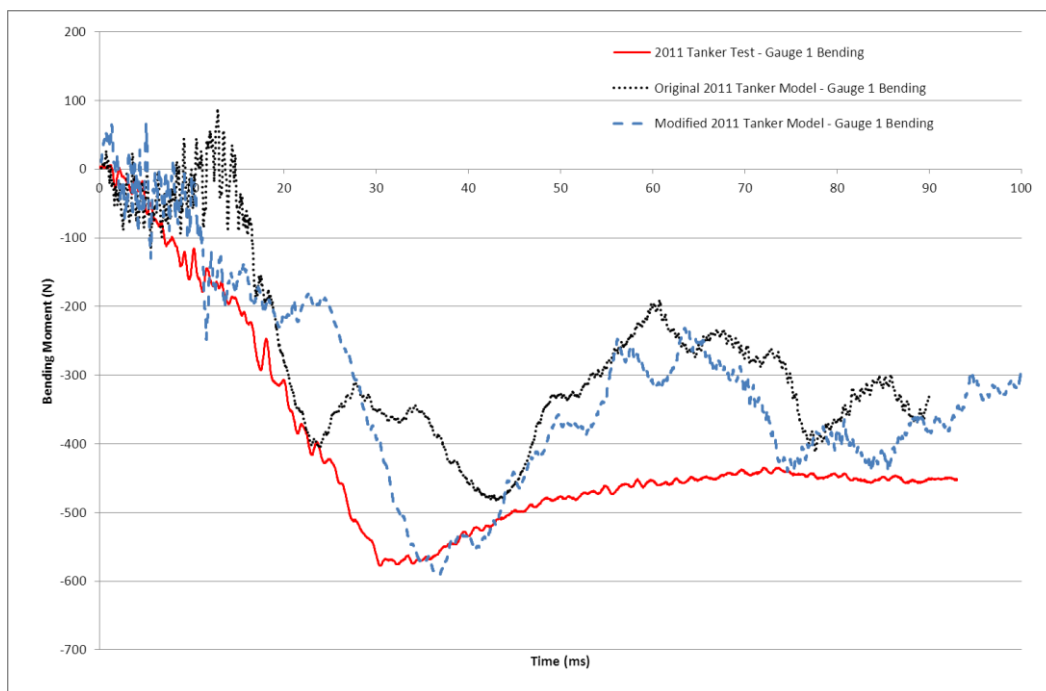Good agreement was found between the pressures recorded during the test and those generated by the model for the overall trends during the impact event (Figure 21). The magnitude of the initial pressure spikes was found, in the finite element model, to reduce rapidly as the measurement distance from the tanker wall increased. Differences in the magnitude of the initial pressure spikes between the finite element model and test values can be attributed to difficulties in correlating the location of the finite element gauge points to the physical locations of the pressure transducers used during the tests.
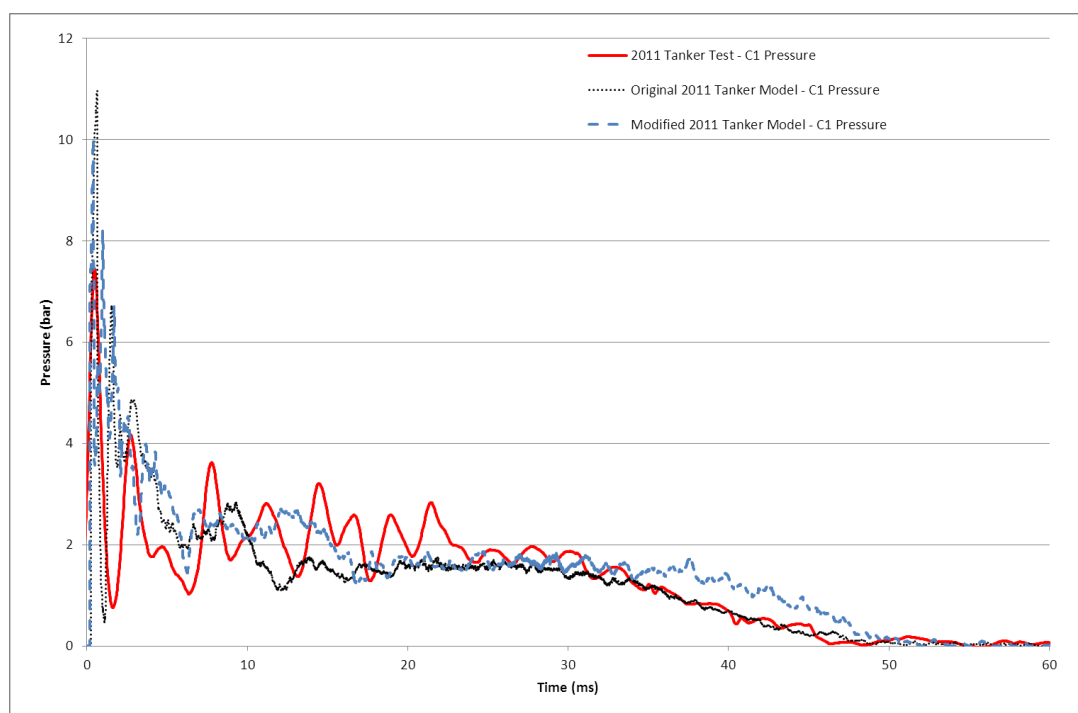
Figure 21. Comparison of pressures near the impact location, compartment C1b

At locations on the convex side of a bulkhead where no fillet weld had been modelled, the (normalised per unit length) bending moments approaching the band were typically in the order of 1,000 N. Where fillet welds had been modelled, these bending moments were significantly lower, with values in the order of 600 N. The addition of the fillet weld would thus appear to reduce the peak bending moment, rather than simply moving the peak to outside the fillet weld location.

The highest levels of plastic strain were observed in the bulkheads, at the top and bottom of the flat chord generated by the impact (Figure 22). The magnitude of the peak plastic strains was in the order of 0.2 (or 20%), a level at which failure may be expected. It was at the top of this flat impact chord where ruptures in the toe of the weld and within the weld between the bulkhead and extrusion band occurred during topple tests on the 2008 and 2011 tankers, respectively.
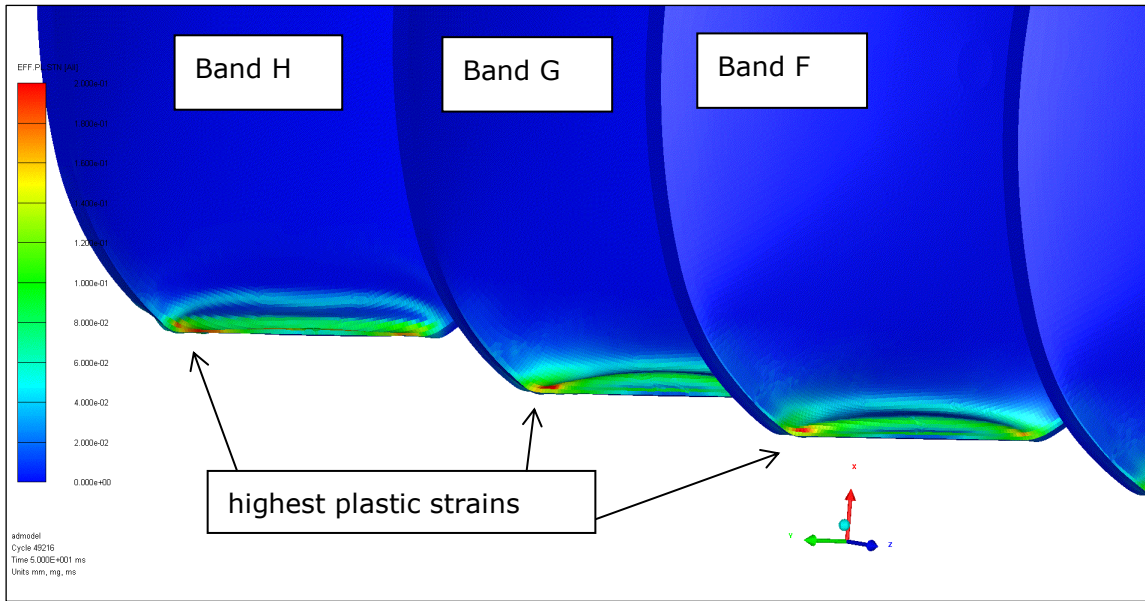
Figure 22. Plastic strain in the bulkheads from the 2011 tanker modified model

Overall, HSL's finite element model for the 2011 tanker agrees well with the topple test data for the 2011 tanker, providing good validation for the model.

3.3.3 Modelling real-world fuel loads

The tanker was modelled with representative loads of fuel oil and petrol, as opposed to water. In the fuel oil scenario, the tanker was filled to the stated capacity for each compartment except one (the third, of six, from the front and the first compartment after the tanker's expanding conical section), which was empty (Figure 23, left). The empty compartment prevented the tanker being overloaded by mass due to fuel oil's higher density than petrol, and was modelled in accordance with standard operator practice. The model of a representative petrol load used a tanker with each compartment filled to its stated capacity (Figure 23, right).
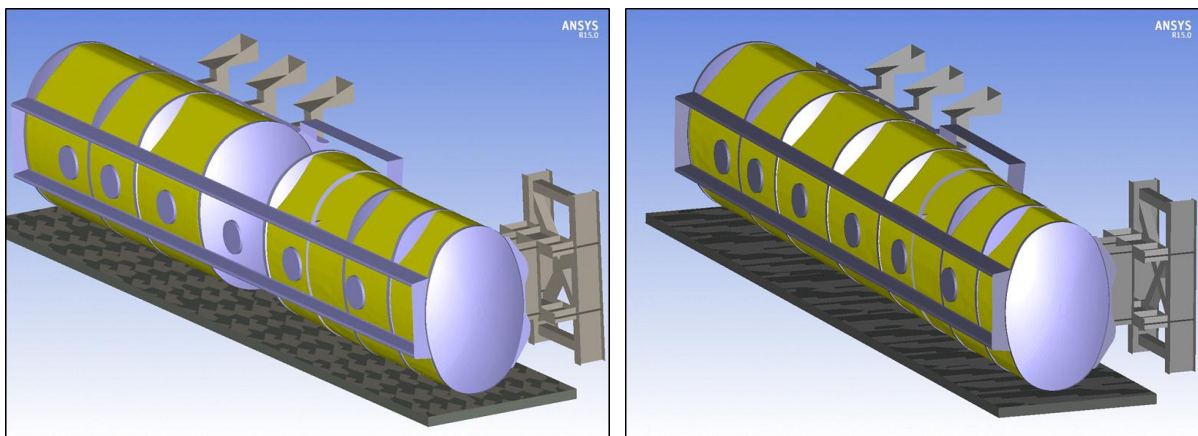
Figure 23. Fuel load models showing initial location of fuel – fuel oil (left) and petrol (right)

The modelled deflection at the front band shows little variation between water and fuel oil, but is greater for petrol. While behaviour at the rear of the tanker is influenced locally for fuel oil by the empty compartment, deflection was significantly higher at the rear of the tanker for fuel oil than for water, with deflection for petrol slightly higher than fuel oil. The levels of plastic strain in the bulkheads for different load liquids increase with increasing levels of deflection (from 20% for water, through 25% for fuel oil, to 34% for petrol, for example in terms of the through-thickness average maximum equivalent plastic strains).

The effect of the different liquid loads on the bending moments would appear to be highly dependent on locations. At the front of the tanker, there was little difference in the bending moments for the different liquids. Near the rear of the tanker, the bending moments due to the fuel oil and petrol were higher than for the water. This corresponds to the higher deflections observed at the rear of the tanker for fuel oil and petrol.

The bending moments near to band E/8 show large differences between the models with water and petrol and the model with fuel oil. In the fuel oil model, compartment 3 was left empty, and band E/8 was the bulkhead separating compartments 3 and 4. Therefore, significant differences would be expected as the pressure due to the fuel oil was only acting on one side of the bulkhead.

The empty compartment (in the fuel oil model) resulted in differences in the stresses in the tanker shell close to the bulkheads separating the filled and empty compartments. The deformation of the bulkhead on the front side of the empty compartment was significantly different in this model to that for water, as the pressure was only applied by fuel oil to the convex side of this bulkhead rather, than to both sides by water. In contrast, deformation of the bulkhead on the rear side of the empty compartment, where pressure was only applied by fuel oil to the concave side of the bulkhead, rather than to both sides by water, was not significantly different in this model to that for water. This is illustrated in Figure 24. This type of deformation was not observed for the petrol model where no compartment was left empty.

While there may be significant local effects for any load liquid if one compartment is empty, the results from the fuel oil modelling may suggest that the topple test conditions, with an equivalent mass of water to a full fuel load distributed evenly throughout the tanker, may not be as severe as some real-life events.

3.3.3.1   Fuel oil model data for WP2

The bending moments were highest at the rear of the tanker. The patterns and values of bending moments were very similar for the fuel oil and petrol models, with the exception of the area adjacent to band E/8, when the compartment was empty for the fuel oil model. There were areas of high bending moment within the impact zone, and also outside the impact zone close to the bands. The high bending moments here lay between the end of the impact zone and the comb, with the highest moments moving up the tanker surface as the size of the impact zone flat increases. The bending moments near to band F/8 were very similar between the fuel oil and petrol models.
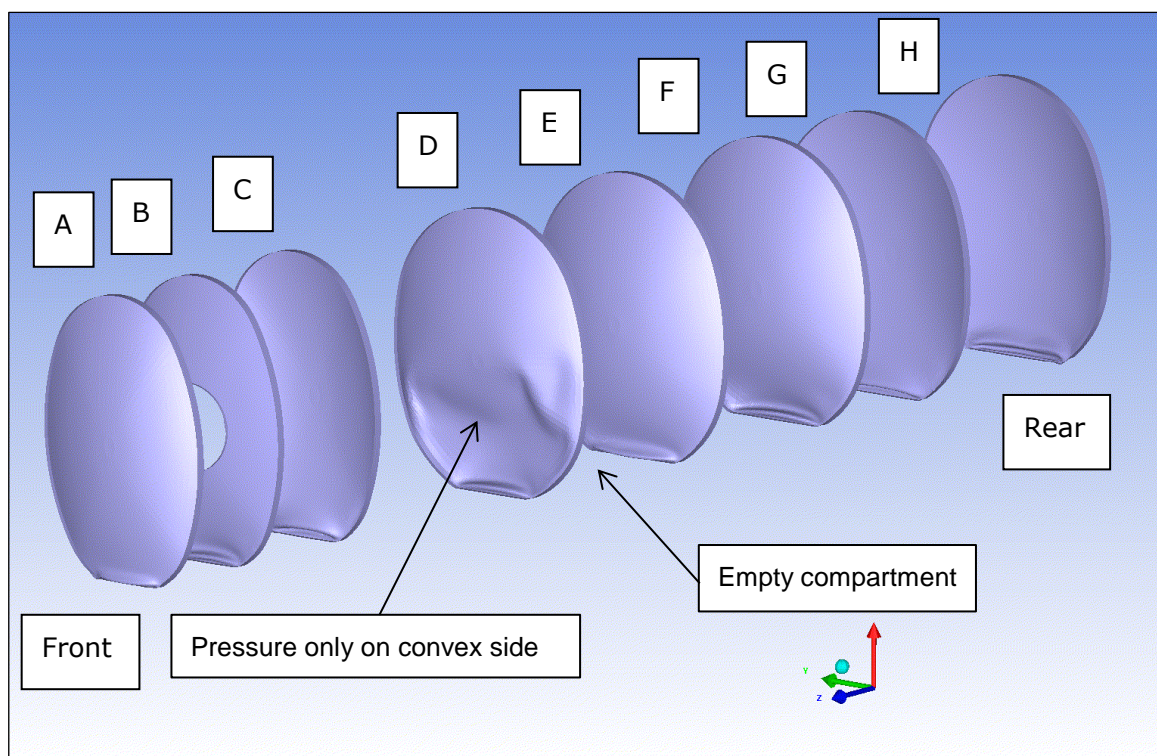
Figure 24. Deformed bulkheads for fuel oil model with impact at 2.6 rad/s clearly showing effect of pressure only on the convex side of bulkhead D

In these areas, on the convex side of the bulkheads, the bending moment increased rapidly as proximity to the weld increased. Despite the large difference in the levels of deflection observed for the different liquids modelled, the maximum levels of bending moment near to the bands varied only slightly. This is likely to be due to the plastic strains in the shell limiting the level of bending moments possible. Values of bending moment and membrane stress were obtained for the weld location by extrapolating the values from the fuel oil model with an impact velocity of 2.6 rad/s. Single values for normalised bending moment (1,460 Nmm/mm), which corresponds to an elastic-plastic, through wall bending stress of 254 MPa, and membrane stress (21.5 MPa) at the front side of the rear extrusion band in compartment 4 were extrapolated from model elements close to the circumferential weld and supplied to WP2 for a detailed assessment of the weld characteristics (ECA). Due to the limiting effect of the plastic strains in the shell, other loading scenarios would be unlikely to give significantly different results.

3.3.3.2 Fuel load conclusions

It is clear that the behaviour of a tanker is dependent on the load that it carries. From the modelling of a tanker with full loads of fuel oil and petrol it has been shown that there is the potential for higher levels of damage to occur than if an equivalent mass of water were used. It has also been shown that loading configuration (for example, running with a compartment empty) can change the pattern of the deformation. This has been shown by the increased deflection at the rear of the tanker, the higher levels of bending moment, and the different behaviour of some bulkheads due to the empty compartment.

3.4   Output for WP2 Engineering Critical Assessment

For the WP2 engineering critical assessment (ECA) of the circumferential welds, the membrane and bending stresses acting on the joint were required. For WP2, TWI created axisymmetric finite element models of the weld geometries to obtain stress intensity factor and reference stress solutions for different crack depths and weld cap geometries. The inputs for these models were bending moment and membrane stress. As these variables were more readily obtained from the WP1 tanker models than through-wall bending stresses, it was bending moments that were supplied. Also, the relationship between bending moment and through-wall bending stress was dependent on the thickness of the section at the point of interest (which was a variable in the ECA work) and the stress-strain relationship assumed.

The bending moments were highest at the rear of the tanker. The patterns and values of bending moments were very similar for the fuel oil and petrol models, with the exception of the area adjacent to band E/8, when the compartment was empty for the fuel oil model. There were areas of high bending moment within the impact zone, and also outside the impact zone close to the bands. The high bending moments here lay between the end of the impact zone and the comb, with the highest moments moving up the tanker surface as the size of the impact zone flat increases. The bending moments near to band F/8 were very similar between the fuel oil and petrol models.

In these areas, on the convex side of the bulkheads, the bending moment increased rapidly as proximity to the weld increased. Despite the large difference in the levels of deflection observed for the different liquids modelled, the maximum levels of bending moment near to the bands varied only slightly. This is likely to be due to the plastic strains in the shell limiting the level of bending moments possible. Values of bending moment and membrane stress were obtained for the weld location by extrapolating the values from the fuel oil model with an impact velocity of 2.6 rad/s. These values of 1,460 Nmm/mm for bending moment and 21.5 MPa for membrane stress were supplied to WP2 for the ECA. Due to the limiting effect of the plastic strains in the shell, other loading scenarios would be unlikely to give significantly different results.

During post-mortem examination of the rear of the 2008 tanker as part of WP2, a through-wall crack along the circumferential weld at the top of the impact zone in Band H+ was examined. Values of the bending moments in this area were obtained by similar extrapolation from the modified finite element model (with a water load) using the same 1.89 rad/s impact velocity as in the 2008 tanker topple test. The results are shown in Figure 25. Bending moments at the band H+ weld plane for the locations assessed ranged from 1,350 Nmm/mm to 1,500 Nmm/mm. These values were supplied to WP2 for the ECA, and were similar to those obtained for the fuel cases modelled at Band F+.
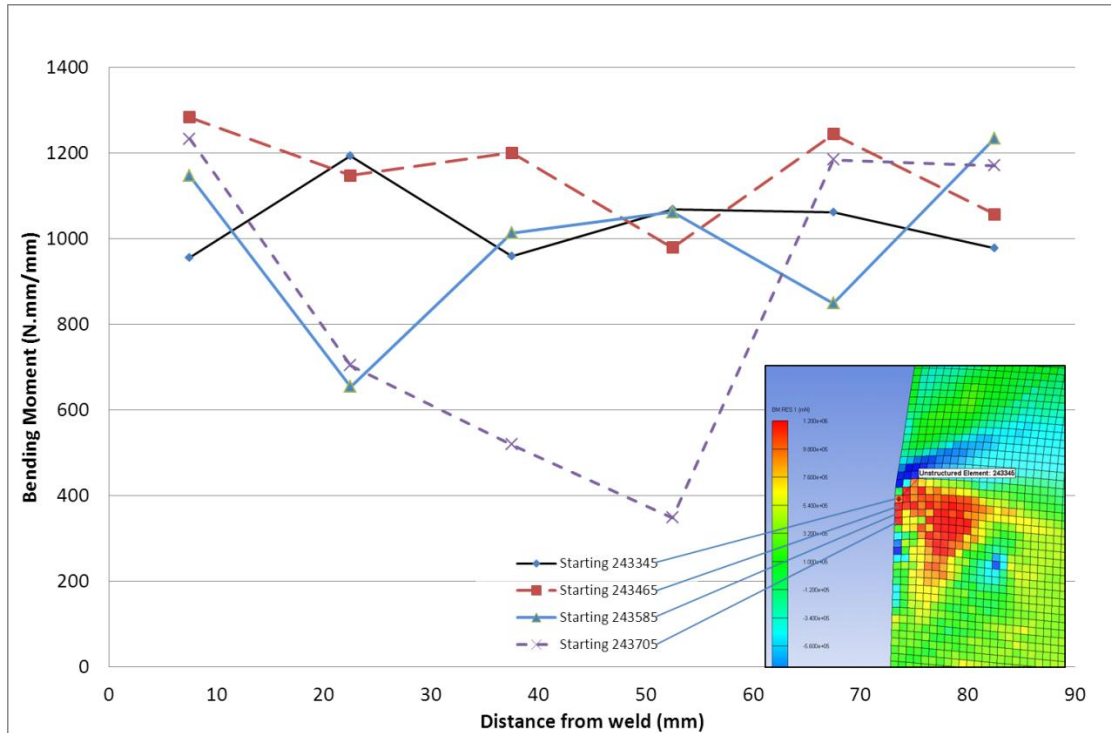
Figure 25. Variation in bending moments with distance from the rear weld (Band H+) for water at 1.89 rad/s

4 WP2 – Detailed Fracture and Fatigue Engineering Critical Assessment

4.1 Research methods

WP2 consisted of five tasks (in addition to cross-consortium peer review activities):

Task 1. Determine the typical in-service life cycle fatigue loadings at worst case locations on the circumferential weld seam, including, when appropriate, the effects of filling and dispensing from compartments;

Task 2. Review a proprietary finite element model of the non-compliant tanker as well as the crack growth and leak-before-break and associated technical documents and studies undertaken by its manufacturer and TWI;

Task 3. Address deficiencies as appropriate (such as fracture toughness properties, fatigue crack growth rates and weld residual stresses, wherever possible using strength and fatigue tests of samples taken from tankers to validate the model);

Task 4. Engage with the manufacturer to solicit and incorporate views as appropriate;

Task 5. Undertake a detailed engineering critical assessment to predict crack growth, likely fatigue life of the circumferential welds and defect sizes under suitable loading conditions, incorporating geometric variability;

The following section summarises the findings of the work undertaken in WP2.

For simplicity and reader convenience, Task 2 is described first and is hereafter referred to as the 'review of previous analyses'. Task 1 is then described, under the title 'fatigue data collection'. Finally, Tasks 3 – 5 are combined as 'Engineering Critical Assessment'.

4.2 Results

4.2.1 Review of previous analyses

In order to provide insight and guidance for the subsequent activities of WP2, TWI has reviewed previous technical reports generated before May 2014, by the manufacturer of the non-compliant tankers ('the manufacturer'), TWI, DfT and other technical bodies in relation to the integrity of the circumferential welds, as well as any appropriate supporting documents and material. The following conclusions have been reached:

1. The methods the manufacturer used to investigate fatigue crack growth are not sufficient to absolutely determine the absence of fatigue cracks in the 2007 tanker assessed.

2. The engineering critical assessment performed by the manufacturer concludes that a 2.0 mm deep flaw is acceptable under rollover conditions. Different conclusions have been reached by an HSE study and TWI work. A sensitivity-study is required to determine the margin of acceptability of the 2.0 mm deep flaw considered, due to the evidence provided by HSE that flaws with depths greater than 2.0 mm exist.

3. The review, carried out by Prof Issler on behalf of the manufacturer, of the significance of the manufacturer's studies in comparison with the HSE's findings demonstrates that there is a gap between experimental observations and testing and theoretical calculations. TWI agrees with several of Prof Issler's observations about

methods to reconcile these differences; however, unlike Prof Issler, from the contrasting evidence presented, TWI cannot draw a definitive conclusion about the acceptability of a 2.0 mm deep flaw.

4. The review of TWI work by Prof Issler on behalf of the manufacturer highlights potential sources of over-conservatism in the ECAs conducted by TWI. The assumptions made by TWI were explicitly stated and documented in the TWI reports and scope of work. Within the current work programme, specific tasks have been designed to further study these potentially over-conservative assumptions through experimental testing and advanced numerical modelling.

5. The manufacturer has developed a detailed finite element model of a ten-banded tanker and performed a stress analysis of this tanker subjected to various loads to derive inputs for their ECA calculations. Based on a review of the model and the post-processed results, it is recommended that a more consistent stress extraction method is used in order to ensure that the stresses obtained from the model are in line with the guidelines for stress extraction recommended in BS 7608.

In addition to the main conclusions above, TWI has also considered several aspects of the work reviewed where additional detail could be provided to further substantiate the argument.

On the evidence provided within the documents reviewed, therefore, TWI does not conclude that the manufacturer's analyses have demonstrated that:

- Under normal operations, the tankers will definitely remain safe after six years of use;

- The critical flaw depth in rollover conditions exceeds 2.0 mm rather than 1.2 mm.

Note that TWI is not concluding that these statements are incorrect, merely that they had not, on the basis of the review, been demonstrated to the satisfaction of the relevant standards.

4.2.2   Fatigue Data Collection

4.2.2.1   Overview

In order to determine typical in-service life cycle fatigue loadings on the circumferential welds of the non-compliant tankers, DfT provided a tanker manufactured in 2010 for testing within WP2. A fatigue data collection exercise was carried out using Wincanton Group Ltd premises in Thurrock.

A route for the tanker was chosen by Wincanton in correspondence with TWI and DfT. The route was selected as it followed a typical journey that Wincanton-operated tankers would follow and comprised significant portions of motorway, A and B class roads, including urban sections.

The tanker was instrumented with electrical resistance strain gauges on the outer surface of the tanker adjacent to the circumferential welds, and with two accelerometers, one each on the front and rear of the chassis. Data was recorded with the tanker unladen while the vehicle was driven on a route representative of a typical tanker journey (sections of motorway, Class A and B roads), and put through a series of manoeuvres. The vehicle was then filled with water of an appropriate volume to represent the mass when fully laden with petrol. Data was recorded during the filling

operation. The route was repeated and the tanker was then emptied, again with data recording. The data was then post-processed to determine the frequency of occurrences of stress ranges of interest at each instrumented position.

The representative fatigue stresses so derived were then used to calibrate and validate a finite element model of the welded joint of interest (which was subsequently used as part of the ECA).

The tanker used is illustrated in Figure 26 and Figure 27, as well as the band naming convention; Band A/10 at the front (or A for simplicity), and J/10 (or J) at the rear.
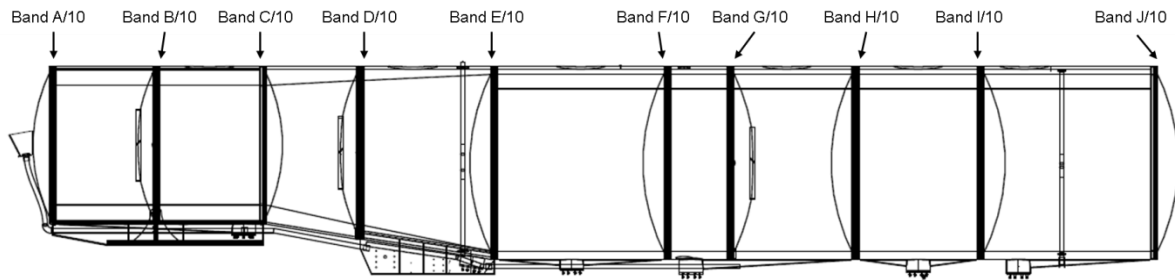
Figure 26. Planar view of the 2010 tanker, with the band naming convention
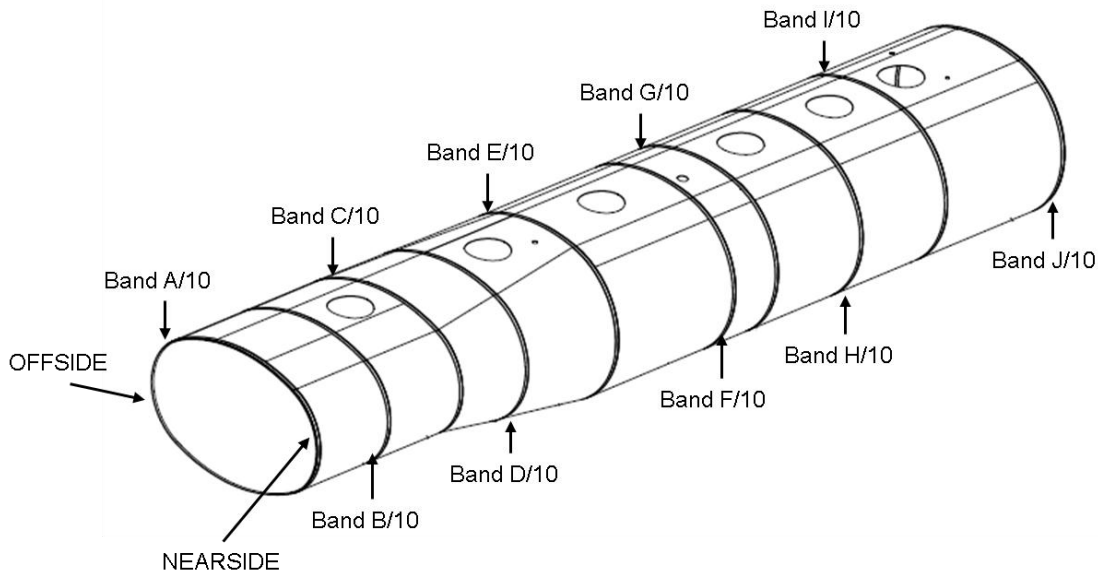
Figure 27. Isometric view

4.2.2.2 Internal fillet weld inspection

Before any instrumentation was applied to the tanker an ultrasonic inspection was performed in order to determine the existence and location of internal fillet welds for each of the tanker band welds. The objective of this inspection was to provide information about the fillet weld geometry and location in order to prevent strain gauges being attached at positions where fillet welds were present, which could then result in inaccurate post-processing of the strain data.

It was noted that a continuous fillet weld joining the toe of the extrusion profile to the tanker shell was present in bands C, D, E, F and G from positions 3 to 9 o-clock (bottom half of the tanker).

From 9 to 3 o-clock positions (top half of the tanker), the fillet weld was "stitched", typically alternating 100mm weld and 600mm gap.

Bands A, B, H, I and J were noted to be stitched over their full circumference, but the stitched weld pattern was irregular and varied considerably throughout the ten bands of the tanker.

The approximate location and toes of each internal fillet weld were marked with indelible ink on the outer face of the tanker shell to facilitate marking out of the strain gauge positions relative to the weld toes.

4.2.2.3   Tanker instrumentation

In total, 62 electrical resistance strain gauges were attached to the outer surface of the tanker at various positions around the circumference and along the length of the tanker. Two of the gauges were attached to unstrained locations to assess the extent of electrical noise, for example from passing under overhead power lines.

The strain gauge plan was developed based upon the findings of the previous work. From the results of these simulations, TWI identified regions where significant stresses acted normal to the circumferential welds. These regions in turn represent locations where it is likely that fatigue damage may be most severe. Based on this review of the simulation results, the following strategy was employed for the strain gauge positions:

- Tanker bands B and G were the most densely instrumented. Along these bands, both circumferentially- and axially-oriented strain gauges were placed at multiple positions along the circumference. In particular, the regions near the cradle featured a cluster of four axial gauges and one circumferential (hoop) gauge so that local biaxial stresses could be calculated and linearly extrapolated back to the hypothetical crack plane.

- Tanker bands C and D were the next most densely instrumented. For these two bands, circumferentially-oriented gauges were not employed, but axial gauges at the same circumferential position having different longitudinal offsets from the welds were used to enable linear stress extrapolation back to the hypothetical crack plane.

- For the remaining circumferential welds, a single axial gauge was placed on the offside of the tanker where the cradle is attached to the tanker. This allowed for strain data to be collected from the same position from each circumferential weld.

- Two "remote" axial gauges were placed on the tanker away from the circumferential welds and other local stress raisers. One was placed half-way between bands E and F and one was placed between bands I and J. In both cases, the gauges were located on the offside, mid-height. The purpose of these two gauges was to provide additional model validation/calibration in regions where high strain gradients were not expected.

Additionally, two tri-axial accelerometers were mounted on the tanker. The front accelerometer was mounted on the chassis behind the king pin and the rear one was mounted on the chassis at the rear of the tank at half width.

4.2.2.4   Unladen fatigue data collection

The unladen fatigue data collection exercise was carried out on Tuesday 13 and Wednesday 14 May 2014.

Two controlled emergency stops were performed during the unladen fatigue data collection exercise and the following additional events/incidents were recorded:

- Number of recorded speed humps: 20
- Number of recorded pot hole events: 6
- Number of roundabout encounters: 92

4.2.2.5   Filling compartments fatigue data collection

The tanker was filled with an equivalent mass of its normal petrol capacity on 13 June 2014. The compartments were filled sequentially from compartment (pot) 1 at the rear of the tanker to compartment (pot) 6 at the front of the tanker.

4.2.2.6   Laden fatigue data collection

The laden fatigue data collection exercise was carried out on 13 June 2014.

Two controlled emergency stops were performed during the laden fatigue data collection exercise, at nominally the same location as those completed for the unladen test, and the following additional events/incidents were recorded:

- Number of recorded speed humps: 6
- Number of recorded pot hole events: 5
- Number of roundabout encounters: 94

Note that any discrepancy between the figures for the laden testing and unladen testing may not be due to different routing but may arise from a lack of recording of the event.

4.2.2.7   Emptying compartments fatigue data collection

The tanker was emptied on 16 June 2014. The compartments were emptied sequentially from compartment (pot) 1 at the rear of the tanker to compartment (pot) 6 at the front of the tanker. During the emptying process, the tanker instrumentation was active and recorded the resulting strains and accelerations.

4.2.2.8   Data processing

All measured strains were converted into stresses acting normal to the circumferential welds (axial stresses). The precise method used depended on the number and orientation of gauges at a given location.

4.2.3   Construction of duty cycle (distance-based approach)

The approach employed by the tanker manufacturer in its analyses assumed that a typical high utility duty cycle in industry includes the following:

- Annual travel distance: 220,000 km;
- Number of loads delivered per day: 6;
- Ratio of laden vs unladen travel distance: 50/50.

The route travelled for both the laden and unladen fatigue data collection exercises was an actual route that would be travelled by a fuel tanker and comprised 150 miles (241 km). Due to the repetitions included in the route planning, the responses obtained over this 150 miles could be assumed to be representative of a typical UK tanker route in the Southeast of England.

To employ a distance-based duty cycle construction, first, all of the data recorded from the unladen test was concatenated. This concatenation did not include the figure-of-eight manoeuvres or the emergency stop tests. This is in accordance with BS 7910 as such events would not produce a significant impact on the fatigue life. Similarly, all of the laden data was concatenated.

Once the unladen and corresponding laden fatigue data has been concatenated, it was further partitioned into data corresponding to Class A roads and data corresponding to Class B roads. This partitioning will create sets of fatigue data comprising X_A miles of Class A data and X_B miles of Class B data, where $X_A + X_B = 150$ miles. A rainflow counting procedure was performed to determine the number of cycles per stress range for the Class A data and the Class B data. This gave the stress range histogram corresponding to X_A miles of Class A road and the stress range histogram corresponding to X_B miles of Class B road. These were then normalised to provide stress range histograms per mile.

Finally, in order to generate the annual, 220,000 km (136,701 miles) duty cycle, an assumed ratio of Class A to Class B road travel was selected. Then the appropriate linear combination of the normalised Class A and Class B histograms was applied to obtain the annual duty cycle. In addition to the fatigue data arising from road travel, the stress range histograms arising from emptying and filling 6 times per day were added. In the results that follow, the fatigue life is presented in terms of years. It is important to note that each "year" is taken to mean a distance travelled of 220,000 km.

An alternative, time-based approach was also considered, but due to various ambiguities that may arise with such an approach, the above distance-based approach was preferred. Correspondence with Wincanton recommended the 220,000 km per year figure which is in exact agreement with the manufacturer; therefore, an annual distance of 220,000 km was employed in the fatigue stress spectra generation. It was determined that the dependence on Class A and Class B roads was insignificant. Therefore, the fatigue stress spectra used assume 60% Class A road and 40% Class B road travel annually. This is the ratio identified during the fatigue data collection exercise.

4.2.4   Engineering Critical Assessment - procedures

4.2.4.1   Overview

A detailed engineering critical assessment has been undertaken in order to assess the structural integrity of circumferential welds in the non-compliant tankers that may contain crack-like defects. The objective of the ECA was to assess the acceptability of lack of fusion defects in terms of their fracture and fatigue integrity. All assessments have been performed in accordance with the methods and guidance of BS 7910:2013.

The review of previous analyses highlighted several potential shortcomings in past ECAs such as overly conservative geometric considerations and the treatment of residual stresses. In order to refine the assessment calculations, the effect of the weld cap has

been analysed, the effect of a potentially more realistic welding residual stress profile, and the fatigue data collected from UK roads have been considered. Consequently, the primary objective of the ECA detailed below was to quantify the effects of geometric variability and to refine previous fatigue and fracture assessments, considering the sensitivity of the results to changes in the input data.

4.2.4.2   Failure assessment diagram methods

The failure assessment diagram specified by BS 7910:2013 is comprised of two axes: K_r and L_r. The K_r axis quantifies the proximity of the flawed structure to fracture failure, and the L_r-axis quantifies the proximity of the flawed structure to plastic collapse. A curve called the failure assessment line separates the acceptable (or safe) region from the unacceptable (or potentially unsafe) region. Points contained within (below) the failure assessment line are considered acceptable, whereas points outside of the failure assessment line are considered unsafe.

4.2.4.3   Joint geometry

Considerable attention has been paid to the geometric variability associated with the tanker band joint. In particular, the review of previous analyses highlighted the need for careful consideration of the effect of the weld cap height, weld cap width and the presence of axial misalignment. In previous TWI and HSE studies, a "flush ground" joint was considered, in effect, assuming no weld cap was present. This approach is conservative, and, in the absence of measurements of the weld dimensions and their statistical variance, appropriate, but the level of potential over-conservatism was not fully quantified. Therefore, to refine the previous ECA work, the effect of the presence of a weld cap/overfill and axial misalignment on the stress intensity factor solutions and plastic collapse loads was quantified.

A finite element model of the joint was developed (Figure 28). The local model comprises the extrusion band profile, two lengths of tanker shell (either side of the extrusion band), and a bulkhead. The underlying geometry is an axisymmetric model of the tanker band joint. The mean radius of the tanker joint model was assumed to be 2,000 mm, approximately equal to the average of the major and minor axes of the tanker along its length. The nominal shell wall thickness was 5.0 mm. The extrusion band profile was provided by the manufacturer as an AUTOCAD sketch that was imported into model and used to define the axisymmetric part. A bulkhead was modelled based upon engineering drawings provided. A "seam" was created to represent the unfused surface between the extrusion band beyond the positioning lip and the inner surface of the tanker shell.
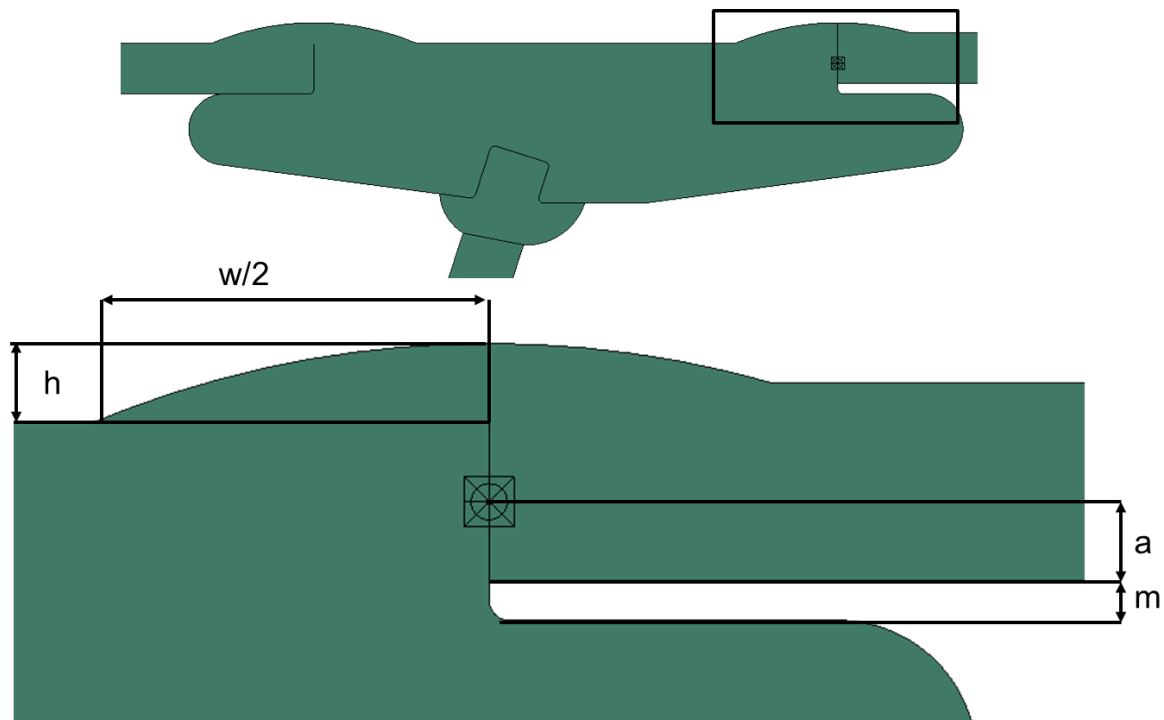
Figure 28. Definition of the geometry dimensions. Weld cap height is h; weld cap width is w; crack height is a and misalignment is m.

Once the base geometry had been defined, a weld cap profile was added. The weld cap was defined by two parameters: the height, h, and the width, w. The weld cap profile assumed the shape of a circular arc passing through three points: one being the apex of the circle positioned on the crack plane a distance h beyond the nominal tanker outer surface, and the other two points were positioned w/2 either side (in the axial direction) of the apex of the weld cap, as shown. In general, the position of the "peak" of the weld cap may or may not be directly above the crack plane, and the weld cap will not assume the idealised shape of a circular arc. However, it is not feasible to model all possible weld cap profiles, and the definition of weld cap geometry assumed in this report is sufficient to represent the typical weld cap shape and therefore capture the mechanics of the welds under consideration.

Axial misalignment was modelled by offsetting the section of tanker shell containing the defect from the axis of symmetry by a distance m.

It is important to note that the flaw depth, a, is measured from the inner surface of the tanker shell and not from the tanker extrusion band. This provides a consistent definition of crack depth when misalignment is present. Thus, for a geometry case with misalignment, m, and crack depth, a, the crack tip is located m+a from the unfused surface between the extrusion band and the inner surface of the tanker shell.

It is known from engineering drawings and inspection of tankers that an internal fillet weld is typically present from the 3 o'clock position to the 9 o'clock position for most bands. Additionally, this fillet weld is continued by "stitching" the extrusion band toe to the inner surface of the tanker shell from the 9 o'clock position back to the 3 o'clock position for some bands (though the stitching pattern is frequently irregular and the presence is not well defined). Nevertheless, the presence of the additional internal fillet

weld is significant as the positions experiencing the peak stresses under design (ADR) load cases and the most damaging fatigue stress spectra are along the bottom of the tanker. Therefore, the last geometric modification considered was the presence of an additional internal fillet weld joining the toe of the extrusion band to the inner surface of the tanker shell. In order to reduce the scope of the parametric study, the leg length of the fillet weld was assumed to be 12.0 mm, independent of whether or not misalignment was present. The detailed geometric representation of the fillet weld in the model is not considered to be particularly important; the primary factor is that the existence of the fillet weld provides an alternative load path that may significantly reduce the crack tip loading and thus the crack tip stress intensity.

4.2.4.4   Definition of geometry cases

The 2011 tanker was laser scanned prior to topple testing by HSL and the preliminary measurements were provided to TWI. Based on the HSL preliminary measurements, ten geometric cases were defined to analyse the sensitivity of the engineering critical assessment to geometric variability. The ten cases were defined as follows:

Cases 01-08 comprised of all possible permutations of maximum and minimum values: 3 variables (cap height, cap width, and misalignment), 2 values (maximum and minimum) resulting in $2^3 = 8$ permutations. Note that because of the definition of the geometry, it is not possible to pair a value of axial misalignment, m with a value of weld cap height, h, if m > h. Therefore, when appropriate, the value of axial misalignment has been adjusted to the maximum possible for that arrangement.

Case 09 uses the average values for the geometric dimensions. Although the measurements were taken from a single tanker, Case 09 can be seen as a representative average to compare to the more extreme cases 01-08 (Figure 29).

Case 10 is defined as the flush-ground geometry with no misalignment as previously considered in HSE and TWI studies. This case was used as a benchmark for comparison with the other 9 cases.
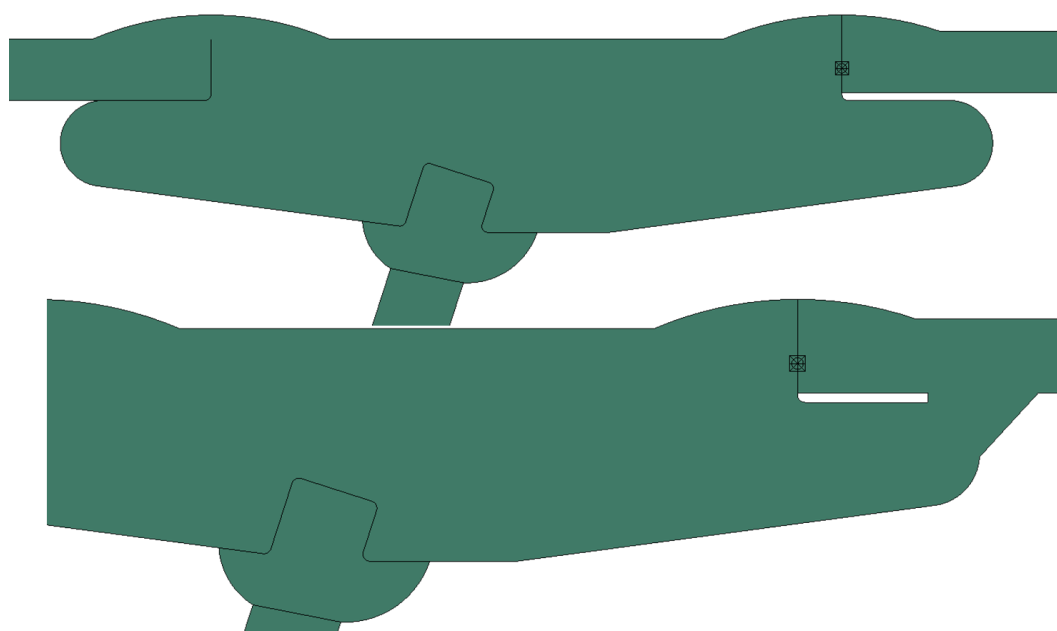

Figure 29. Geometry for Case 09 (the average dimension model) with and without the additional internal fillet weld.

4.2.4.5   Material properties

Detailed information about the properties of the materials under consideration in an ECA allows for a reduction in conservatism by expanding the acceptable region underneath the failure assessment line. A full stress-strain curve allows for an Option 2, material-specific failure assessment line to be generated. In contrast when only single point tensile properties or information from literature is available, an Option 1 failure assessment line is used.

Additionally, the material properties obtained from mechanical testing in this research has allowed for a quantitative comparison and characterisation of the variation of material properties across multiple non-compliant tankers to be assessed.

The tensile properties and fracture toughness properties of the materials (parent and weld metal) were measured and used in the detailed ECA.

4.2.5   Engineering Critical Assessment – applied load cases

In order to determine the fatigue life, two items are required: (1) the fatigue stress spectrum, and (2) a set of primary stresses. The fatigue stress spectrum was determined from the full-scale tanker fatigue data collection exercise. The primary stresses were determined to be the most severe stresses experienced in the tanker under any one of four ADR design load cases.

The primary stresses allow for determination of the critical defect size, i.e. the maximum tolerable flaw size. The fatigue stresses allow for the determination of how long it takes an initial sub-critical defect to become critical (the fatigue life).

Two sets of distinct load cases were considered: ADR load cases and rollover. The ADR load cases were considered as realistic design load cases that the tanker is likely to be subjected to (or experience) throughout the duration of its operation. In addition to the ADR load cases, two "rollover" load cases were considered: (1), a previously analysed case arising from a 2 bar pressure-impulse simulation, and (2), the topple test load case as analysed by HSL in WP1.

4.2.5.1   ADR load cases

The "ADR load cases" were obtained from ADR (2013). As per Section 6.8.2.1.2 of ADR:

"The tanks and their fastenings shall be capable of absorbing, under maximum permissible load, the forces exerted by:

In the direction of travel: twice the total mass" (ADR Load Case 1);

"Vertically upwards: the total mass" (ADR Load Case 2);

"Vertically downwards: twice the total mass" (ADR Load Case 3);

"At a right angle to the direction of travel: the total mass" (ADR Load Case 4).

In order to obtain the primary stresses arising from the ADR load cases, the manufacturer's existing proprietary finite element model of a 10-banded tanker was modified. The main objectives of the modification were to improve the quality of the finite element mesh (to facilitate more controlled stress extrapolation) and to change the geometry to match that of the specific (2010) tanker employed in the fatigue data collection exercise.

The analysis of the modified tanker model resulted in the definition of the ADR load case for the present ECA. This was done by identifying the largest tensile net section, membrane and through-wall bending stresses at each of the tanker bands for each ADR load case. The results indicated that both tanker bands B/10(-) and E/10(+) at the triple joint position, where the cradle gusset plate is joined to the tanker shell in the vicinity of the chassis rails, were the most severely stressed, under ADR Load Case 3. Based on considerations of the proportion of the net section stress that was through-wall bending stress, it was determined that tanker band E/10(+) was the critical position to assess. At this location, under ADR Load Case 3, the following stresses are present:

- Net section stress: 67.45MPa

- Through-wall bending stress: 40.53MPa

- Membrane stress: 26.92MPa

The primary stresses above therefore define the "ADR Load Case" considered in the ECA. Note that all of the most highly stressed positions under the ADR design load cases are located along the bottom of the tanker for each tanker band. From measurements and observations within the scope of this work, these positions experiencing the highest stresses are positioned along bands where an additional internal fillet weld is present (as this weld is typically present from the 3 o'clock to the 9 o'clock position).

4.2.5.2 Roll-over load cases

Two "rollover" load cases were considered.

The first load case comes from the results of the previously analysed 2 bar pressure-impulse simulation. In summary, in order to simulate the conditions of a rollover, the manufacturer simulated the tanker being subjected to a short-term 2 bar internal pressure impulse. The dynamic stress analysis was analysed to identify the time increment at which the peak tensile stress occurred. At this time increment and at the position of the peak tensile stress, the local section stresses (membrane and through-wall bending) were evaluated.

The peak through-wall bending stress attained was approximately 150 MPa. At this location, the membrane stress was negligible. Therefore, in the present ECA, "rollover load case 1" is considered to be a pure, through-wall bending stress with the applied bending stress equal to 150 MPa.

The second rollover load case comes from the results of the HSL topple tests and modelling conducted within WP1. This simulation involved the tanker filled with fuel oil and a rotational velocity at impact of 2.6 rad/s. Henceforth, this load case is referred to as the "Rollover load case 2". The key results of the HSL modelling were as follows:

- The critical location is Band F/8(+) just above the impact zone;

- The high stress area extends circumferentially along a length of approximately 250 mm;

- The local, normalised bending moment is 1,460 Nmm/mm which corresponds to an elastic-plastic, through-wall bending stress of 254 MPa;

- The section membrane stress is 21.5MPa;

- When the results of the finite element simulation are compared to the experimental test results, the FE predications agree with the experimental measurements to within 5% at the closest gauge locations on Band F/8.

Therefore, similar to rollover load case 1, this load case is predominantly through-wall bending and involves stresses that generate plasticity (above yield stress).

The methods in BS 7910 predominantly relate to stress-based assessments, i.e. when the nominal stress is lower than the yield strength of the flawed section. For both rollover load cases, the stresses are well above the yield strength of the flawed section, as the yield strength of the weld metal has been taken to be 133 MPa. Therefore, the methods of a stress-based assessment may not be wholly appropriate for the fracture assessment of the tanker joints under rollover load cases. Moreover, the stresses experienced during the rollover (or topple test) are dynamic and thus occur over short time-periods. In order to assess these cases, the associated strain rates that the section under consideration experiences are required in order to ensure that a suitable fracture toughness definition is being applied. A standard, stress-based ECA assumes quasi-static fracture toughness properties.

The methods of a stress-based ECA have been applied to both the rollover load cases described. However, the applicability and accuracy of the ECA calculations should be considered in the context of the preceding discussion.

4.2.5.3   Summary of the primary stress load cases

The primary stress load cases are summarised as follows:

- ADR load case - design load case; position subject to most significant stresses amongst all ADR load cases.

  - Membrane stress: 40.53 MPa

  - Through-wall bending stress: 26.92 MPa

- Rollover load case 1 - derived from 2 bar pressure-impulse simulation.

  - Membrane stress: 0.00 MPa

  - Through-wall bending stress: 150 MPa

- Rollover load case 2 - derived from HSL fluid-structure interaction fuel oil simulation of topple test.

  - Membrane stress: 21.50 MPa

  - Through-wall bending stress: 254 MPa

4.2.5.4   Secondary stress

In the previous TWI ECA, due to the absence of any residual stress measurements or additional guidance, the defects under consideration were assumed to be subject to yield magnitude, tensile residual stresses as recommended by BS 7910. This assumption was thought to be a potential source of over-conservatism.

In order to improve upon the residual stress input for the engineering critical assessment, TWI was provided the tanker band joint welding procedure specification (WPS) by the manufacturer. The WPS was used to develop a detailed thermo-elastic-

plastic finite element simulation of the welding process in order to predict the likely residual stress profile.

The finite element simulation of the welding residual stresses involved the use of material properties from literature, and therefore provides only indications of the likely residual stress profile. However, when compared to the experimental measurements, the agreement is relatively good.

The calculation of critical defects that follows considers both yield magnitude tensile residual stresses as well as the welding simulation residual stresses.

4.2.6   Fatigue crack growth calculations

Fatigue crack growth calculations were undertaken based upon the guidance of BS 7910 Clause 8.

4.2.7   Failure assessment line

A material specific (Option 2) failure assessment line was constructed based on the lower bound tensile stress-strain curve and was produced in accordance with BS 7910:2013.

4.2.8   Engineering Critical Assessment - results

4.2.8.1   Stress intensity factor solutions

For each geometry case (01 – 10), with and without the additional internal fillet weld, linear elastic finite element simulations were performed in order to evaluate the stress intensity factors. For each axisymmetric model, sharp cracks were modelled ranging in size from 0.75 mm deep flaws to 7.0 mm deep flaws, depending on the height and presence of the weld cap. Typically, 15 - 20 different cracks were modelled for each geometry, allowing for a smooth parameterisation of the geometry-specific stress intensity factor solution.

4.2.8.2   Considerations for finite length defects

The discussion of results thus far has considered only fully circumferential flaws, based upon the results of the axisymmetric finite element simulations. However, it is reasonable to consider finite length surface flaws, and therefore this section considers how to approximate the stress intensity factor solutions for finite length flaws from the existing fully circumferential solutions.

In BS 7910:2013, the stress intensity factor solution most similar to the tanker band joint under consideration is the curved shell with a fully circumferential or finite length internal surface flaw (Solution M.6). In this case, the long (fully circumferential) flaw solution is the upper bound of finite length flaw solutions. As the length of a finite length inner surface flaw increases, the stress intensity factor solution converges to the long surface flaw solution. Therefore, the effect of the finite length can be quantified.

Once the stress intensity factor solution is known for a long flaw, the stress intensity factor for a finite length flaw for the same geometry can be approximated.

4.2.8.3   Plastic collapse solutions

The definition of the load ratio in an ECA is given by:

$$L_r \quad = \quad \frac{P_a}{P_c}$$

Where

L_r is the load ratio;

P_a is the applied load (or applied stress);

P_c is the collapse load (or collapse stress).

The plastic collapse load is evaluated using finite element analysis by assuming an elastic-perfectly-plastic material behaviour and incrementally and proportionally applying the loads until the plastic zone at the crack tip spreads through the remaining ligament ahead of the crack tip. The level of applied loads when this criterion is met is considered to be the collapse load. This definition of the collapse load is more specifically the net section collapse load, as it refers to the collapse of the local section containing the flaw. For the various geometry cases assessed, when flaws are shallow or, more frequently, when the additional internal fillet weld is present, collapse of the joint occurs away from the section containing the flaw.

BS 7910:2013 notes that such a remote (or global) collapse can be used as the plastic collapse load in an assessment, but it might be overly conservative. An alternative approach to employing a global collapse solution is to use what's known as a J-based approach. When appropriate, or the global solution was deemed too conservative, this J-based method was used.

4.2.8.4   ADR load case assessment

The critical defect sizes were determined by changing the crack size until the assessment point intersected the failure assessment line. For fully-circumferential flaws, the smallest critical defect amongst all geometry cases was 2.1 mm, and this occurred in a geometry in which significant misalignment was present. The critical defect size for a long surface flaw in the "average" geometry (Case 09) was 2.56 mm whereas it was 2.48 mm for the flush ground joint (Case 10). A chart of the results is shown in Figure 30, which includes the results for finite length flaws, as well as those obtained from using the curved shell solutions in BS 7910.

When the additional internal fillet weld is present, the critical defect size is almost always larger than the deepest flaw modelled. For this reason, it may be assumed that surface flaws under the ADR load case when the additional fillet weld is present are acceptable.
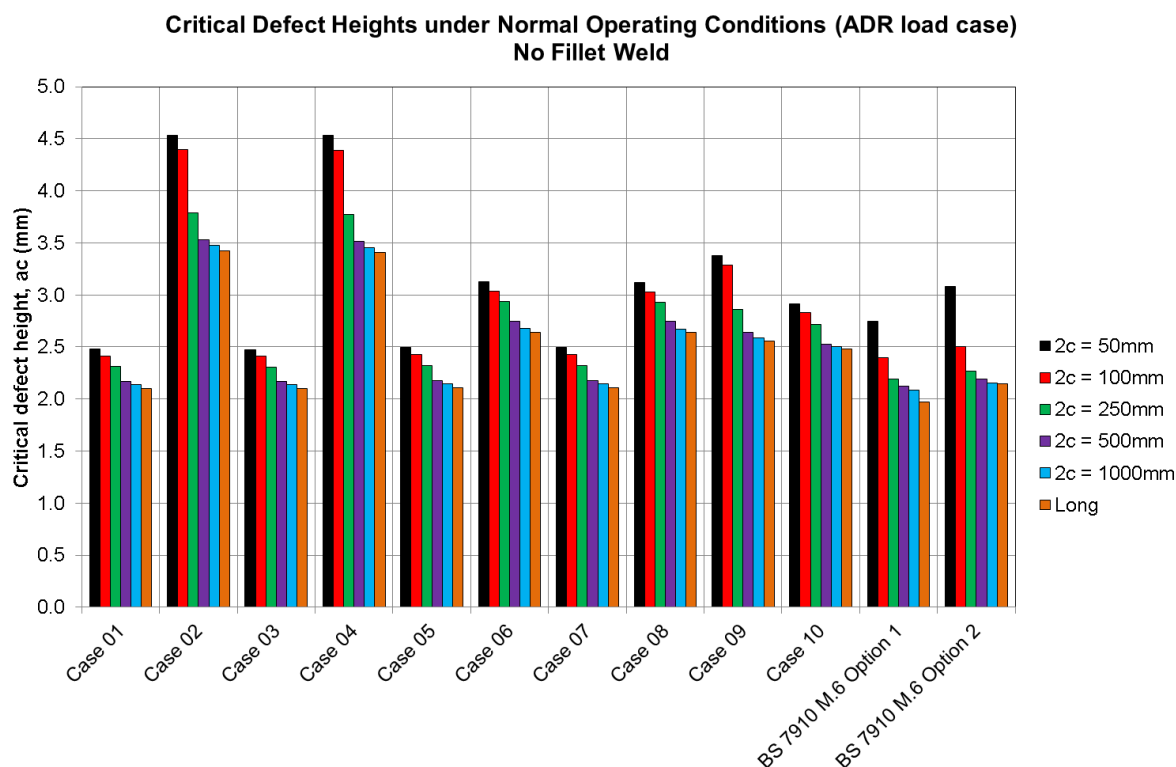
Critical Defect Heights under Normal Operating Conditions (ADR load case)
No Fillet Weld

Figure 30. Critical defect heights, ADR load case without internal fillet weld.

As a consequence of the potential for large critical defect heights for the inner surface flaws analysed, consideration was given to leak-before-break and through-thickness flaws. Calculations were performed using the BS 7910:2013 solutions for through-thickness flaws in curved shells (solutions M.6 and P.10.1). Under the ADR load case, the critical defect length, 2a, for a through-thickness flaw is 21.4 mm. In light of other calculations performed, it is expected that this may be a conservative approximation of the critical through-thickness flaw length for the actual joint geometry.

Consequently, although the critical defect length for a through-thickness flaw in the joint is most likely larger than the BS 7910 solution of 21.4 mm, it is not likely that any leak-before-break type consideration will lead to significant changes in the acceptable flaw sizes or fatigue life calculations. This conclusion is similar to that reached by HSE in one of the reviewed previous analyses.

The most highly stressed location under the ADR load case is positioned on tanker band E/10(+) with a nearly equivalently stressed position also on tanker band B/10(-). Fatigue life calculations were performed for each of the joint geometries with and without additional internal fillet welds. When the additional internal fillet weld is present, the calculated fatigue life is in excess of 20 years for all geometries considered. However, inspection of the several tankers during the course of the project indicated that in some highly stressed regions that are susceptible to fatigue damage, an internal fillet weld is either not present or is present but stitched. The gap between the welded lengths and un-welded lengths (with a stitched fillet weld) may prevent the strengthening of the joint by the fillet weld. Additionally, the quality of the fillet weld will have an effect on how well it may reduce the stresses acting on flaws in the circumferential seam welds. Therefore, the fatigue life calculations (and the lower bound

fatigue life estimation curve discussed below) do not consider the presence of an internal fillet weld.

When an additional internal fillet weld is not present, for a 100 mm long surface flaw with an initial defect height of 2.0 mm, assuming the one-stage Paris law and that the fatigue stress ranges are pure membrane stress, the "average geometry" Case 09 has a fatigue life of 14.8 years. Other geometries produce calculated fatigue lives ranging from 3.8 to 39.3 years. Note, however, that the geometries considered do not necessarily represent geometric configurations that have been measured from actual tanker joints. Consequently, TWI is not suggesting that measurements of actual joints to-date indicate that there is the potential for a fatigue failure after 3.8 years, as indicated by one case modelled. Instead, a parametric study was used to understand the relationship between geometry and fatigue life, as described in more detail below (4.2.8.6).

4.2.8.5   Roll over load case assessments

Critical defect sizes have been calculated for rollover load case 1 and rollover load case 2 for the geometry cases 09 and 10 without the additional internal fillet weld.

For rollover load case 1 which was derived from the 2-bar pressure-impulse scenario:

- The "average" geometry" Case 09, without an additional internal fillet weld and making allowance for some stable ductile tearing to occur, has a critical defect height of 2.5 mm.

- The flush ground joint geometry, Case 10, without an additional internal fillet weld has a critical defect height of 1.5 mm for a long surface flaw based on the Option 2 failure assessment line.

- For comparison with Case 09, the BS 7910 curved shell solution with a long internal surface flaw and nominal wall thickness of 7.0 mm (with the stresses adjusted appropriately) has a critical defect height of 2.2 mm for the Option 2 failure assessment line, and a critical defect height of 2.1 mm for the Option 1 failure assessment line.

- For comparison with Case 10, the BS 7910 curved shell solution with a long internal surface flaw and nominal wall thickness of 5.0 mm (with the stresses adjusted appropriately) has a critical defect height of 1.0 mm for the Option 2 failure assessment line, and a critical defect height of 0.8 mm for the Option 1 failure assessment line.

For rollover load case 2, which was derived from the WP1 fuel oil, 2.6 rad/s impact velocity simulation:

- The "average" geometry, Case 09 without an additional internal fillet weld has a critical defect height of 1.1 mm for a long surface flaw when allowance is made for some stable ductile tearing.

- The flush ground joint geometry (Case 10), without an additional internal fillet weld has a critical defect height of 0.90 mm for a long surface flaw when allowance is made for some stable ductile tearing.

- For comparison with Case 09, the BS 7910 curved shell solution with a long internal surface flaw and nominal wall thickness of 7.0 mm (with the stresses adjusted appropriately) has a critical defect height of 1.35 mm for the Option 2

failure assessment line, and a critical defect height of 1.2 mm for the Option 1 failure assessment line.

- For comparison with Case 10, the BS 7910 curved shell solution with a long internal surface flaw and nominal wall thickness of 5.0 mm (with the stresses adjusted appropriately) has a critical defect height of 0.13 mm for the Option 2 failure assessment line, and a critical defect height of 0.1 mm for the Option 1 failure assessment line.

For both cases, the critical depth will increase when finite length flaws are considered, but this has not been quantified.

The results of the critical defect calculations for rollover case 2 should be interpreted in the context of the previous discussion: the applied stresses are significantly higher than the yield stress (in fact, the applied elastic-plastic bending stress 254 MPa is nearly equal to the ultimate tensile strength of the material, 270 MPa). For this reason, a stress-based ECA may not be the most appropriate assessment method to characterise the integrity of the joint in the presence of defects.

For the cases when an internal fillet weld is present, the collapse is dominated by global collapse (i.e. collapse of the tanker shell and not the section containing the flaw). This can be explained as follows:

- The presence of the internal fillet weld significantly reduces the stress intensity factor for bending stress.
- The load ratio is independent of the crack height because global collapse is occurring before local collapse.

Consequently, the results indicate that for geometry cases 09 and 10, when the additional internal fillet weld is present, both rollover load cases are insensitive to the presence of a crack-like defect and are primarily dominated by gross yielding of the tanker shell.

4.2.8.6   Fatigue life calculations

The objective of this section is to derive and propose a lower bound fatigue life estimation equation that takes into account the research results, but can also be applied to any weld geometry that may be encountered. This will allow a finite set of measurements, taken from the critical bands of a non-compliant tanker, to be used to arrive at a conservative estimate of the safe operating life of the tanker under normal operating conditions.

In the present study, normal operating conditions are defined in terms of primary stresses and fatigue stresses as follows:

- The primary stresses that a tanker circumferential seam weld is likely to experience at any point in its operating life are the severe stresses arising from the ADR design load cases as previously described.
- The fatigue stresses that a tanker circumferential seam weld will experience annually are derived from the 220,000 km fatigue stress range histogram from the most highly stressed bands (bands E and B for a 10-banded tanker).

To determine the fatigue life of a component using failure assessment diagram methods, four pieces of information are required: the initial defect size; the critical defect size; the fatigue stresses; and the primary stresses.

In the context of the present study the initial defect size has been selected to be a 2 x 100 mm flaw. There are multiple justifications for this flaw size selection. Firstly, during the metallographic examination exercise, a 2.19 mm height defect was found. It was not possible to characterise the length, other than to conclude it was less than 200 mm. Secondly, consideration of a 2 x 100 mm flaw provides consistency and direct comparison with the findings of the previous HSE report (2013). In the HSE fatigue life assessment report, a 2 x 100 mm surface flaw was chosen to be the initial defect size. The HSE justification for selecting the height of 2 mm was that the lack of fusion, arising from the presence of the positioner lip on the extrusion profile (when unfused), results in a 2 mm lack of fusion defect. The HSE justification for selecting the length of 100 mm was that there was relatively little change in the critical defect height for flaws longer than 100 mm. Finally, the tanker manufacturer has also identified and reported on a 2 x 80 mm defect. Therefore, the 2 x 100 mm initial flaw size is consistent with TWI, HSE and the manufacturer's own experimental measurements.

For each geometry case analysed, the critical defect size was calculated. Therefore, the fatigue life is determined to be the time (in 220,000 km years) that is required to grow an initial 2 x 100 mm flaw by fatigue to the critical defect size.

Joint geometry was found to significantly influence the fatigue life. Most importantly, misalignment and weld cap reinforcement appear to compete against each other:

- Increasing the weld cap reinforcement increases the fatigue life of the joint.
- Increasing misalignment decreases the fatigue life.

To determine the relationship between the weld cap size, misalignment and fatigue life, consider a normalised geometry factor, β, that effectively represents the amount of weld reinforcement, where:

β = (h – m)/5

h = weld cap height (mm)

m = misalignment (mm)

If β is small (approaching 0.0), then misalignment dominates, and the joint will have a short fatigue life.

If β is large (approaching 1.0), weld cap height dominates, and the joint will have a long fatigue life.

This definition of β does not include the weld width, w. It was found that w had only a minor influence on the fatigue life and measurements indicated that the weld width was relatively consistent. It is, therefore, reasonable to ignore w.

A quadratic relationship between the fatigue life and β has been established, and used for a conservative estimation of the fatigue life (for a joint without additional internal fillet weld) to be made based on the weld cap height and misalignment, assuming an initial 2 x 100 mm flaw. This is illustrated in Figure 31, where the fatigue life is plotted against the non-dimensional geometry parameter. In order to further validate the quadratic relationship, three additional cases were considered and are included in the plot as triangles. To take account of the spread in the data and also to incorporate a

margin of safety, a lower-bound, offset quadratic curve is also shown (the thick red line in the Figure).

Using this quadratic equation, it is possible to estimate the fatigue life of a joint based on the non-dimensional geometry parameter β. Alternatively, if a specific fatigue life (F) is required, then the corresponding value of β can be calculated. Since β depends on both h and m, this allows for the specification of a range of acceptable weld cap height and misalignment combinations that would achieve the target fatigue life. Look-up tables have been derived from this equation that could be used to assess the fatigue life based on measurements of the weld joint geometry (an example is provided in Table 6).

Using this approach, for example, if the weld cap height is 2.00 mm, and the misalignment is 1.00 mm, then the model suggests a fatigue life of 8.85 years (highlighted cell in the Table).
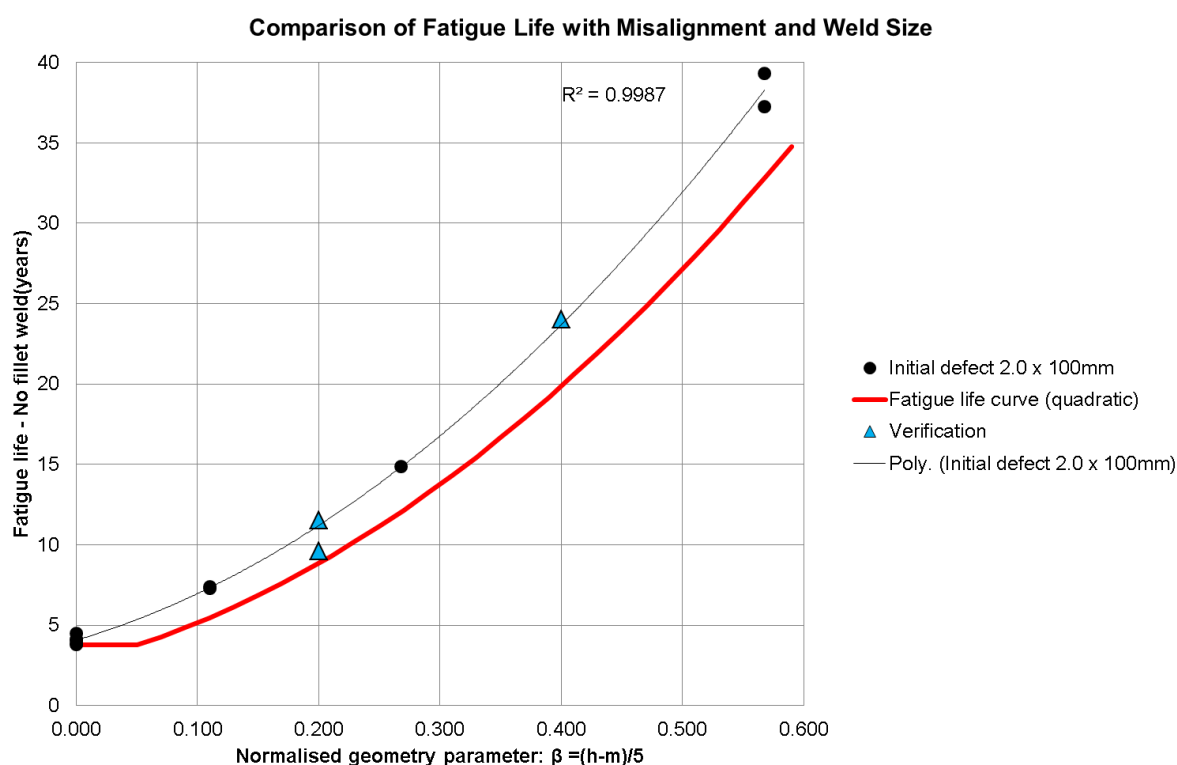
Comparison of Fatigue Life with Misalignment and Weld Size

$R^2 = 0.9987$

- ● Initial defect 2.0 x 100mm
- ▬ Fatigue life curve (quadratic)
- ▲ Verification
- — Poly. (Initial defect 2.0 x 100mm)

Fatigue life - No fillet weld(years)

Normalised geometry parameter: $\beta = (h-m)/5$

Figure 31. Parabolic relationship between fatigue life (each year equivalent to 220,000 km) and non-dimensional geometry parameter β.

Note that the presentation and development of the fatigue life estimation curve is only provided as an **informative example**. That is, it is clear from the results that it is possible to obtain a way of estimating the fatigue life of the joints based upon measurements. The fatigue life estimation look-up tables have been provided as a way to demonstrate a method that would enable field measurements to be easily converted into an estimated fatigue life. However, the final presentation of the table will necessitate agreement on the assumptions used to derive the results, i.e. the initial defect height, defect length and fatigue stresses. Additionally, the feasibility and validity of implementing such a method depends on the ability to measure accurately the weld cap height and misalignment from the external surface of the tanker.

Table 6. Fatigue life calculation look-up table based on the lower bound fatigue life estimation curve ("Not Valid" if m > h)

		Misalignment, m (mm)												
		0.000	0.125	0.250	0.375	0.500	0.625	0.750	0.875	1.000	1.125	1.250	1.375	1.500
Weld cap height, h (mm)	0.000	3.75	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid
	0.125	3.75	3.75	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid
	0.250	3.75	3.75	3.75	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid
	0.375	4.41	3.75	3.75	3.75	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid
	0.500	5.15	4.41	3.75	3.75	3.75	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid
	0.625	5.96	5.15	4.41	3.75	3.75	3.75	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid
	0.750	6.85	5.96	5.15	4.41	3.75	3.75	3.75	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid
	0.875	7.81	6.85	5.96	5.15	4.41	3.75	3.75	3.75	Not Valid	Not Valid	Not Valid	Not Valid	Not Valid
	1.000	8.85	7.81	6.85	5.96	5.15	4.41	3.75	3.75	3.75	Not Valid	Not Valid	Not Valid	Not Valid
	1.125	9.96	8.85	7.81	6.85	5.96	5.15	4.41	3.75	3.75	3.75	Not Valid	Not Valid	Not Valid
	1.250	11.15	9.96	8.85	7.81	6.85	5.96	5.15	4.41	3.75	3.75	3.75	Not Valid	Not Valid
	1.375	12.41	11.15	9.96	8.85	7.81	6.85	5.96	5.15	4.41	3.75	3.75	3.75	Not Valid
	1.500	13.75	12.41	11.15	9.96	8.85	7.81	6.85	5.96	5.15	4.41	3.75	3.75	3.75
	1.625	15.16	13.75	12.41	11.15	9.96	8.85	7.81	6.85	5.96	5.15	4.41	3.75	3.75
	1.750	16.65	15.16	13.75	12.41	11.15	9.96	8.85	7.81	6.85	5.96	5.15	4.41	3.75
	1.875	18.21	16.65	15.16	13.75	12.41	11.15	9.96	8.85	7.81	6.85	5.96	5.15	4.41
	2.000	19.85	18.21	16.65	15.16	13.75	12.41	11.15	9.96	8.85	7.81	6.85	5.96	5.15
	2.125	21.56	19.85	18.21	16.65	15.16	13.75	12.41	11.15	9.96	8.85	7.81	6.85	5.96
	2.250	23.35	21.56	19.85	18.21	16.65	15.16	13.75	12.41	11.15	9.96	8.85	7.81	6.85
	2.375	25.21	23.35	21.56	19.85	18.21	16.65	15.16	13.75	12.41	11.15	9.96	8.85	7.81
	2.500	27.15	25.21	23.35	21.56	19.85	18.21	16.65	15.16	13.75	12.41	11.15	9.96	8.85
	2.625	29.16	27.15	25.21	23.35	21.56	19.85	18.21	16.65	15.16	13.75	12.41	11.15	9.96
	2.750	31.25	29.16	27.15	25.21	23.35	21.56	19.85	18.21	16.65	15.16	13.75	12.41	11.15
	2.875	33.41	31.25	29.16	27.15	25.21	23.35	21.56	19.85	18.21	16.65	15.16	13.75	12.41
	3.000	35.65	33.41	31.25	29.16	27.15	25.21	23.35	21.56	19.85	18.21	16.65	15.16	13.75

The main assumptions in the fatigue life calculations and in the development of the lower bound fatigue life estimation curve are the initial defect size and the use of pure membrane stresses. There are several factors that can result in different fatigue life calculations. For example:

Initial defect height. Starting with a smaller initial defect height will increase the calculated fatigue life, as it will take longer to grow a smaller defect to the critical defect size. Alternatively, increasing the initial defect height will significantly decrease the calculated fatigue life, for the opposite reason. For the present calculations, TWI has chosen to select 2.0 mm as the initial defect height. TWI has observed a 2.19 mm defect, whereas HSE have reported evidence of a 2.4 mm deep defect, and therefore, it is feasible to take a conservative approach and assume that all fatigue life calculations should start with the maximum observed crack height, i.e. an initial defect height of 2.4 mm. However, TWI has also observed a defect with height 1.00 mm. Therefore, potentially the initial crack height could be considered to be less than 2.0 mm.

Initial defect length. The fatigue life calculations are less sensitive to the initial defect length than they are to the initial defect height. However, if the initial length is increased, then the fatigue life will decrease, and if the initial length is decreased, then the fatigue life will increase. TWI has chosen a 2 x 100 mm defect as the initial defect size as it is not unexpected that such a defect is present. HSE have also chosen this as the initial defect size for their fatigue life calculations for similar reasons and based upon destructive examination of sections from the non-compliant tankers.

Degree of bending. The "degree of bending" is defined as the ratio of the bending stress (P_b) to the total stress ($P_b + P_m$) where P_m is the membrane stress. A degree of bending of 0 would indicate pure membrane stress, and a degree of bending of 1 would indicate pure bending stress. It is recognised that the degree of bending varies from band-to-band and that it has been observed that a degree of bending of up to 0.6 can be achieved in a few, short length, fatigue-sensitive locations. A standard conservative assumption is to assume pure membrane fatigue stresses, as has been employed for the fatigue life calculations presented in this report. However, including a suitable, conservative level of degree of bending can increase the fatigue life.

Variable amplitude loading assumption. As noted, BS 7910 recommends that for variable amplitude loading a safety factor of 2 is applied. This is due to the complexities that variable amplitude loading has on fatigue crack growth rates. TWI has not applied this safety factor of 2, based on consideration of the fact that pure membrane stresses have been assumed and the fatigue life estimation curve has been deliberately "offset" to give lower-bound estimates.

In order to assess the fatigue life calculations based on the assumptions previously described, a small sensitivity study has been performed in order to assess the effect of the initial defect height, initial defect length and the degree of bending. In the results that follow, the fatigue life is presented in terms of "years". Note that, by definition, this is a 220,000 km year and may not refer to an actual one-year time period, as this will depend on the tanker operator. The main conclusions from the sensitivity study are that:

- When the initial defect height is decreased to 1.5 mm, and the safety factor of 2 is included, then the fatigue life for the average joint geometry (Case 09) is 18.5 years, which is about 25% longer than the fatigue life for Case 09 with an initial defect height of 2.0 mm and no safety factor.

- When the initial defect height is increased to 2.25 mm, and the safety factor of 2 is included, the fatigue life for Case 09 is 4.8 years, which is about a 70% reduction from Case 09 with an initial defect height of 2.0 mm and no safety factor.

- When the initial defect length is decreased to 75 mm, and the safety factor of 2 is included, the fatigue life is 11.6 years which is about 80% of the calculated fatigue life for Case 09 with an initial defect size of 2 x 100 mm and no safety factor.

- Because the stress intensity factor associated with bending stress is much lower for the non-compliant tanker joint than the corresponding stress intensity factor for membrane stress, when the degree of bending is varied, the fatigue life increases, as expected. When the safety factor of 2 is included and the degree of bending is between 0.25 and 0.5, the fatigue life is between 17.2 years and 53 years. Therefore, it would be possible to adjust the lower bound fatigue life estimation curve by a factor to incorporate expected levels of degree of bending.

4.2.8.7   Macro- and microscopic examination of sections from non-compliant tankers

Part of WP2 was concerned with the examination of actual samples removed from non-compliant tankers to confirm the existence of crack-like, lack of fusion defects; to measure the flaws when present; to observe any evidence of fatigue crack growth; and to provide "post-mortem" examination of sections taken from the topple testing carried out in WP1. In particular, the metallographic examination of sections taken from the tankers provides experimental evidence of the defect sizes used in the engineering critical assessment.

The main findings concern a section taken from the rearmost band of the topple-tested 2008 tanker, from the impacted side. In this sample, the circumferential weld ruptured during the rollover test as a result of a lack-of-fusion defect arising from the positioner lip on the extrusion band. That is, an initial surface-breaking defect tore, in a ductile manner, through-wall thickness, resulting in a through-wall flaw with length 320 mm. Metallographic examination of the sample revealed the presence of a lack of fusion defect that was approximately 1.0 mm deep. In order to undertake a fractographic assessment of the specimen only half of the through-wall flaw was broken open. The fracture surface revealed lack of fusion along the entire length, 230 mm, of the specimen. The other half was not broken open but radiography confirmed lack of fusion was present. Without measuring the actual crack length from the other side of the through-wall defect, it can only be conclusively stated that the initial defect was at least 230 mm long.

In the context of the finite element analysis critical defect size calculations:

- The section from the 2008 tanker that ruptured through-wall had an initial axial misalignment of approximately 0.5 mm, a local weld cap height slightly larger than 1.0 mm, an initial defect height of 1.0 mm, and a length in excess of 230 mm (i.e. essentially a "long" surface flaw).

- The calculations for the "average" joint geometry (Case 09), assumed a local weld cap height of 2.0 mm and axial misalignment of 0.66 mm. For this geometry, under the topple test conditions derived from the FE modelling in WP1,

the ECA performed in WP2 predicted a critical defect height of 1.1 mm when allowance for stable ductile tearing is made.

Therefore, taking into consideration that the Case 09 geometry had a larger weld cap height than the section from the 2008 tanker that failed, it is expected that the critical defect height for the section that failed should be slightly less than 1.1 mm. In this section, a 1.0 mm lack of fusion defect has resulted in failure, therefore this case provides strong support for the accuracy and validity of the critical defect height calculations performed for the topple test conditions.

A macro image of the cross section of the failed joint is shown in Figure 32.
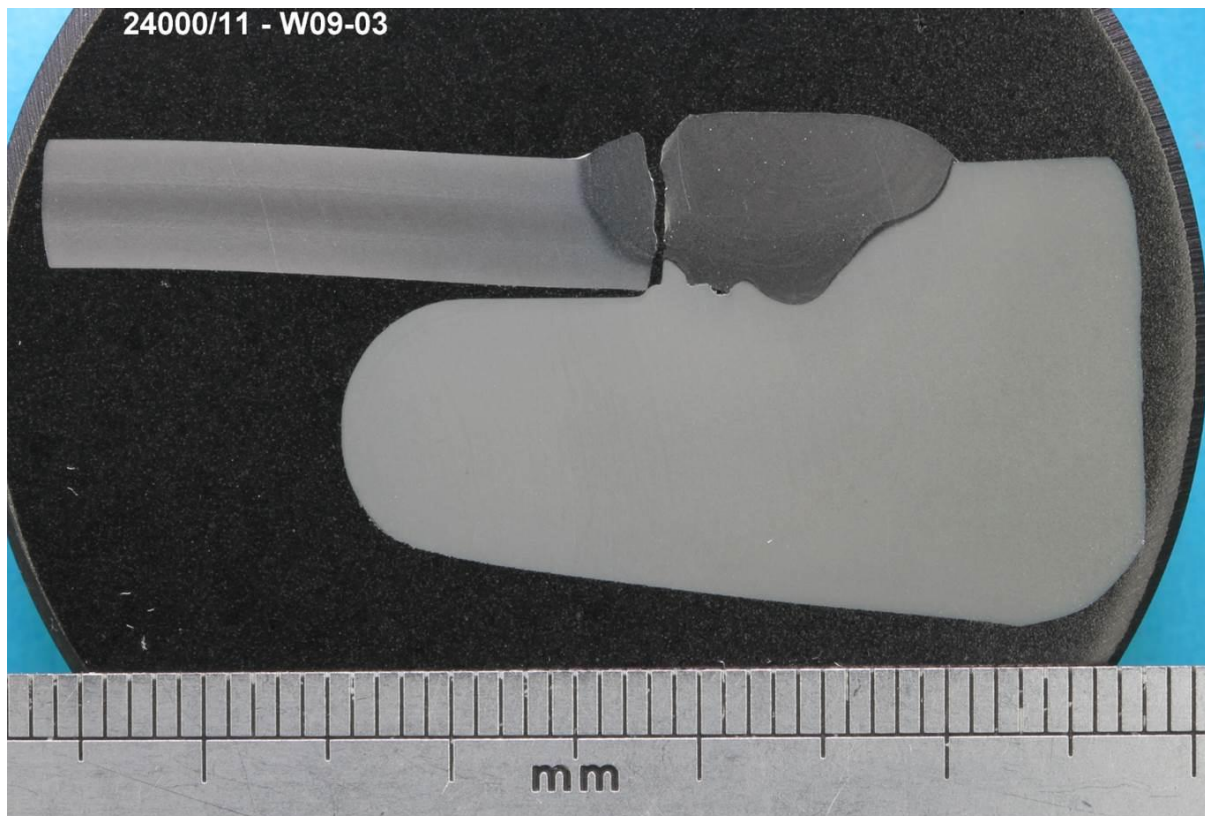
Figure 32. Cross-section of the sample from the 2008 tanker that ruptured through-wall thickness as a result of the lack-of-fusion defect at the positioner lip under the topple test experiment

The metallographic examination of multiple sections removed from the non-compliant tankers also found that:

- Examination of the section from the 2011 tanker revealed only relatively small (total height less than 1.0 mm) lack of side wall fusion, embedded-type defects. This class of defects would not be of as significant concern as the surface-breaking flaws also analysed for this research.

- Examination of sections from a 2009 tanker revealed both a 2.19 mm and 2.04 mm deep surface-breaking defect. These defects were not located directly at the positioner lip but instead a small distance offset. The length of both defects was between 40 mm and 50 mm. Correspondence with the manufacturer has

confirmed that the location of these defects corresponds to the position of an external tack weld and machining that during manufacture of the tankers is typically at most 50 mm long. These measurements agree with this explanation.

- Based on the ECA of the rollover event, it is possible that the finite length defects observed in the 2009 tanker sample could be critical under rollover. However, it is recommended that further analysis is performed to conclusively determine this.

- No evidence of fatigue crack growth was observed. The samples taken from the 2008 and 2011 tankers were removed from the sides of the tanker and therefore the samples were not in particularly fatigue-damaging locations. The samples taken from the 2009 tanker were located along the cradle welds where fatigue crack growth may be expected. Most samples prepared had additional internal fillet welds present and therefore, fatigue crack growth was not expected. However, even for samples without additional internal fillet welds, no fatigue crack growth was observed.

Finally, independent of the assumptions made about the degree of bending of fatigue stresses, the measurements of the joint geometry made during the post-mortem examination can be used to provide indicative information about where typical field measurements of the joints may be located in the look-up table. For each section from the 2009 and 2011 tankers, the weld cap height and misalignment were measured. These values were used to calculate the non-dimensional geometry parameter β for each joint. 60% of the values of β measured exceed 0.35.

Note that in the look-up table, diagonals from top-left to bottom-right correspond to constant β-values (hence the constant fatigue life along diagonals). Assuming pure membrane stresses in the current fatigue life look-up table, a β value of 0.35 would correspond to a 16.65 year life and a value of 0.4 would correspond to a 19.85 year life. Thus, taking into consideration the fatigue calculation sensitivity study where introducing a degree of bending can increase the expected fatigue life significantly, the fatigue life calculations agree with the observation of no fatigue crack growth: since the tankers sectioned and analysed had been on the road for less than 6 years and the anticipated fatigue life could, on average, be greater than 20 years (potentially much greater depending on the degree of bending), then it would not be expected that significant, observable fatigue crack growth would have occurred in the sections analysed.

5 Key findings

This Chapter begins by presenting the key findings from each Work Package, and ends with an overall summary and discussion of the results.

5.1 WP3 – accident data and regulatory implications

WP3 consisted of two tasks (in addition to cross-consortium peer review activities and production of this overall summary report):

Task 1 – Determine representative rollover and collision loads, and;

Task 2 – Identify regulatory implications and potential amendments.

Tank rupture was thought, at the outset of the project, likely to be rare, so the research net was cast wide and involved an international review of multiple sources:

i. Published international research literature (from 1995 – 2014)

ii. DfT statistics/records (including STATS19[13] data, ADR and RIDDOR reports)

iii. Local news media articles

iv. Detailed (in-depth) truck accident databases (RAIDS)[14]

v. Stakeholder surveys (of tanker operators, repairers and international experts)

In total, 116 papers and articles were identified by the literature search. No directly relevant UK published research was found. However, various studies of relevance were identified internationally, in particular from the USA and Germany, but also Spain, Netherlands and China.

From a detailed analysis of the STATS19 data from 2007-2013, indications are that six-axle articulated 'FL vehicles' (tankers above 7.5 tonnes mgw licensed to carry flammable liquids) are involved, on average, in 1.58 injurious collisions per 100 registered vehicles per annum, which is almost 20% lower than the rate of all 6-axle artics in Great Britain. The annual rollover involvement rate of these FL vehicles is 0.11 per 100 registered vehicles per annum, which is 43% lower than the rate of all 6-axle artics.

Taking into account damage-only accidents (based on estimates from German data and GB news reported incidents), it is estimated that there are around 81 to 108 collisions in GB each year involving 6-axle FL vehicles >7.5 tonnes mgw and severe enough to cause an injury or tow-away damage. Further, it is estimated that 5 to 7 of these collisions involve the FL vehicle rolling over.

There were approximately 220 of the non-compliant tankers originally operating on GB roads, out of a total FL vehicle fleet of around 3,400 vehicles registered (on average between 2007 and 2012). Assuming these non-compliant tankers have similar usage characteristics and are thus likely to have similar involvement rates to all FL tankers, expressed in terms of rates per 100 registered vehicles, it can therefore be estimated that these non-compliant tankers were likely to be involved in up to 7 collisions per year, severe enough to cause an injury or tow-away damage. This would include one rollover collision occurring every 2.2 years, on average.

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[13] The database of police-reported road accidents in Great Britain

[14] The Road Accidents In-Depth Studies database

Based on German records on frequency of load spillage, it could be anticipated that there might be up to 5 spillage incidents involving 6-axle FL vehicles >7.5 tonnes mgw per annum in GB. If the likelihood of spillage in a collision was identical, a collision of a non-compliant tanker involving spillage could be expected to occur once every three years.

It is possible that factors such as the low centre of gravity of petroleum tankers (compared to other types of tanker), a greater focus on driving standards and driver safety within the industry, and greater investments in vehicle-based safety technologies amongst the major oil company fleets, all together lead to accident, rollover and spillage frequencies somewhat lower than those experienced by other types of FL-registered articulated vehicles. This hypothesis could not be fully tested during the research. The safety record of petroleum fuel tankers may well be better, on average, than these other tankers, but no amount of safety interventions can be certain to eliminate all risks.

As an absolute minimum, articulated petroleum road fuel tankers have been found to have overturned in 6 separate incidents in the UK over the last four years. If the UK fleet of such vehicles is as high as 1,500, as stakeholders and our own estimates suggest it might be, this implies a rollover frequency of no less than 0.1 per 100 registered vehicles per year (0.1%). This provides a lower bound estimate for the 220 non-compliant tankers of an overturn incident involving them likely once every 4.5 years on average.

An alternative way of expressing this risk of rollover is to consider the probabilities of one or more of the non-compliant tankers overturning in any given period of time. For the 120 non-compliant tankers thought (at the time of drafting this report) to still be in use on UK roads, there is estimated to be a 51% chance of at least one overturning in the next 6 years, and a 66% probability when a period of 9 years is considered.

Some stakeholders consulted during this research have suggested that the rollover risks for the non-compliant tankers might be even lower than the lower bound estimates made above, perhaps by a factor of two. No supporting evidence has been provided, but if the probability of an individual non-compliant tanker overturning in any one year were actually 0.05% (rather than 0.1%), then over a six year period, the probability of at least one rollover incident involving one of the 120 non-compliant tankers would be 30%, and over 9 years it would be 42%. So while the absolute rollover probabilities based on estimates provided by industry stakeholders may be somewhat lower than those indicated by the research, they are still of the same order of magnitude.

In total, 15 officially reported incidents from the period 2005 – 2013 have been identified as being relevant to this study, i.e. involved a flammable liquid tanker. The name of the tank manufacturer is not provided in ADR or RIDDOR incident reports. Eight of the 15 accidents (53%) involved a spilt load, of which 6 cases were major spills, i.e. >1,000 litres. These major spills were associated with overturning in 5 out of 6 cases and a side impact (impact with the jib of a mobile crane) in the other. Three of the 6 cases were with aluminium tanks, the other three were of unknown material. One case involving an overturn but not leading to spillage was with a steel tank.

The analysis of STATS19 data, RIDDOR reports and local news reports on tanker accidents involving spillage of flammable liquids indicates a high probability of quite significant under-reporting of ADR incidents to DfT. The best available estimate is that only around 10% of the incidents that should be reported (albeit based on a strict interpretation of the ADR requirements) are actually reported to DfT; of the 5 to 7 rollover incidents involving FL vehicles likely to be occurring each year in GB, only 0.6 per year are currently being reported as ADR Incidents to DfT. In order to improve the

level of reporting, the following ideas might be considered: Enhanced guidelines and training for operators; a dual reporting system; an ongoing periodic review of local news media articles; and a web-based central data repository. Future research would benefit from additional information on tank, vehicle, and damage in the incident reports.

A review of the RAIDS databases (Road Accident In-Depth Studies), which contains the HVCIS and TCIS detailed truck accident databases, has identified over 80 incidents involving articulated tankers (not restricted to ADR tankers). Rollovers and rear impacts are identified as the main collision mechanisms, but load spillage was rare.

The international literature, statistics, in-depth databases and stakeholder survey responses all point to the importance of rollover as a contributory factor in major fuel spillage incidents. They also suggest that rear impacts with other heavy vehicles are often contributory to more minor spillages.

Rollovers appear to be rare, and historical trends suggest they have become rarer still over recent years, probably as a result of preventative technologies and safer driving interventions. Major spills associated with overturning have also reduced in frequency.

Technologies and driver training cannot, however, prevent all rollovers; such incidents do still happen, even amongst major oil company fleets. For an overturn to result in major spillage, the evidence gathered indicates that a combination of overturning and sliding is usually involved, with rupture of the tank arising from scraping or puncturing impacts with road-side objects and structures as the tanker slides on its side. A simple, low speed overturn by 90 degrees onto a rigid flat surface without significant sliding or other secondary impacts, as used in Work Package 1 (WP1) to validate a mathematical model of the said tankers, appears very unlikely to lead to significant fuel spillage.

No evidence has been found to indicate that failures of circumferential welds have played any significant role in real-world fuel spillage incidents, although none of the non-compliant tankers are known to have been involved in such incidents.

In overturning without sliding, previous testing reported in the international literature suggests roll rates of 100 to 150 deg/s (1.75 to 2.60 radians/s) are likely at the point of impact of the tank with the ground. For the testing and modelling carried out in WP1, a simple tilt and topple test achieving a roll rate at impact within this range would appear, therefore, to be a realistic representation of that scenario.

Possible regulatory enhancements include (tractor unit) fuel tank design/location, tank material specifications to better protect against damage as overturned tankers slide along the ground, and greater impact protection for tanker pipework.

There are important limitations, however, affecting the above analyses. The most important of these is that fuel spillage incidents appear to be highly complex, involving various factors and secondary impacts. This means it is not possible to identify a single "average" accident configuration. This is compounded by the general paucity of detailed information on specific accidents, particularly the exact tank failure mechanisms and/or damage patterns.

Extending the testing and modelling work to cover other relevant impact scenarios that do involve a significant risk of load loss, however, is likely to be much more complex. Three major mechanisms are indicated to interact in quite complex ways to induce such risks; rollover, sliding and tearing. It may well not be feasible to devise a single test that achieves all three, so a step-by-step approach may be appropriate.

A review of existing regulations identified two performance-based test procedures that may at least form starting points for the development of enhanced requirements for road fuel tankers, to further reduce the risks of major spillage in complex, but realistic, rollover events. These are the static rollover test used in UN(ECE) Regulation No. 66 (which is similar to the topple test used in WP1) and the front pillar pendulum impactor test used in UN(ECE) Regulation No. 29. The frontal impact pendulum test procedure within Regulation No. 29 was also assessed but was found not to be readily usable as a rear impact test for tankers.

US research suggests that rollovers can be severe enough to cause the tank to roll by more than ninety degrees before it hits the ground, so a vertical drop test of a tank onto its roof, perhaps from a height in the range 1 – 2 metres, could be considered representative of some real-world scenarios. In this context, a modified version of the static roof strength test in UN(ECE) Regulation No. 29 might also be relevant and would likely be simpler to implement and carry out than a dynamic drop test. The accident data evidence, however, suggests that full overturns by 180 degrees may be very rare in GB.

It should also be emphasised that much of the earlier research reviewed relates to tanks used in other countries, not necessarily complying with ADR, often many years ago, so caution is needed when translating the results into UK tanks in use in 2014.

5.2   WP1 – full scale testing and associated modelling

WP1 consisted of five tasks (in addition to cross-consortium peer review activities):

Task 1. Develop an independent non-proprietary structural hydrodynamic model of the non-compliant tankers, validate this model against the results of tanker tests, and report modelling findings.

Task 2. Design, construct and commission a test rig for tests of tankers, including selecting and procuring suitable instrumentation for data gathering.

Task 3. Undertake tests on tankers, including preparing the tankers, assessing the tanker test method and results, and reporting the findings.

Task 4. Determine suitability of tankers for full scale tests and acquire tankers, as appropriate, in accordance with project objectives as specified by DfT.

Task 5. Capture collision and/or deformation data from relevant impacts, for example by laser scanning, to corroborate the modelling and tanker tests, and reconcile any inconsistencies.

The test outcomes demonstrated that the topple test was a reliable test method providing repeatable test data suitable for validating HSL's Finite Element (FE) modelling. The impact velocities for the tanker tests lay within the range reported (in WP3) for real-world rollovers.

Overall, the finite element model of the topple test with a water load for the non-compliant the 2011 tanker correlated reasonably well with the topple test data, providing good validation of the model.

The highest levels of plastic strain in the finite element model with a water load were observed in the bulkheads, at the top and bottom of the flat generated by the impact. The magnitude of the peak plastic strains was in the order of 0.2 (or 20%), a level at which failure may be expected. It was at the top of this flat where ruptures in the toe of

the weld and within the weld between the extrusion band and the bulkhead occurred during the topple tests on the 2008 and 2011 tankers, respectively.

The circumferential welds of ten 8- and 10-banded non-compliant tankers manufactured between 2007 and 2011 were radiographed (four prior to the project). The results of the radiography both informed the selection of tankers for the topple and road tests and provided information on the condition of the welds in a range of non-compliant tankers manufactured over a five year period. Two 8-banded, 6-compartment tankers, one manufactured in 2008 and the other in 2011, were selected for topple tests. One 10-banded 6-compartment tanker (2010) was selected for road tests to gather fatigue data in WP2. The radiography for the 2011 tanker showed the highest proportion of lack of fusion indications in the welds, whilst the 2010 and 2008 tankers showed the lowest.

The tankers selected for test were all fully ADR inspected and, where necessary, remedial work (except to the circumferential welds) was conducted to ensure that the tankers satisfied the test requirements, and were roadworthy and loadworthy. In addition, the tankers selected for topple test were subject to a second radiography examination, and to internal surveys of the fillet welds. The 2011 tanker was subject to an additional internal survey of circumferential weld misalignment, and an external laser scan survey of the circumferential weld caps.

General design and construction differences between and amongst 8- and 10-banded tankers which were relevant to the research have been established. Specific design and construction differences were found in the extrusion profiles, the bulkhead (or baffle) welding to the extrusion bands and the fillet welds.

HSL developed a topple test with a water load whereby a prepared tanker was tilted under controlled conditions until it became unstable and fell onto its offside under the influence of gravity. The tankers were instrumented with pressure transducers, strain gauges and accelerometers to record data for the impact, logged at 50,000 samples per second. Tests were recorded using various high and standard speed cameras.

The tankers were filled to be at, or very close to, their maximum rated load mass (31,380 kg), which was below their rated volume for fuel. Both were filled with 31,376 litres of water (31,376 kg), with each of their compartments filled to about 70% of its maximum capacity. The ground impact velocities for the tanker tests were between 1.82 and 1.93 rad/s, values which lie within the range of 1.75 rad/s to 2.62 rad/s reported for real-world rollovers. The offside of the tanker impacted uniformly along its length, with less than 7 ms between the impact of the front and rear.

After the test, the offside (impact side) of the tankers exhibited a similar deformation shape with the impact area flattened. The deformation profile was similar along the length of the tankers, with the level of deformation increasing from front to rear. The deformation data, both as a reduction in tanker diameter and as the chord length of the flat section, were similar for each tanker tested. The impact caused a permanent reduction in tanker diameter of approximately 100 mm (at the rear) and 82 mm (front) of the 2008 tanker; and of approximately 107 mm (rear) and 82 mm (front) of the 2011 tanker.

Both tankers ruptured during impact. There was a visible leak from the 2008 tanker between the rear bulkhead and extrusion band at the top of the impact area. Subsequent visual inspection found a rupture within the weld between the rear bulkhead and extrusion band at the top of the impact area, and no visible damage at the bottom

of the impact area. During post-mortem examination, an apparent through-wall crack along the circumferential weld at the top of the impact zone was observed in damage to the older (2008) tanker. Pneumatic pressure tests found that all compartments in this tanker had lost their internal integrity. There was a visible leak from the 2011 tanker between the front bulkhead and the extrusion band at the top of the impact area. Subsequent visual inspection found a rupture in the toe of the weld between the front bulkhead and the extrusion band at the top of the impact area, and a crack in the toe of this weld at the bottom of the impact area. Pneumatic pressure tests of this tanker found that internal integrity had been lost between compartments 1 and 2 and between compartments 4 and 5, while the other bulkheads and compartments had maintained their internal integrity. Physical samples of these damaged areas were taken from both tankers for use in WP2.

In the context of inspecting the damage sustained by the tankers used in the tests, the manufacturer of the non-compliant tankers indicated that the damage around the joints between the extrusion band and the bulkhead/baffles for both tankers is consistent with that seen in real-world rollovers.

The orientation of the bulkhead curvature was found to have a large effect on the bending moments modelled in the tanker shell near to the extrusion bands. In simple terms, the bending moments were higher on the convex side of the bulkheads. Although the resolution of the finite element model was not sufficient to consider the extrusion bands and welds in detail (this detail was considered in WP2), fillet welds were found to affect behaviour near to the extrusion bands.

Impact velocity within the ranges modelled (1.89 to 2.0 rad/s for water, 2.0 to 2.6 rad/s for fuel oil and 2.0 rad/s for petrol) did not have a major influence on the results from the finite element models. The patterns and values of bending moments were very similar for the fuel oil and petrol models, with the exception of the area adjacent to the empty compartment for the fuel oil model (which was not empty in the petrol scenario).

Using the 2.6 rad/s fuel oil model, single values for bending moment (1,460 Nmm/mm) and membrane stress (21.5 MPa) at the front side of the rear extrusion band in compartment 4 were extrapolated from model elements close to the circumferential weld. Extrapolating the same way using the 1.89 rad/s water model gave bending moments up to 1,500 Nmm/mm for the location on the rear circumferential weld of compartment 6 corresponding to the through-wall crack in the topple-tested 2008 tanker. These values were supplied to WP2 for the detailed ECA. The limiting effect of the plastic strains in the shell means that other loading scenarios would be unlikely to give significantly different results.

The fuel oil case modelled the tanker with one compartment empty, as occurs in practice because fuel oil has higher density than petrol. Modelling a tanker with a representative load of fuel oil or petrol, as opposed to water, led to significantly higher predicted deformation at the rear of the tanker. Pressures, stresses and bending moments for the fuel oil and petrol simulations were also higher at the rear of the tanker than was the case with water. The most significant feature of the fuel oil finite element model was the behaviour around the empty compartment, with considerable differences when compared to the water model. This may suggest that the topple test conditions, with the load distributed evenly throughout the tanker, may not be as severe as some real-life events.

5.3 WP2 – detailed Engineering Critical Assessment

WP2 consisted of five tasks (in addition to cross-consortium peer review activities):

Task 1. Determine the typical in-service life cycle fatigue loadings at worst case locations on the circumferential weld seam, including, when appropriate, the effects of filling and dispensing from compartments;

Task 2. Review a proprietary finite element model of the non-compliant tanker as well as the crack growth and leak-before-break and associated technical documents and studies undertaken by its manufacturer and TWI;

Task 3. Address deficiencies as appropriate (such as fracture toughness properties, fatigue crack growth rates and weld residual stresses, wherever possible using strength and fatigue tests of samples taken from tankers to validate the model);

Task 4. Engage with the manufacturer to solicit and incorporate views as appropriate;

Task 5. Undertake a detailed engineering critical assessment to predict crack growth, likely fatigue life of the circumferential welds and defect sizes under suitable loading conditions, incorporating geometric variability;

TWI has critically reviewed a significant number of technical documents concerning the integrity of the non-compliant tanker circumferential welds in the context of current best practice (as dictated by relevant codes and standards). Additional supporting material has been made available to TWI by DfT to complement this exercise where appropriate.

Additionally, TWI has undertaken both laden and corresponding unladen testing of a 2010 tanker whilst instrumented with strain gauges and accelerometers. For each circumferential weld and each instrumented position around the circumference, the strain data has been converted into stresses acting transverse (normal) to the weld seams. The resulting stress time-series were used to calculate the number of cycles per stress range at each location. These stress-range histograms were then used in fatigue crack growth calculations.

Finally, an engineering critical assessment (ECA) has been performed to assess the fracture and fatigue integrity of crack-like defects in the circumferential welds of the non-compliant tankers. The ECA takes into consideration the geometric variability of the tanker band joint; the fatigue stress spectra measured from the full-scale fatigue data collection exercise; likely residual stress profiles as obtained from a thermo-elastic-plastic welding simulation based on the manufacturer's weld procedure specification; material properties obtained through mechanical testing, and information and insight obtained from the review of previous analyses.

A review of the previous analyses reached the following conclusions:

- The methods the manufacturer used to investigate fatigue crack growth are not sufficient to absolutely determine the absence of fatigue cracks in the 2007 tanker assessed.

- The engineering critical assessment performed by the manufacturer concludes that a 2.0 mm deep flaw is acceptable under rollover conditions. Different conclusions have been reached by an HSE study and TWI work. A sensitivity-study is required to determine the margin of acceptability of the 2.0 mm deep flaw considered, due to the evidence provided by HSE that flaws with depths greater than 2.0 mm exist.

- The review, carried out by Prof Issler on behalf of the manufacturer, of the significance of the manufacturer's studies in comparison with the HSE's findings demonstrates that there is a gap between experimental observations and testing and theoretical calculations. TWI agrees with several of Prof Issler's observations about methods to reconcile these differences; however, unlike Prof Issler, from the contrasting evidence presented, TWI cannot draw a definitive conclusion about the acceptability of a 2.0 mm deep flaw.

- The review of TWI work by Prof Issler on behalf of the manufacturer highlights potential sources of over-conservatism in the ECAs conducted by TWI. The assumptions made by TWI were explicitly stated and documented in the TWI reports and scope of work. Within the current work programme, specific tasks have been designed to further study these potentially over-conservative assumptions through experimental testing and advanced numerical modelling.

- The manufacturer has developed a detailed finite element model of a ten-banded tanker and performed a stress analysis of this tanker subjected to various loads to derive inputs for their ECA calculations. Based on a review of the model and the post-processed results, it is recommended that a more consistent stress extraction method is used in order to ensure that the stresses obtained from the model are in line with the guidelines for stress extraction recommended in BS 7608.

All welded structures contain imperfections of some kind. Good workmanship rules typically provide acceptance criteria based on quality control, but a more comprehensive assessment of the non-compliant tanker circumferential welds has been performed using Engineering Critical Assessment (ECA) techniques, to assess their fitness for service. An ECA is an analysis, based on fracture mechanics principles, of whether a given flaw is safe (from brittle fracture, fatigue or plastic collapse) under specified loading conditions.

Detailed examination of sections from the tankers has revealed that lack of fusion indications may correspond to crack-like defects in the circumferential welds, so the ECA was used to identify the maximum acceptable defect size (also known as the critical crack size) under various loading conditions, such as normal operational conditions and during rollover events.

To determine the safe operating life of a component containing a defect, two main factors are required: the primary loads and the fatigue stresses. The primary loads allow for the determination of the critical defect size, and the fatigue stresses (fluctuating stresses due to vibration, road conditions, etc) allow for the determination of how long it will take a sub-critical defect to grow by fatigue and become critical. In the context of the present study, "normal operational conditions" were defined by the fatigue stresses obtained from on-road tests with an instrumented tanker, scaled to represent the 220,000 km annual mileage on a mixture of road types. The primary loads under "normal operational conditions" were defined to be the ADR design load cases, which the tanker should expect to encounter at any point during its operational life. The two rollover loading conditions were based on outputs from the FE model (topple test) described above and, separately, a pressure-impulse simulation.

To improve the accuracy of the analysis, and remove potentially overly-conservative assumptions arising from a more simplistic approach, detailed information on materials properties and the geometry of the tanks and welds under investigation was utilised and analysed using a finite element model of the welded joints. Residual stresses arising

from the welding process itself were also assessed, using state-of-the-art welding simulation software, and factored into the analysis.

The main findings of the ECA of the circumferential welds related to the safe operating life of the tankers are that:

- Provided an initial defect is present, the analysis of fatigue data (for a 2010 ten-banded tanker) identified the cradle positions above the fifth wheel coupling and above the front of the rear longitudinal support members as the most susceptible to fatigue crack growth.

- Under normal operating conditions, the minimum critical defect height is greater than 2.0 mm and may be as large as 4.0 mm or more, depending on the presence of an additional internal fillet weld between the toe of the extrusion band and the inner surface of the tanker shell, the magnitude of misalignment between the shell and extrusion band, and the size of the weld cap.

- Assuming an initial defect size of 2 x 100 mm (i.e. a 2 mm deep by 100 mm long surface-breaking flaw) based on observations that such a flaw would not be unexpected, the fatigue life of the joint (i.e. the time required to grow the 2 x 100 mm defect to a critical size) is greater than 20 years when the said internal fillet weld is present and continuous (or suitable if intermittent).

- When a continuous (or potentially intermittent) internal fillet weld is not present, the fatigue life of the joint is influenced significantly by the misalignment and weld cap geometry. For this case, a parametric study involving over 300 simulations was used to derive a quadratic relationship between the fatigue life (assuming an initial 2 x 100 mm flaw) and a geometry parameter that incorporates the weld cap height and misalignment. This allows a conservative estimate of the fatigue life of a joint (without the said internal fillet weld) to be easily determined from a look-up table (derived from the quadratic relationship) using measurements of misalignment and weld cap height, which can be taken relatively quickly with a profile or laser gauge. A sensitivity study was undertaken to highlight the influence of bending stresses in the fatigue spectrum and initial flaw size assumptions on the calculated fatigue life.

The main findings regarding the ECA of the circumferential welds related to the rollover conditions are that:

- For the rollover case derived from the topple tests and associated FE modelling with fuel oil, and allowing for some ductile tearing to occur, the critical defect height in an "average" weld geometry is 1.1 mm when no internal fillet weld is present. Here the "average" weld geometry relates to measurements from the 2011 tanker and may, therefore, not be truly representative of all non-compliant tanker joints.

- This critical defect height agrees well with the experimental observation of the through-wall rupture of a circumferential weld resulting from a lack of fusion defect 1.0 mm deep and longer than 230 mm in a section taken from the impacted side of the 2008 tanker, paying due regard to the specific geometry of the weld involved. Less severe embedded flaws observed in the 2011 tanker have not been analysed to a similar degree but are considered to be much less problematic. The difference between the tankers is thought to be due to a change

in the welding process during manufacture, which occurred in mid-2010 (identified as part of WP1).

- Considering the rollover load case derived from the pressure-impulse simulation, and allowing for some ductile tearing to occur, the critical defect height in the "average" weld geometry is 2.5 mm when no internal fillet weld is present.

- When a well-made and suitable internal fillet weld is present, the integrity of the tank in a rollover is not governed by the quality of the circumferential weld, but by the strength of the parent metal of the tank shell or other factors such as the bulkhead to extrusion band joint(s), which were seen to fail in the topple tests.

The metallographic examination of multiple sections removed from three non-compliant tankers found that:

- A 320 mm long, through-wall rupture of a circumferential weld was observed in a section of the impacted side from the 2008 tanker. The rupture was due to an initial lack of fusion defect at the positioner lip on the extrusion band. The height of the initial defect was approximately 1.0 mm and it was over 230 mm long.

- Examination of the section from the 2011 tanker revealed only relatively small (total height less than 1.0 mm) lack of side wall fusion, embedded-type defects. This class of defects would not be of as significant concern as the surface-breaking flaws also analysed for this research.

- Examination of sections from a 2009 tanker revealed both a 2.19 mm and 2.04 mm deep surface-breaking defect. These defects were not located directly at the positioner lip but instead a small distance offset. The length of these defects was between 40 mm and 50 mm. Such defects could be critical under rollover conditions if an additional internal fillet weld was not present. However, further analysis would be required to conclusively determine the criticality of these defects.

- No evidence of fatigue crack growth was observed. The samples taken from the 2008 and 2011 tankers were removed from the sides of the tanker and therefore the samples were not in particularly fatigue-damaging locations. The samples taken from the 2009 tanker were located along the cradle welds where fatigue crack growth may be expected. Most samples prepared had additional internal fillet welds present and therefore, fatigue crack growth was not expected. However, even for samples without additional internal fillet welds, no fatigue crack growth was observed.

5.4   Overall summary and conclusions

At the outset of the research, there were estimated to be 220 of the non-compliant tankers on the UK's roads. Feedback from industry indicates that, when operating normally, petroleum road fuel tankers of the kind being assessed are predominantly six-axle articulated vehicles, delivering some 180 tonnes of fuel per day to forecourts, and travelling some 220,000 km each per year in the process. This duty profile is, stakeholders suggest, more intensive than many other types of tankers carrying flammable liquids. Based on official fuel consumption statistics, and feedback from such industry stakeholders, the overall UK road fuel tanker fleet is estimated to be in the range 1,000 – 1,500 vehicles. The 220 non-compliant tankers would thus account for approximately 15 - 20% of the UK fleet.

A wide range of evidence points to the importance of rollover as a contributory factor in major fuel spillage incidents. The evidence also suggests that rear impacts from other heavy vehicles are often a contributory factor in more minor spillages.

Rollovers appear to be rare, and historical trends suggest they have become rarer still over recent years, probably as a result of preventative technologies and safer driving interventions. Major spills associated with overturning have also reduced in frequency.

Technologies and driver training cannot, however, prevent all rollovers, e.g. those resulting from a driver losing concentration or even consciousness, or from sudden, evasive manoeuvre steering inputs. For an overturn to result in major spillage, the evidence indicates that a combination of overturning and sliding is usually involved, with rupture of the tank arising from puncturing impacts with road-side objects or structures.

No evidence has been found to indicate that failures of circumferential welds have played any significant role in real-world fuel spillage incidents, although none of the non-compliant tankers are known to have been involved in such incidents.

Statistics from road accidents involving FL-registered six-axle articulated vehicles in Great Britain[15] were combined with evidence on tow-away, damage-only incident frequencies. If the likelihood of involvement and spillage in a collision was identical to all other FL-registered tankers, on average, a collision of one of the 220 non-compliant tankers involving spillage could be expected to occur once every three years or so, and a rollover collision involving such a tanker could be expected once every 2 - 3 years.

It is possible that factors such as the low centre of gravity of petroleum tankers (compared to other types of tanker), a greater focus on driving standards and safety within the petroleum industry, and greater investments in vehicle safety technologies amongst the major oil company fleets, together lead to accident, rollover and spillage frequencies somewhat lower than those experienced by other types of FL-registered articulated vehicles. This hypothesis could not be fully tested during the research. The safety record of petroleum fuel tankers may well be better, on average, than these other tankers, but no amount of safety interventions can be certain to eliminate all road safety risks. If only those rollover cases known to have involved an articulated petroleum road fuel tanker over recent years are considered, the (lower bound) rollover collision frequency expectation for the non-compliant tankers becomes once every 4 - 5 years on average (and the lower bound risk of rollover per tanker per year is estimated to be 0.1%). In probability terms, if the 120 non-compliant tankers thought (at the time of drafting this report) to still be in use on UK roads are likely to overturn at this same (lower bound) overall average frequency as all petroleum road fuel tankers, then there is estimated to be a 51% chance of at least one overturning in the next 6 years, and a 66% probability when a period of 9 years is considered.

Some stakeholders have suggested that the rollover risks for non-compliant tankers are even lower than these lower bound estimates, perhaps by a factor of two. Using their estimates means, however, that while the absolute rollover probabilities may be somewhat lower than those indicated above, they are of the same order of magnitude (30% and 42% probabilities for the 6 and 9 year periods respectively).

A topple test has been developed and shown to be both repeatable and a reasonable representation of real-world articulated tanker rollover events in terms of the velocity of

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[15] "FL-registered" vehicles are those tractor units licensed to pull semitrailers carrying flammable liquids.

impact of the tank with the ground. Two non-compliant, eight-banded tankers were subjected to this topple test, laden with water uniformly distributed across all compartments to give the same overall mass as if carrying fuel (70% by volume as water is denser than fuel)[16]. The performance of both tankers was similar, despite some differences in their detailed design and construction. These differences include the profile of the extrusion bands that join sections of the shell together, how the bulkheads/baffles have been welded to the extrusion bands, the presence and location of internal fillet welds and the extent of lack of fusion indications in the circumferential welds.

In both tests the tanks leaked from impact damage to an end bulkhead where it joined the extrusion band (which formed the end of the main tank shell). One leaked at the rear bulkhead and the other at the front, both at the top of the impact area. All of the compartments in the 2008 tank lost their internal integrity (although the leaks between them were very slow) and in the 2011 tank, internal integrity was lost between two pairs of compartments but not between all of them. Neither test showed any obvious external visual indication of failure of the circumferential welds resulting in leakage, but on closer inspection, an apparent through-wall crack along the rear circumferential weld at the top of the impact zone of the 2008 tanker was observed. In the context of inspecting the damage sustained by the tankers used in the testing, the manufacturer of the non-compliant tankers indicated that the damage around the joints between the extrusion band and the bulkhead/baffles for both tankers is consistent with that seen in real-world rollovers.

A finite-element (FE) model of the non-compliant tankers used in the topple tests, including both structural and fluid elements, was developed. Overall, the predicted structural deflections, fluid pressures, and significant bending moments and stresses agreed reasonably well with those measured in the tests. Further, the model predicted that the highest strains occurred in the bulkheads near the top and bottom of the deformed impact area on the tanker, at plastic levels which might lead to failure, as was found in the tests. The validated model was also used to assess the likely performance of the non-compliant tankers when loaded with fuel oil, with one compartment empty, and with petrol, in all compartments (both are industry practice). For all cases modelled, impact-related deformation of the bulkheads had a larger effect on the stresses within the tank structure than the effects of fluid pressure. The fuel oil and petrol modelling suggests that fuel loads may lead to more severe effects, such as higher deformations, than for the water loads tested, especially if a compartment is empty.

The highest bending stresses, derived by the fuel oil model, were found to be 254 MPa acting near the circumferential welds, together with much lower estimated membrane stresses. The limiting effect of the plastic deformations in the tank shell means that other loading scenarios would be unlikely to give significantly different results. In contrast, a 2-bar pressure impulse simulation (conventionally used as a design load case for rollovers) generated bending stresses no higher than 150 MPa at the same location.

All welded structures contain imperfections of some kind. Good workmanship rules typically provide acceptance criteria based on quality control, but a more comprehensive assessment of the non-compliant tanker circumferential welds has been performed using Engineering Critical Assessment (ECA) techniques, to assess their fitness for service. An ECA is an analysis, based on fracture mechanics principles, of whether a given flaw in a

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[16] Testing with fuel was deemed infeasible, mainly, but not solely, due to safety and environmental concerns.

given welded joint is safe (from brittle fracture, fatigue or plastic collapse) under specified loading conditions.

Detailed examinations have revealed that lack of fusion indications may correspond to crack-like defects in the circumferential welds, so the ECA was used to identify the maximum acceptable defect size (also known as the critical crack size) under various loading conditions, such as normal operational conditions and during rollover events.

To determine the safe operating life of a component containing a defect, two main factors are required: the primary loads and the fatigue stresses. The primary loads allow for the determination of the critical defect size, and the fatigue stresses (fluctuating stresses due to vibration, road conditions, etc) allow for the determination of how long it will take a sub-critical defect to grow by fatigue and become critical. In the context of the present study, "normal operational conditions" were defined by the fatigue stresses obtained from on-road tests with an instrumented tanker, scaled to represent the 220,000 km annual mileage on a mixture of road types. The primary loads under "normal operational conditions" were defined to be the ADR design load cases, which the tanker should expect to encounter at any point during its operational life. The two rollover loading conditions were based on outputs from the FE model with fuel oil (topple test) and, separately, the 2-bar pressure-impulse simulation described above.

The ECA related to the safe operating life of the circumferential welds found that:

- Provided an initial defect is present, the fatigue data (for a 2010 ten-banded tanker) identified the cradle positions above the fifth wheel coupling and above the front of the rear longitudinal support members as most susceptible to fatigue crack growth.
- Under normal operating conditions, the minimum critical defect height is greater than 2.0 mm and may be as large as 4.0 mm or more, depending on the presence of an additional internal fillet weld between the toe of the extrusion band and the inner surface of the tanker shell, the magnitude of misalignment between the shell and extrusion band, and the size of the weld cap.
- Assuming an initial defect size of 2 x 100 mm (i.e. a 2 mm deep by 100 mm long surface-breaking flaw) based on observations that such a flaw would not be unexpected, the fatigue life of the joint (i.e. the time required to grow the 2 x 100 mm defect to a critical size) is greater than 20 years when the said internal fillet weld is present and continuous (or suitable if intermittent).
- When a continuous (or potentially intermittent) internal fillet weld is not present, the fatigue life of the joint is influenced significantly by the misalignment and weld cap geometry. For this case, a parametric study involving over 300 simulations was used to derive a quadratic relationship between the fatigue life (assuming an initial 2 x 100 mm flaw) and a geometry parameter that incorporates the weld cap height and misalignment. This allows a conservative estimate of the fatigue life of a joint (without the said internal fillet weld) to be easily determined from a look-up table (derived from the quadratic relationship) using measurements of misalignment and weld cap height, which can be taken relatively quickly with a profile or laser gauge.

The ECA of the circumferential welds related to the rollover conditions found that:

- For the rollover case derived from the topple tests and associated FE modelling with fuel oil, and allowing for some ductile tearing to occur, the critical defect height in an "average" weld geometry is 1.1 mm when no internal fillet weld is

present. Here the "average" weld geometry relates to measurements from the 2011 tanker and may, therefore, not be truly representative of all non-compliant tanker joints.

- This critical defect height agrees well with the experimental observation of the through-wall rupture of a circumferential weld resulting from a lack of fusion defect 1.0 mm deep and longer than 230 mm in a section taken from the impacted side of the 2008 tanker, paying due regard to the specific geometry of the weld involved. Less severe embedded flaws observed in the 2011 tanker have not been analysed to a similar degree but are considered to be much less problematic. The difference between the tankers is thought to be due to a change in the welding process during manufacture, which occurred in mid-2010.
- Considering the rollover load case derived from the pressure-impulse simulation, and allowing for some ductile tearing to occur, the critical defect height in the "average" weld geometry is 2.5 mm when no internal fillet weld is present.
- When a well-made and suitable internal fillet weld is present, the integrity of the tank in a rollover is not governed by the quality of the circumferential weld, but by the strength of the parent metal of the tank shell or other factors such as the bulkhead to extrusion band joint(s), which were seen to fail in the topple tests.

No evidence of fatigue crack growth was observed in the samples taken and examined from the non-compliant tankers assessed in this research.

A review of existing regulations identified two performance-based test procedures that may at least form starting points for the development of enhanced requirements for road fuel tankers, to further reduce the risks of major spillage in complex, but realistic, rollover events. These are the static rollover test used in UN(ECE) Regulation No. 66 (which is similar to the topple test used in this research) and the front pillar pendulum impactor test used in UN(ECE) Regulation No. 29.