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DEVELOPMENT
of a TANK CONTAINER
IMPACT TEST STANDARD

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by

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16. Abstract

This project was initiated in response to a need to develop a performance standard for impact testing of tank containers, equivalent to that presently described by CSA.B620 or AAR.600, as a proposed ISO requirement for certification of tank containers for rail transport.

An extensive review of existing publications and standards in the field of rail impact testing was conducted. In addition, rail impact models were developed to simulate rail impacts and delineate significant parameters.

The main conclusions and results of the study are as follows:

- Acceleration measured at the tank container corner castings is the closest representation of the shock input to the system (i.e. independent of container design);
- Within the expected range of system parameters, reduction of the acceleration time-history data to the Shock Response Spectrum represents the best compromise in characterizing the input to the tank container;
- A test method based on acceleration time-history measurements reduced to the shock response spectrum is both feasible and practical; and
- The study indicates the need for accelerometer-based data collection and analysis in order to set test levels which will ensure that tank containers will survive the normal impacts that occur during routine freight yard switching manoeuvres.

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16. Résumé

C'est du besoin de développer une norme de performance des conteneurs-citernes aux essais au choc qu'est né le projet faisant l'objet du présent rapport et, plus particulièrement, l'objectif de mettre au point une norme équivalente à la norme CSA. B620 ou à la norme AAR.600, dans le cadre d'une proposition de l'ISO visant la certification des conteneurs-citernes pour le transport ferroviaire.

Une vaste revue a été effectuée des publications et des normes existantes dans le domaine des essais de tenue au choc des wagons de transport ferroviaire. De plus, les chercheurs ont mis au point des modèles de simulation des impacts de wagons et ils ont cherché à en définir les paramètres significatifs.

L'étude faisant l'objet du rapport a produit les conclusions et les résultats ci-après :

- l'accélération mesurée sur les pièces de coins des conteneurs-citernes est la représentation la plus fidèle de la conséquence d'un choc sur le système mécanique (indépendamment du type de conteneur);
- à l'intérieur de la plage des paramètres du système, la conversion des données d'accélération en fonction du temps, en spectre de réponses au choc, représente le meilleur compromis en matière de caractérisation de la sollicitation d'un conteneur-citerne lors d'un impact;
- une méthode d'essai fondée sur les mesures d'accélération variant en fonction du temps convertie en spectre de réponses au choc est à la fois réalisable et exploitable;
- l'étude démontre la nécessité de collecter des données d'accélération et de les analyser afin d'établir des conditions d'essais garantissant la résistance des conteneurs-citernes aux tamponnements courants dans les opérations de triage.

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EXECUTIVE SUMMARY

This project was initiated in response to a need to develop a performance standard for impact testing of tank containers, equivalent to that presently described by CSA.B620 or AAR.600, as a proposed ISO requirement for certification for rail transport.

The work conducted in the course of this project involved several aspects:

- Collecting information from current literature, relevant standards, and experts in the field of railway transport, tank containers, and testing;
- Consulting with test agencies, tank container manufacturers, and vehicle transportation experts; and
- Creating and testing simplified computer rail impact test models.

An extensive review of existing publications describing shock testing methods, train dynamics, and crash impact testing was conducted. Numerous international railway and tank container standards were procured and reviewed. Test agency authorities and tank container manufacturers in several countries (principally Canada, the United States, France, Germany, and South Africa) were contacted and visited to obtain their opinions and comments regarding existing and proposed rail impact test methods. Finally, rail car and tank container models were developed to simulate rail impacts and delineate significant parameters and physical phenomena.

In accomplishing the program objectives, a performance standard was developed which will ensure that results are repeatable for any given tank configuration when tested at various test facilities using different test apparatus. The performance standard also ensures that the potentially damaging effects of longitudinal rail impacts are correctly measured and characterized. Test requirement levels are not specified, however, since there was no appropriate data from which to develop test levels. Actual testing was not within the scope of this project. The study indicates the need for long-term accelerometer-based data collection and analysis in order to set test levels which will ensure that tank containers will survive the normal impacts that occur during routine freight yard switching manoeuvres.

The main conclusions and results of the study are as follows:

- Acceleration measured at the corner castings is the closest representation of the shock input to the system (i.e. independent of container design);
- Within the expected range of system parameters, reduction of the acceleration time-history data to the Shock Response Spectrum represents the best compromise in characterizing the damage potential without unnecessarily complicating the test procedure;
- A simple scaling factor is presented which allows for testing tank containers that are not loaded to their maximum payload capacity (by weight); and
- A proposed draft standard is presented for longitudinal rail impact testing of tank containers.

SOMMAIRE

Le projet faisant l'objet de ce rapport est né du besoin d'élaborer une norme de performance à l'essai au choc des conteneurs-citernes, dont les exigences seraient équivalentes à celles actuellement définies par les normes CSA.B620 ou AAR 600, ce besoin étant créé par la proposition ISO visant la certification du matériel ferroviaire remorqué.

La recherche menée selon les termes du projet comportait plusieurs aspects :

- collecte d'informations dans des publications courantes, des normes pertinentes et auprès de spécialistes dans les domaines du transport ferroviaire, des conteneurs-citernes et des essais connexes;
- consultation d'organismes d'essais, de fabricants de conteneurs-citernes et d'experts en transport par véhicules;
- création et validation de modèles informatiques simplifiés d'essai au choc des wagons de chemin de fer.

Les chercheurs ont parcouru de manière intensive la documentation existante traitant des méthodes d'essai au choc, de la dynamique des trains et des essais de tenue aux impacts. Un bon nombre de normes internationales concernant le transport ferroviaire en général et plus particulièrement les conteneurs-citernes ont été étudiés. On a communiqué avec des agences d'essais et avec des constructeurs de conteneurs-citernes dans plusieurs pays (notamment au Canada, aux États-Unis, en France, en Allemagne et en Afrique du Sud) et on s'est rendu chez eux afin de connaître leur avis et leurs observations au sujet des méthodes actuelles et proposées en matière d'essai d'impact de wagons de chemin de fer. Enfin, des modèles ont été développés pour la simulation d'impacts de matériel ferroviaire afin de définir des paramètres significatifs et de cerner les phénomènes physiques qui interviennent.

La poursuite des objectifs du programme a conduit à l'élaboration d'une norme qui fera en sorte que l'essai de n'importe quelle configuration de citerne, à des installations différentes et au moyen d'appareils également différents produira toujours des résultats comparables. La norme de performance proposée vise également à assurer que l'on mesure correctement les impacts longitudinaux des wagons et qu'on en établisse les caractéristiques distinctives. Comme on ne dispose pas de données de base appropriées, on s'est abstenu de spécifier des conditions d'essais. Aucun essai comme tel n'était prévu par la recherche. L'étude a montré la nécessité d'analyser des données d'accélération recueillies sur une période assez longue afin de pouvoir définir des conditions d'essais qui garantiront la bonne tenue des conteneurs-citernes lorsqu'ils sont soumis aux impacts normaux causés par le tamponnement des wagons au cours des opérations de triage.

Enfin, l’étude a donné les conclusions et les résultats ci-après :

- les valeurs d’accélération mesurées à l’endroit des pièces de coin des conteneurs-citernes, tous types confondus, constituent la meilleure représentation de la réponse aux chocs;
- à l’intérieur de la plage prévue de paramètres du système, la conversion des données d’accélération en fonction du temps, en spectre de réponses au choc est le meilleur compromis des points de vue de la caractérisation du potentiel de dommages et de la complexité des procédures d’essais;
- l’étude a permis de définir un facteur d’échelle simple applicable à l’essai de conteneurs-citernes qui ne sont pas remplis au maximum de leur capacité (en poids);
- l’étude a débouché sur un projet de norme d’essai d’impact longitudinal des wagons transportant des conteneurs-citernes.

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ABBREVIATIONS AND DEFINITIONS

Buffers
The equipment (attached to each end of European rail cars) which is intended to receive the shocks incidental to train movement.

Corner castings
Those parts which are normally present at each of the eight corners of the tank container which are designed to interface with the corner fittings.

Corner/securement fittings
The positive lock mechanisms (hereafter referred as the corner fittings or corner pegs), used to secure a tank container to the bed of a railway car, which within the context of this standard must meet the requirements of ISO 1161.

Coupling (Railway)
The physical linking of one rail car to another by means of either manually operated or automatic linkages (couplers).

Coupling (Vibrational)
The situation where vibrational modes are not independent of one another but where energy can be transferred from one mode to the other.

CSTT (Centre For Surface Transportation Technology)

Damping Ratio
The ratio of actual damping coefficient to the critical damping coefficient.

Draft Gear
The equipment (typically North American), connecting the coupler at each end of the rail car to the centre sill, which is intended to receive the shocks incidental to train movement and coupling.

End of Car (EOC) Cushioning
A cushioning device added to the end sills of flat cars and freight cars which is intended to attenuate rail impact shock.

Gravitational Units (g’s)
Unit of acceleration equal to approximately 9.81 metres per $(\text{second})^2$.

Hammer Car/Wagon
The device, normally a rail car/wagon, used to either impact directly the test car conveying the tank container under test, or to carry the tank container under test. In the context of this report, the hammer car is the car carrying the tank container.

Hertz (Hz)
Unit of measurement related to frequency which is equal to the number of cycles per second.

IEC (International Electrotechnical Commission)
A world wide federation of national standards institutes which develops technical standards including those pertaining to shock and vibration measurement.

IMDG (International Maritime Dangerous Goods Code)
The United Nations' authorised codes and regulations pertaining to sea transport of dangerous goods.

IMO (International Maritime Organisation)
A body within the UN which monitors and recommends regulations published in the IMDG.

ISO (International Organisation for Standardization)
A world wide federation of national standards institutes which develops international standards through its technical committees.

MAWP (Maximum Allowable Working Pressure)
The maximum working pressure for which the tank container was designed.

MDOF (Multi-Degree-Of-Freedom)
Referring to a model which requires two or more co-ordinates to completely define the position of the system at any instant.

Octave
A doubling of frequency.

Q (Quality Factor)
A measure of the sharpness of resonance of a Single Degree Of Freedom system; it is approximately equal to one-half the reciprocal of the damping ratio

SDOF (Single Degree Of Freedom System)
A system for which only one co-ordinate is required to completely describe the configuration of the system at any instant of time.

SRS (Shock Response Spectrum)
A plot of the maximum response experienced by a Single Degree Of Freedom system, as a function of its own natural frequency, in response to an applied shock.

Tank Container
A container suitable for the carriage of gases, liquids and solid substances which within the context of this standard meets the requirements of ISO 1496-3.

T E S (T E S Limited)

Test Car/Wagon
The device, normally a railway flat car, used to support the tank container under test (hereafter referred as the test car).

2DOF (Two-Degree-Of-Freedom)
Referring to a system which requires exactly two co-ordinates to completely define the position of the system at any instant.

INTRODUCTION

1.1 Scope

This project was initiated in response to a need to develop a performance standard for impact testing of tank containers, equivalent to that presently described by CSA.B620 or AAR.600, as a proposed ISO requirement for certification for rail transport.

This evaluation addresses only longitudinal impact of the tank container configured for rail transport. Modes of failure resulting from high cycle (fatigue) stress of low amplitudes is not addressed in this report.

This project does not attempt to determine or define loading regulations with regard to tank containers, nor does the report address the distribution of flat car types used for conveyance of tank containers.

No tank container testing was actually performed for the purpose of this report. Testing was not part of the scope of the project. Consequently, no attempt was made to define a test level against which test data could be compared. The collection and analysis of test data is reserved for a subsequent phase of the project.

1.2 Historical Background

The rail impact test standards currently being applied throughout several different countries reflect a diversity of test method, test apparatus and test requirements. In addition, the rationale underlying each different standard has been generally either lost or forgotten. Given the global interchange of tank containers from country to country, it is in the best interest of the Canadian public, as well as the world community's, to have a uniform, repeatable, and reproducible longitudinal impact test procedure for tank containers.

1.3 Report Structure

The main body of the report is organized into several sections as follows:

1. **Introduction.** This section provides a general overview of the purpose and objectives of the evaluation study.
2. **Existing Standards.** This section presents the most important similarities and differences between the existing rail impact test standards.
3. **Shock Test Methods.** This section briefly describes the most commonly used forms of shock data presentation and reduction with their respective advantages and disadvantages.

4. **Computer Analysis.** This section presents a description of a number of rail impact test models that were used to analyse the dynamics of test car/tank container interaction during simulated rail impacts.
5. **Consultation - Surveys and Site Visits.** This section presents some of the most important information that was gathered during informal and formal consultation with experts in the field of rail impact testing.
6. **Results.** This section briefly describes the most important results from the evaluation study.
7. **Conclusions** This section lists the main findings of the report.
8. **Recommendations.** This section provides recommendations on the next phase of development of the Rail Impact Test Standard.

2 EXISTING STANDARDS

2.1 Existing Standards and Normative References

A number of standards and references were reviewed in the preparation of the Draft Standard. References used for background knowledge, or of a general nature, may be found in the Bibliography section. Standards and references which had a direct bearing on the development of the Draft Standard, or which are referenced in the text, are as follows:

Shock/Impact Testing:

1. ISO 6487:1987, Road vehicles - Measurement techniques in impact tests - Instrumentation;
2. IEC 68-2-27:1987, Basic environmental testing procedures Part 2: Shock;
3. SAE J211:1988, Instrumentation For Impact;
4. MIL-STD-810D:1983, Environmental Test Methods and Engineering Guidelines; and
5. MIL-STD-810E:1989, Environmental Test Methods and Engineering Guidelines.

Freight Containers/Tank Containers:

1. ISO 668:1988, Series 1 freight containers - Classification, dimensions and ratings;
2. ISO 1161:1984, Series 1 freight containers - Corner fittings - Specification; and
3. ISO 1496-3:1995, Series 1 freight containers - Specification and testing - Part 3: Tank containers for liquids, gases and pressurised dry bulk.

Rail Impact Test Standards:

1. AAR.600:1990, Specifications For Acceptability Of Tank Containers;
2. CSA.B620-1987, Highway Tanks and Portable Tanks for the Transportation of Dangerous Goods (Appendix A);
3. EDC/TES/023/000/1991-07, Testing of ISO Tank Containers;
4. UIC 592-4:1985, Swap Bodies Which Can Be Handled By Grabs, Technical Conditions; and
5. CNEST 001:1996, Résistance aux effets de l'inertie longitudinale - Méthode d'essai dynamique.

2.2 Rail Impact Test Standards

Existing rail impact test standards/procedures from several different countries were collected and reviewed. The distinguishing characteristics of these standards are presented in Table 1 for comparison.

Table 1 Rail Impact Test Standards Currently In Use

COUNTRY	TEST STANDARD	MAXIMUM TEST LEVEL	MEASURING POINT	MEASURING DEVICE	TEST DATA FORMAT	TEST CAR	TEST CONTAINER
Canada	CSA.B620 Proc. A4.2	8mph: forward and rearward speed of test car into buffer cars	not specified	radar equipment or electric timers	not applicable	flat car w/std draft gear and underframe - set for coupling	"loaded"
	CSA.B620 Proc. A4.3	force equal to 4 times weight of loaded tank container	between corner castings and test car	load cells	peak force	as for A4.2 but with brakes set - coupling not specified	"loaded"
United States	AAR.600	force equal to 4 times weight of loaded tank container	between corner castings and test car	load cells	peak force	free standing w/AAR approved draft gear	"loaded to rated capacity"
France	CNEST001	max. speed determined by test std being applied	bottom four corner castings	acc.	peak acc. 16 Hz upper frequency limit (fc)	free standing	loaded to 97% vol. capacity w/H_2O
Germany	UIC 592-4	2g-3g	bottom corner castings	acc.	peak acc. 16 Hz (fc)	free standing	loaded with H_2O
South Africa	EDC/TES/023/000 Test 1	8mph-forward	reflectors	IR transducers	N/A	8mph into bufferwagon	loaded to rated capacity
	EDC/TES/023/000 Test 2	8mph -rearward	reflectors	IR transducers	N/A	coupled to bufferwagon	loaded to rated capacity
	EDC/TES/023/000 Test 3	4g	corner castings	acc.	peak acc. 40 Hz (fc)	free-standing	loaded to rated capacity

As evident from Table 1, there are a number of significant differences among the current existing standards/procedures for rail impact testing. The similarities and differences are discussed in Sections 2.2.1 through 2.2.6.

2.2.1 Scope of Existing Rail Impact Standards

Although not noted in Table 1, the existing rail impact test standards vary somewhat in scope. For the purpose of this project, and the proposed Draft Standard, the longitudinal rail impact test is intended to test those portable tank containers as defined in ISO 1496-3, Table 1, namely IMO/IMDG Type 1, 2, and 5, which are suitable for the carriage of gases, liquids and solid substances (dry bulk) by rail conveyance.

2.2.2 Instrumentation

Current rail impact test procedures can be broadly divided into three (3) categories:

1. Procedures which measure the impact severity in terms of the reaction forces at the point of attachment of the tank container to the test car;
2. Procedures which measure the impact severity in terms of acceleration of the tank container corner castings; and
3. Procedures which use velocity at time of impact in conjunction with prescribed test apparatus to characterize the severity of impact.

It is important to note that some test agencies use accelerometers in their current test procedures whereas load cells are specified in the standard. As shown in Table 1, the only test agency responding to the survey which reported using load cells as required by the CSA.B620 and AAR.600 standards is CSTT. Test agencies in France, Germany, and South Africa are currently conducting rail impact tests in which accelerometers are used to measure and characterize the shock severity. EDC, in South Africa, also uses velocity measurement when they are conducting the CSA.B620 (procedure A4.2) test.

Load Cells:
The use of load cells in the impact test requires alteration of the normal (in-service) configuration of the tank container. Since, in theory, the load cell measures the force of the tank container reacting against the corner peg (fittings) of the test car, the load cell must be inserted in series between the tank container corner casting and the test car corner peg. Due to the physical size of load cells, this is difficult to implement in practice. One test agency has managed this problem by dispensing with the corner peg altogether and using a special adapter bracket to mount the load cell between the test car and the tank container. In this arrangement, the load cell is constrained at the test car end by a rigid stop. The tank container is then allowed during the impact to move relative to the test car on a slider mechanism. Lead pellets at the ends of the load cell distribute the load of impact over the ends of the load cell; thus helping to prevent premature failure of the load cells.

There are several possible objections to this configuration. Firstly, the normal (in-service) configuration of the tank container is subtly altered. The tank container, instead of having all six (6) degrees of freedom constrained at each corner now has zero constraints at the rear (opposite impact end) corners and at most two (2) constraints at the front corners. It could be argued that in actual service, there will rarely be full (6x4=24 degrees-of-freedom constrained) constraint due to slack in the fittings and hence the test configuration is not as radically different as suggested. However, there is no question that by not attaching the tank container to the test car, the tank container under test is spared the rebound jolt. In other words, the tank container can be expected to make only one impact of the load cell.

Secondly, the use of a sliding mechanism to control the movement of the tank container, and the use of lead end caps to distribute the load over the load cell both introduce artificial (i.e. not normally found in service) elements of coulomb and viscous damping which are difficult to quantify.

Another objection to load cells, irrespective of the system dynamic issues, is that they are not easily calibrated for dynamic use; there is presently no known satisfactory method for calibrating load cells dynamically (Ref.1). In general, however, the same signal conditioning and measuring instrumentation may be used for dynamic force transducers as for accelerometers (Ref.2).

CSA.B620 and AAR.600 are currently the only existing test standards which specify the use of load cells. Although CNEST.001 and EDC/TES/023/000 (EDC) both purport to meet the requirements of CSA.B620 and/or AAR.600, these procedures do not specify the measurement of force with load cells; acceleration at the lower corner castings is specified instead. A key question with regard to the study of the existing standards is whether the peak force, measured at the corner castings with load cells, is numerically equivalent to the product of the mass of the tank container times the peak acceleration measured at the corner castings.

Accelerometers:

In contrast to load cells, accelerometers can be easily fitted to either the test car or tank container with no alteration of the natural mounting configuration of the tank container with respect to the test car. Figure 1 shows a typical installation of an accelerometer mounted to the leading corner casting of a tank container. An adapter plate has been made by the test agency which allows for quick and easy installation of the accelerometer. There is no practical reason why accelerometers similar to the one shown in Figure 1 could not be fitted to an actual in-service tank car (perhaps for the purpose of measuring in-service shock).

Contrary to load cells, accelerometers are easily calibrated by known methods to known standards. For example, test standards which specify accelerometers in their procedures may reference standard IEC 6487; the test agency performing the actual test is then required to ensure their accelerometers conform to this standard.

The major difficulty that has arisen in the use of accelerometers for rail impact testing has been in trying to determine an appropriate filter cut-off frequency. The problem is the peak value acceleration used to determine the test velocity is very dependent upon the filter frequency, and to a lesser extent, the filter characteristics. Two of the three test agencies which reported using peak acceleration as a test criterion use 16 Hz (Hz) as the cut-off frequency while the third uses 40 Hz. The arguments supporting the use of either 16 or 40 Hz have been generally heuristic in nature with no corroboration from a knowledge of the structural frequencies involved. It should be noted that standards SAEJ211 and IEC 6487 both provide guidance on the selection of filter cut-off frequencies; however, the lowest value suggested by both these standards (for whole body movement) is 100 Hz.. As discussed in Section 3 there are alternative methods of characterizing shock impacts which are not as sensitive to the signal conditioning equipment and practices.

The test agencies conducting the impact tests were contacted and requested to fill out questionnaires providing explanations and/or comments regarding the use of load cells or accelerometers. The survey responses are presented in the Appendix A of this report.

2.2.3 Test Apparatus

Test Car:
The current rail impact test procedures also vary with respect to the test apparatus. Depending upon the procedure being conducted, the test car braking system (just prior to impact) and the coupling mechanism may be configured as:

1. Stationary with brakes off, couplers not specified (AAR.600, EDC);
2. Stationary with brakes off, couplers not applicable (CNEST.001, UIC 592-4); or
3. Stationary with brakes on, couplers set for automatic coupling (CSA.B620 Proc.A4.3).

Note that couplers on European rail stock are manual and therefore “coupling” is not automatic. An example of European style “buffers” and couplers is shown in Figure 2. Draft gear also varies widely in performance characteristics; however the influence of the draft gear or “buffers” will depend on how the test procedure is conducted and the manner in which the test level is specified. Figure 3 shows an example of the North American style coupler system and Figure 4 shows the flat car used by CSTT for tank container impact testing.

Figure 1. Accelerometer Mounting Location On Corner Casting

Figure 2. Buffers (Draft gear) and Couplers - European Style Equipment

Figure 3. Couplers - North American Style Equipment

Figure 4.   Tank Car and 55 ft Flat Car at CSTT's Impact Test Track

Tank Container:
The manner in which the tank container under test is prepared prior to test is also quite important. Whether the tank container is fully loaded to its gross maximum weight or not, will have a direct bearing on the motion of the tank container and hence on the stress levels within the constituent parts. The following points of interest are noted with respect to the current impact standards/procedures:

1. None of the current standards/procedures explicitly require the tank container to be pressurised prior to test;
2. CSA.B620 requires the tank container to be "loaded";
3. AAR.600 and EDC require the tank container to be "loaded to rated capacity" with unspecified material; and
4. UIC 592-4, and CNEST001 specifically require the tank container to be loaded to volumetric capacity with H_2O.

In practice, the test material (as-tested payload) is generally always water. Therefore, the discrepancy between the as-tested payload mass and the rated payload mass can be significant, particularly for small volume containers with high rated payload capacity. This issue is discussed in more detail in Section 4.2.4.

With regard to pressurization of the tank container, at present it does not appear that any test agencies conducting impact tests require the container contents to be pressurized, despite the fact that IMO Type 1 and 5 containers can be used to convey payloads under relatively high pressure (as reference, Table 1 in ISO 1496-III lists the ISO type code for IMO/IMDG tank types against the minimum test pressure). Taking for example, an IMO Type 1 container with a Maximum Allowable Working Pressure (MAWP) of 4 bar, the resultant force on the end caps of the container due to the MAWP could be in the order of 100 Meganewtons (MN). Conversely, the inertial force on the end caps resulting from the fluid contents accelerating at a constant 100 metres/second^2 would be only 2.8 MN (assuming 28,000 litres of water as payload). The actual resulting stresses due to the combined loading will vary on a case-to-case basis. However, as a conservative measure, it would be prudent to impact the tank container whilst it is pressurized.

The configuration of the tank container mounting may also vary between:

1. Being restrained at the front castings only with no horizontal restraint at the opposite end (CSA.B620 as applied by CSTT);
2. Number of restraints unspecified (AAR.600); or
3. Being restrained at all four lower corner castings (CNEST001, UIC 592-4, and EDC as applied by the relevant test agencies).

2.2.4 Test Procedure

The current rail impact test procedures also vary with respect to the manner in which the shock input is imparted to the tank container. For a performance standard, which generally applies for the majority of the test procedures (CSA.B620 Proc.A4.2, EDC

Tests 1 and 2 are exceptions in that the test level is directly dependent upon the configuration of the test apparatus), it should not matter how the shock input is imparted to the tank container if the “input” meets the defined requirements and those requirements are consistent throughout the various test standards/procedures. Unfortunately, this is not necessarily the case as is discussed in Section 2.2.5.

2.2.5 Test Levels

Test levels are difficult to compare amongst the various test standards/procedures due to the current differences in measuring and characterizing the shock input. For example:

1. Peak acceleration recorded after filtering will be strongly dependent on the upper frequency cut-off limit and to a lesser extent the characteristic (roll-off rate) of the filter;
2. Peak acceleration recorded at the corner castings and peak force (measured between the corner castings and the test car) divided by the tank container mass will not necessarily be equal; and
3. Peak acceleration recorded at the corner castings from an impact involving an under-weight payload (with respect to the rated payload) will in general be higher than an acceleration involving a payload at the rated weight.

The issue described in point 2 is discussed in more detail in Section 4 in connection with the modelling results. With respect to point 3, not all the current standards/practices address the discrepancy between the as-tested and rated payload mass. Some procedures specify a “correction” factor for calculating an adjusted impact velocity to compensate for Test Masses which are below the Rated Mass value. This correction factor is discussed in more detail in Section 4.

3. SHOCK TEST METHODS

3.1 Introduction

Several methods can be used to prescribe or characterize a shock event. Each of these, from a purely mathematical viewpoint, could be used in the performance of a rail impact test. For example, a shock test machine, which may be described in more or less elaborate detail, could give repeatable and reproducible results. Normally, however, the use of such a machine is limited to test items having small mass. Although a dedicated test fixture could be built to the scale required for rail impact testing, it would have to be carefully calibrated on a regular basis.

The sheer size and complexity of such a machine could very well place it out of the reach of the transport organizations and manufacturers that require it. Moreover, the standardized jig that would be required for repeatable testing would remove the tank container from the real world of the shunting yard, where concurrent tests could be made. If this is added to the necessity for frequent calibration, then it becomes apparent that a more suitable method of characterizing the shock applied to the tank container is required, rather than a more elaborate test apparatus.

The first step in obtaining such a method is to understand dynamically what happens to the flat car and the tank container from the moment of impact with other rolling stock until all movement has settled to a steady state. This will enable us to define the measurement points, and to select the most suitable instrumentation, and the appropriate mathematical tools to describe the expected results.

Initially, the tank container is at rest relative to the flat car, and both are moving down the track towards other rolling stock. Alternatively, they could be stationary and be approached by a shunting engine or other cars. Each comprises a system of mechanical components and energy is exchanged between the systems on impact. The result is a change in velocity of both systems, with respect to the track.

The individual components of both systems have the mechanical characteristics of mass, stiffness, and dissipative resistance (damping factor), which may be internal, as a characteristic of the material used in construction, or evident as friction between moving surfaces. The impact causes each component to vibrate at its natural frequencies, until the energy is dissipated in the resistive component. This is not the entire story, since the velocity change is initially mediated by the compression of the coupling. The energy is then returned to the flat car/tank container system, minus the energy dissipated in the coupling and minus the energy transferred to the rest of the components. In this way, the shock due to impact radiates outward from the coupling, and is reflected back from the boundaries of the system, which results in a very complex shock pulse throughout the entire structure. Since the result of the impact is a change of velocity over time, instead of an instantaneous change, it is appropriate to use acceleration as the quantity to be measured.

From this description, it is reasonable to assume that the acceleration at the tank container is different from that at the point of impact. This is the point at issue when conducting a shock test with rolling stock from different sources, and with tanks containing different loads. It is impractical to define everything from the impact velocity to the make of flat car in order to standardize the shock transferred to the tank container. The solution suggests itself, however: *measure the acceleration at the point of attachment.* This should eliminate the effect of the rolling stock. In practice, the result of the impact is an uneven combination of vibrations of various frequencies and amplitudes, some of which build up well after the initial impact, before eventually dying out.

Measurements are made using an accelerometer mounted on the base frame of the tank container at the point of attachment. The output for a sudden change between steady velocities is a pulse, i.e. the step transition is differentiated. However, within the mechanical and electrical limits of the accelerometer, changes in velocity (acceleration) result in a steady deformation of the sensor and a constant voltage is produced. Since the seismic mass mirrors the original steady state of the test bed, the movement at the base of the tank container is recorded in absolute terms and is independent of the movement of the test bed.

The most commonly used methods of shock data representation can therefore be classified as either non-spectral (time domain) or spectral (frequency domain).

3.2 Non-spectral Representation

Single Number:

One of the most elementary methods of representing shock data is to record and display single values determined from the time-history record, such as the highest positive or negative value, peak-to-peak value, duration, etc. This is in fact the method specified in all the existing rail impact test standards, which specify either peak force or peak acceleration. Normally, however, for such a representation to be valid, the time-histories should be of approximately the same wave shape.

In certain applications, single figures of merit may be valid. For example, in a system responding "statically" to the input, the input pulse period would be significantly longer (2-3 times or more) than the period of lowest natural frequency. The peak value of the input pulse may then be sufficient to characterize the shock adequately.

In other applications, the derived velocity change of the input excitation may be sufficient to characterize the shock. For example, systems isolated from the input excitation by shock mounts will exhibit natural frequencies having long periods compared to the input duration. In such cases, the acceleration pulse is integrated and a step velocity change becomes a reasonable approximation to the original input.

The use of either representation presumes prior knowledge of the system characteristics. In some instances both representations may be used to completely characterize an item's sensitivity to damage. For example, standard ASTM D3332 specifies the method to test the fragility of a packaged item when subjected to both peak acceleration (while the velocity change is held to a minimum) and velocity change (peak acceleration held to a minimum). The limits set by both criteria then establish a "damage boundary" which identifies the test item's fragility to shock. In general, ASTM D3332 is intended for small size (relative to tank containers) items where the destruction or loss of several test items can be tolerated.

3.3 Shock Response Spectrum - Theory

Original time-history traces of shock inputs, or the response to shock inputs, can be extremely difficult to compare and analyze due to the response of the structures to vibration. Such vibration can produce an extremely complex time history waveform which tends to obscure the more meaningful information contained in the shock measurement (Ref.3). Often, a more concise method of describing the effect of the time history waveform, rather than the waveform itself, is what is actually desired. Consequently, the original time-history measurement is often reduced to a spectral (frequency) representation, which is more suitable for engineering use. The most common form of spectral representation used in specifying shock tests is the shock response spectrum.

The shock response spectrum can be considered as the maximum response of a set of linear Single Degree of Freedom (SDOF) oscillators, each having a different natural frequency, whereby the responses (of the oscillators) are plotted as a function of frequency.

In physical terms, the peak response of each SDOF oscillator to an arbitrary input may be obtained by attaching some form of stylus to the responding mass. A device consisting of an array of such oscillators, known as a "reed gage", actually exists and has been used for quantifying shocks for several decades.

In mathematical terms, the peak response of each SDOF oscillator can be computed using the superposition (convolution or Duhamel) integral, whereby the effect of simultaneously super-imposed actions is equal to the sum of the effects of each individual action. This principle is of great benefit in calculating the shock response spectrum as will be shown in the following example (adapted from Application of B&K Equipment to Frequency Analysis by R.B. Randall).

Consider an arbitrary input to a physical system as shown in Figure 5a; note that the physical system in question could be a SDOF oscillator. The input function, f(t), can be considered to be comprised of a number of contiguous impulses (delta functions) each weighted, or scaled, by the value of f(t) at that point.

In general, each physical system (e.g. a SDOF oscillator) will have a characteristic impulse response such as h(t), shown in Figure 5b. Each impulse contained in the input function, f(t), produces an impulse response which begins at a point in time coincident with the time origin of the impulse. Furthermore, every impulse response will be scaled in proportion to the value of f(t).

For example, in Figure 5c the impulse response resulting from the impulse $f(t_n)$ is shown as being delayed by an amount equal to t_n. The peak value from this single impulse is $f(t_n)*h(t_p-t_n)$. Another impulse response, the one resulting from the impulse just preceding impulse $f(t_n)$, is also shown in Figure 5c. Although only two impulse responses are shown in Figure 5c, for clarity, it is evident that there will be an impulse response corresponding to every impulse shown in Figure 5a. In the limit, as Δt approaches zero, there will be an infinite number of impulses and an infinite number of impulse responses.

By the principle of superposition, the output signal at time t_p, $g(t_p)$, will consist of the sum (or superposition) of all the suitably scaled and time-delayed impulse responses (as shown in Figure 5d). In mathematical terms:

$$g(t_p) = \sum_{n=-\infty}^{\infty} f(t_n)h(t_p - t_n)$$

The general response, at any time t, can similarly be expressed as:

$$g(t) = \sum_{n=-\infty}^{\infty} f(t_n)h(t - t_n)$$

In the limit, as Δt approaches zero, the output response can be expressed as:

$$g(t) = \int_{\infty}^{\infty} f(\tau)h(t-\tau)d\tau$$

To calculate the shock response spectrum, it then becomes merely necessary to substitute for f(t) and h(t), the actual input and characteristic impulse response functions, respectively. In most cases of interest, the input function will be the time varying motion of the support base or foundation. The characteristic impulse response can be derived from an examination of the SDOF oscillator model, as explained in the following paragraphs.

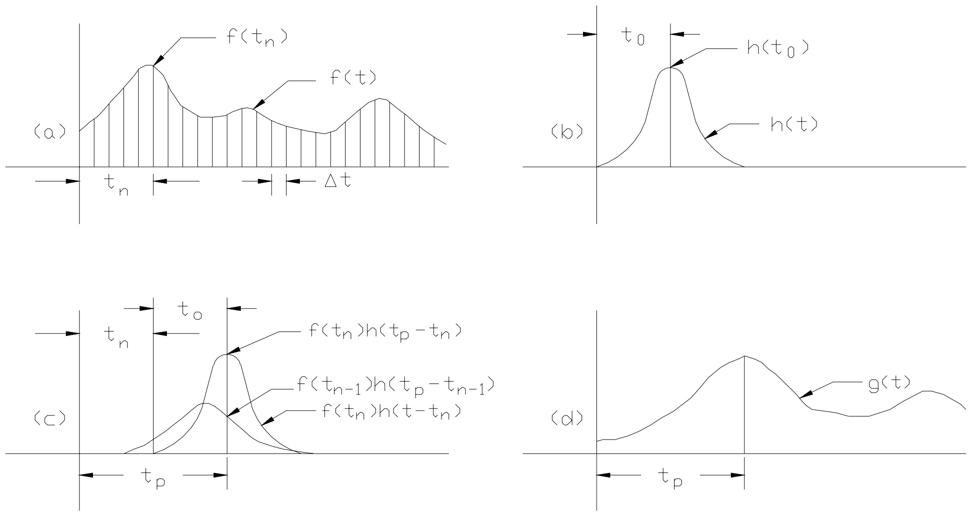


Figure 5. Convolution of an Input Function with an Impulse Response Function

The shock response spectrum is used to characterize the response of a arbitrarily defined series of oscillators to a vibration (acceleration) input to a system undergoing shock testing. In order to simplify the model or representation of the structure or component under test, a SDOF mass-spring-damper model (as shown in Figure 6) is used. This simplified model allows the input function itself to be characterized, without regard to a specific test apparatus or vibration signal. Any number of different inputs can produce a similar shock response spectrum.

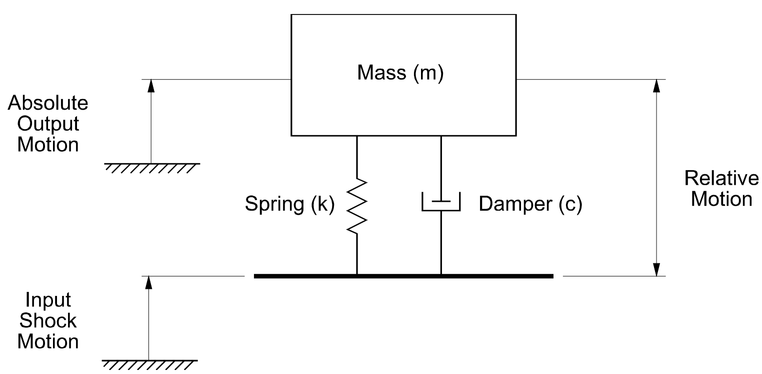


Figure 6. SDOF System Model for Shock Response Spectrum

In the above diagram, the mass, m is connected to a spring of stiffness, k and a damper having mechanical resistance, c.

For a given excitation, the maximum excursion of the elements (condition of maximum velocity) occurs at the frequency for which the mechanical impedance is at a minimum.

The limiting case where the damping, c is zero yields the undamped, ω_n :

$$\omega_n = \sqrt{k/m}$$

In the practical case, where damping is present, it is convenient to assume a force F(t) and to write the equation of motion, for displacement, x:

$$F(t) \quad m\ddot{x} \quad c\dot{x} \quad kx$$

The damping in the system can be expressed in terms of the critical damping by a simple number, called the *damping ratio* (ζ):

$$\zeta = c / c_c$$

By expressing the characteristic equation in terms of ζ, and then substituting the result into the general equation, the result gives the frequency of damped oscillation:

$$\omega_d = \omega_n \sqrt{1 - \zeta^2}$$

Shocks consist of a complex aggregate of individual pulses of various widths and strengths, and except under laboratory conditions, it is unlikely that any two shocks will be identical. If the overall system is linear, the mass deflection in response to an arbitrary excitation becomes:

$$x(t) = \int_0^t f(\xi) h(t - \xi) d\xi$$

Where f(ξ) is the input excitation force and h(t-ξ) is the response to a unit impulse started at t = ξ.

For the single degree of freedom system, substitution of the unit impulse, gives the peak deflection:

$$x(t)_{\max} = \left| \frac{1}{m\omega_n} \int_0^t f(\xi) \sin \omega_n (t - \xi) d\xi \right|_{\max}$$

The shock response is not simply a complex series of vibrations that die away to zero in a neat logarithmic curve. It should be visualized as a three dimensional event, having the dimensions of displacement, frequency and time. Each frequency has a unique decay characteristic. Thus, the displacement versus frequency envelope not only decreases in overall amplitude as a function of time, but it changes shape as the various resonances come into play. Similarly, the displacement versus time envelope changes shape continuously.

Since force is a function of mass and acceleration, it is reasonable to use acceleration as an indicator of force. If the damping ratio is known is known, then the applied force can be calculated.

An intuitive grasp of what is happening can be had be imagining the sound of a grand piano that has been dropped by careless movers. All the strings are set into motion, but not at the same time, since the shock has to travel through the framework. The movement of each string decays at a different rate, so that over the time from the initial impact until all sound ceases, the timbre of the sound changes continuously.

Conceivably, the movement of each string could be monitored and the acceleration at maximum deflection could be calculated. This would generate a series of shock response curves from which the maximum acceleration versus frequency could be plotted. Although a recording of the complete event could be used to recreate the movement of each string, the movement of the strings used to characterize the magnitude of the shock of impact could not be used to recreate the event. However, the in the case of the tank container, the shock is applied at the same point for each test. Thus, the only parameter that is required is the maximum potential severity of the applied force. Since this is the product of mass and acceleration, the frequency term having been eliminated during the process of differentiation, the actual frequency-time envelope becomes relatively unimportant.

The shock response spectrum (SRS) is a plot of the maximum acceleration at each predetermined natural frequency of a SDOF mass-spring-damper system.

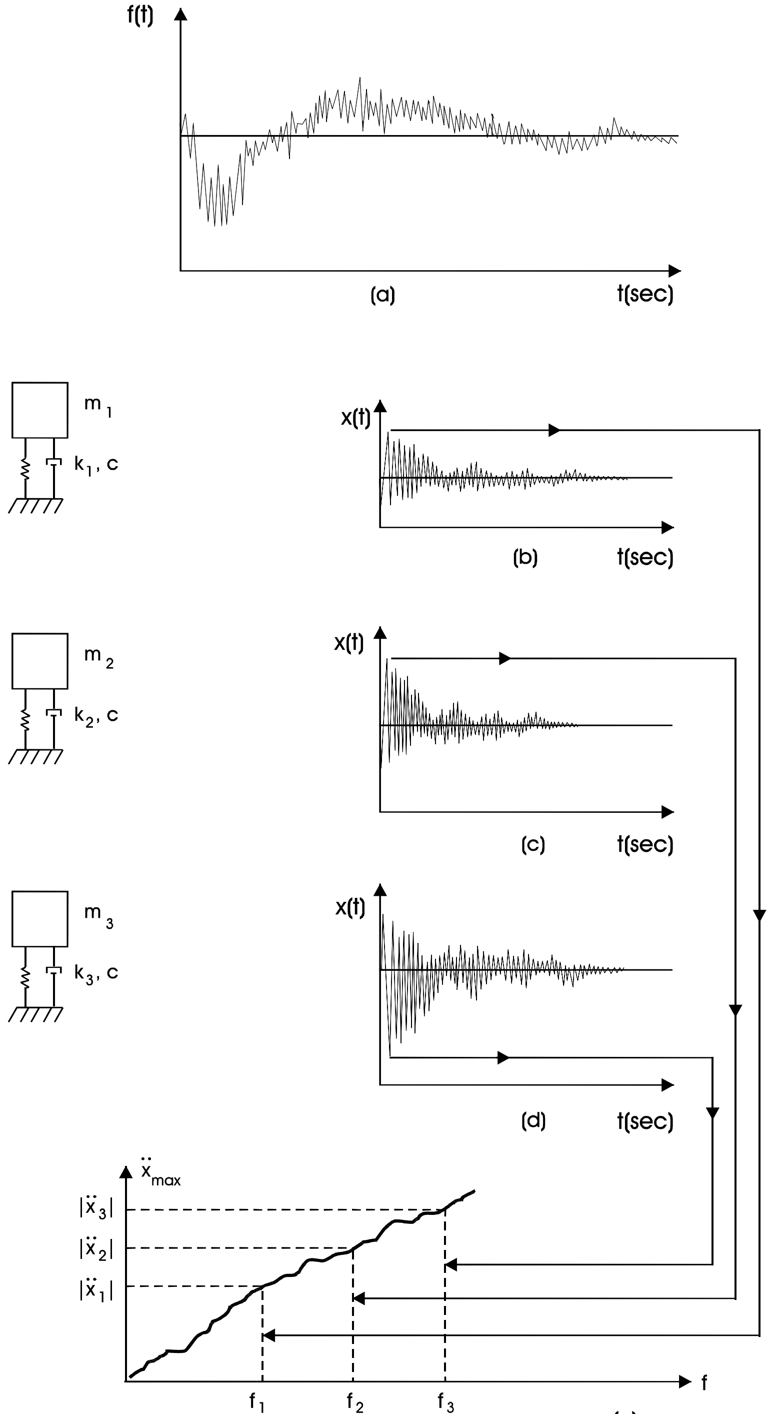
Figure 7(a) shows the variation of acceleration with time for a typical shock. Figures 7(b) to 7(d) show the variation with time of discrete frequencies at a short time (t) after the initial shock. The resulting shock response spectrum is shown on 7(e).

In design work, the fraction of critical damping is often assumed to be zero so that conservative results may be obtained, (Ref.5). In the calibration of shocks for shock testing, the level of damping may be specified as some non-zero, albeit small, value. For example, in MIL-STD-810E, the fraction of damping used in the data reduction to the response domain is 5%. Other than the fraction of damping, and the undamped natural frequency, no other knowledge of the system is required for the transformation to the response domain.

The shock response spectrum has probably seen its widest application in seismic qualification where it is used to determine dynamic shear loading factors for building codes (Ref.6). It is also widely used for characterizing laboratory shock tests and calibrating laboratory shock machines (Ref.2). MIL-STD-810E (Method 516.4) and STANAG 4141 are examples of the former, and latter, respectively.

An example of a rail impact test shock response spectrum, along with its original unfiltered acceleration time-history, is presented in Appendix B of this report. The family of curves represents different levels of (assumed) fraction of damping. Note that for the time domain acceleration representation, the effect of filtering on the peak value is significant. Conversely, the shock response spectrum is reduced from the unfiltered acceleration time history with no post-signal conditioning required. A data sampling frequency of 1 kHz is proposed in the draft standard presented in Appendix E.

A large number of computer model simulations have been developed during the course of this project; the acceleration results of a large number of these simulations have been reduced to the response domain and are discussed in Section 4. The source code, which embodies the algorithms presented in equations 1 through 5 is presented in Appendix F of this report.



(a) Impact input force: (b),(c),(d) Relative displacements of SDOF Systems with damping ξ and natural frequencies w1, w2, and w3. (e) Shock response spectrum

Figure 7. Development of the Shock Response Spectrum Curve

4. COMPUTER ANALYSIS

4.1 Introduction

As the results from the survey responses indicate, there is a lack of consensus as to whether load cells or accelerometers should be used to instrument the rail impact test. As transducers, load cells and accelerometers have inherent advantages and disadvantages. However, a more fundamental issue in connection to instrumenting the rail impact event is the determination of the most appropriate system variable with which to characterize the shock severity.

Due to the large physical mass of tank containers, the problem of characterizing the shock impact is non-trivial. In "conventional" shock testing where the test item being subjected to the shock is small in relation to the shaker table or platform, either the test item has negligible loading effect on the source input creating the shock, or else the loading effect can be mitigated by sophisticated control of the shaker table. Unfortunately, it is not practical to test a tank container (which could weigh up to 36 tonnes) on an electronically controlled shaker table.

Another factor complicating the impact test is that the test car which normally supports the tank container during the impact test is of comparable mass to the tank container. Consequently, there is a strong possibility of coupling between the two sub-systems of the tank container and the test car (where coupling in this context refers to the coupling of vibrational modes and subsequent energy transfer between the two sub-systems).

In order to determine the most practical method of characterizing the rail impact event, computer analysis was performed on a number of simple dynamic models by both T E S and CSTT. There were several objectives to the modelling:

1. Identify which variable, reaction force at the corner castings, or acceleration of the corner castings, was better at characterizing the shock input so as to allow or promote repeatable and reproducible test results;
2. Determine the influence of the input parameters such as tank container stiffness, tank container mass, and impact velocity, on both the resultant output force and/or acceleration time;
3. Determine what effect the tank container mass has on the resultant corner casting acceleration and what scaling factor, if any, can be used to compensate for tank container mass of lower than rated value; and
4. Reduce the modelling output response to the Shock Response Spectrum to see the effect of the system parameters in the frequency domain.

To achieve the first three goals, T E S concentrated on producing a number of linear, viscous damped, 2DOF model simulations. The results are presented in Appendix C of this report. CSTT concentrated on objective No.4 as well as No.2, using models which were less general and more specific to certain equipment (e.g. North American style standard draft gear) in that non-linear force versus displacement generators were used in the models. The results of CSTT's modelling simulations are presented in Appendix D.

4.2 Linear, Viscous Damped, 2DOF Model

4.2.1 Model Description

The system modelled is a two (2) degree-of-freedom (2DOF) system which is diagrammatically depicted in Figure 5. A 2DOF model is considered adequate to determine the effects of the relationship between the relative masses, relative stiffness and relative natural frequencies of the two subsystems on the response of the hypothetical tank container.

The rail car was modelled as a rigid body with mass and inertial properties in a two dimensional (2D) frame. The rail car was constrained to move only in a horizontal direction by connecting the rail car to the ground link through a translational joint having zero friction. The force and/or motion input to the rail car was through a spring and damper parallel arrangement which could be interactively modified by the user. The equations of motion for the tank container and rail car are presented in Equations 1, and 2, respectively.

Equation of Motion For The Tank Container (M2):

$$M2\left[\frac{d^2(X2)}{dt^2}\right] + C2\left[\left(\frac{d(X2)}{dt}\right) - \left(\frac{d(X1)}{dt}\right)\right] + K2\left[X2 - X1\right] = 0$$

(6)

Where: M2 = tank container (rigid body);

$d^2(X2)/dt^2$ = absolute acceleration of tank container centre of gravity;

C2 = equivalent damping coefficient of the tank container;

d(X2)/dt = absolute velocity of the tank container centre of gravity;

d(X1)/dt = absolute velocity of rail car centre of gravity;

K2 = equivalent stiffness of the tank container support structure;

X2 = absolute displacement of the tank container; and

X1 = absolute displacement of the rail car.

Equation of Motion For The Rail car (M1):

$$M1\left[\frac{d^2(X1)}{dt^2}\right]+C1\left[\left(\frac{d(X1)}{dt}\right)-\left(\frac{dU}{dt}\right)\right]+K1[X1-U]+C2\left[\frac{d(X1)}{dt}-\frac{d(X2)}{dt}\right]+K2[X1-X2]=0$$

(7)

Where: M1 = rail car (rigid body);

$d^2(X1)/dt^2$ = absolute acceleration of rail car centre of gravity;

C1 = equivalent damping coefficient of the rail car;

U = relative displacement between the rail car and the ground link;

dU/dt = relative velocity between the rail car and the ground link; and

K1 = equivalent stiffness of draft gear.

The mass of the rail car (M1 in Figure 5) was set at 20 tonnes which is the approximate mass of a 12 metre unladen flat car. The mass of the rail car was not changed throughout the analysis. The initial velocity of the rail car and tank container prior to impact with the fixed, immoveable ground link are depicted in Figure 5 as d(X1)/dt, and d(X2)/dt, respectively.

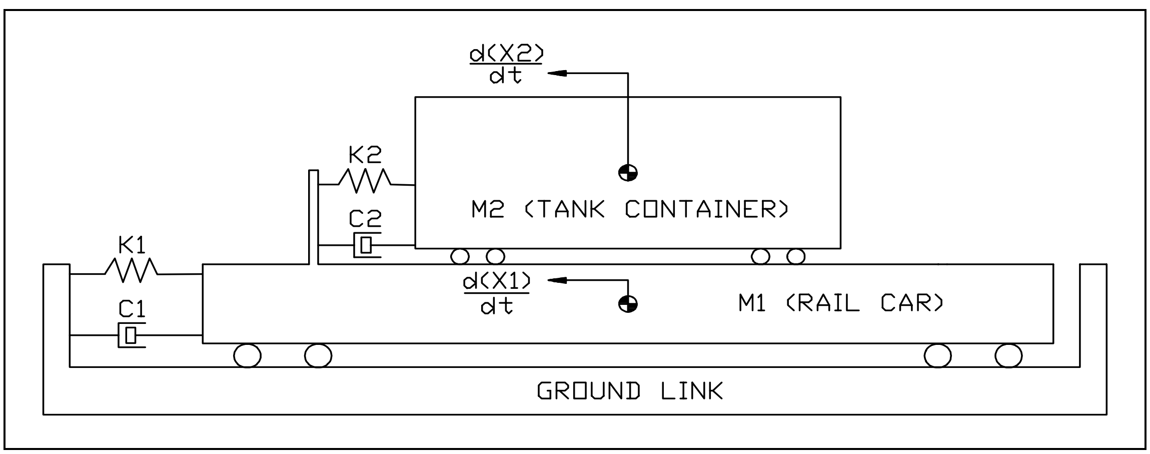


Figure 8. 2DOF System Model For Rail Impact Test

The equivalent longitudinal stiffness of the rail car (K1 in Figure 5) was varied from approximately 450,000 N/m to a maximum of 10,000,000 N/m. The most frequently used value was 1,000,000 N/m. The choice of 1,000,000 N/m represents an estimate based upon manufacturers data for a typical standard draft gear.

The damping coefficient (C1 in Figure 5) for the rail car was derived from an estimated damping ratio of either 0.20 or 0.40; these values are also estimates based on information on typical draft gear. Therefore, the rail car used in the modelling can be considered as an extremely stiff structure with a draft gear incorporating linear stiffness and linear damping. In reality, of course, the rail car would not be infinitely stiff and the draft gear characteristics would be non-linear. However, for the basic purpose of evaluating the interaction of the two major subsystems, these simplifications were considered acceptable.

Similarly, the tank container was modelled as a rigid body in a 2D plane having a mass which varied between 4 and 44 tonnes. The stiffness was varied from as low as 450,000 N/m to as high as 60,000,000 N/m. Very little published information is actually available on the modal and structural characteristics of tank containers. It should be noted, however, that tank container construction technique varies widely (e.g. “Beam” containers versus “Frame” containers). Therefore, during the modelling analysis, the rail car parameters (M1, K1, C1) were normally held constant while the tank container mass and/or stiffness was varied.

The tank container was connected to the rail car by a translational joint with zero friction and hence was also constrained to move only in the horizontal (X) direction. The absolute displacement of the tank container centre of gravity is depicted as X2 in Figure 5.

The shock input to the rail car was effected by inserting within the model a relative velocity generator (as shown in Figure 5 by an enclosed “U”). This relative velocity generator in effect prescribed an initial relative velocity between the ground link and the rail car centre of gravity. The relative velocity was not held fixed but rather was allowed to vary. The scenario being modelled could be considered as a rail car (laden with tank container) impacting a series of buffer cars having infinite inertia (or friction) with the hypothetical couplers at the ends of the cars having their “knuckles” set for “coupling”. This represents the sequence of events which is supposed to occur during Procedure A4.2 of CSA.B620. In practice, during the rail impact testing, this does not necessarily occur. In Europe, for example, the cars do not automatically couple but rather rebound. For the majority of the computer analysis, the initial velocity was set at 4.0 m/s (approximately 8.9 mph).

In the model, the rail car (depicted as the Test Car in Figure 5) is considered as an infinitely stiff structure with a spring/damper arrangement interposed between the rail car and the ground link which is the inertial frame of reference. The spring/damper arrangement , K1 and C1 in Figure 5, is intended to simulate a hypothetical draft gear. The force output in the spring at any point in time during the running of the dynamic analysis represents the summation of the damping force and spring force. These terms are clearly shown in the equations of motion presented with Figure 5. Similarly, the reaction forces of the tank container are embodied in the spring/damper arrangement labelled K2 and C2 in Figure 5. Therefore, the spring/damper force output from the model for K1/C1 roughly corresponds to the value which would be measured by a load cell interposed between the end sill of an actual rail car and the draft gear. Similarly, the force output from the model for K2/C2 roughly corresponds to the values which would be measured by a load cell interposed between the corner casting of an actual tank container and an actual rail car.

4.2.2 Results - Time History Plots

Sample results of a typical simulation are presented in Appendix C as Graph 1 and Graph 2 which depict forces and acceleration, respectively, as a function of time. The initial conditions are zero; at t=0 a step velocity change of 4.0 m/s is introduced between the inertial frame of reference (the ground link) and the rail car. For the parameters given (model 32), the resultant rail car peak acceleration is approximately 30 m/s^2 or approximately 3 g's. The large fraction of damping (0.2) introduced for the hypothetical draft gear is evident in the decaying sinusoidal shape of the "rail car force" curve. The period of vibration corresponding to the rail car system, comprised of rigid body and spring/damper combination, is clearly evident. Similarly, the shorter period corresponding to the stiffer subsystem, comprised of the tank container rigid body and spring/damper representing the support structure, is also evident. The tank container force and tank container acceleration waveforms are similar since the two values are simply related by the mass of the tank container. It should be noted that the tank container acceleration plotted in Graph 2 (and all subsequent graphs) corresponds to the acceleration of the tank container centre of gravity and not the corner castings. In the simplified 2DOF model, the mass of the tank container is assumed to be physically separated from the corner casting by the spring/damper arrangement representing the support structure stiffness. Since the rail car is also assumed to be rigid or infinitely stiff in relation to the draft gear, the acceleration of the rail car is assumed to be approximately equivalent to the corner casting acceleration.

4.2.3 Reaction Forces Versus Frequency

Graph 3 presents the simulation results for model 51 through 58 where the input parameter being varied is the tank container mass. The input is a 4.0 m/s step velocity change and the output parameter is reaction force at the primary and secondary spring/damper elements plotted against the ratio of the hypothetical natural frequencies (the hypothetical natural frequencies are calculated simply as the square root of the stiffness, K, divided by the mass, M, and are assumed independent of each other) where W1, and W2 correspond to the natural frequencies of the primary (rail car), and secondary (tank container) systems, respectively.

It is evident from Graph 3 that the reaction force at the tank container support (modelled by C2 and K2) varies as the tank container mass is increased, as expected. As shown in Graph 4, however, when the force is divided by the mass, the result (which is numerically equal to the tank container acceleration) is still shown to be strongly dependent on the ratio of the natural frequencies. It can be shown (e.g. Graph 6) that only when the natural frequency of the secondary system is significantly higher (by two-three times or more) than the primary system, does the reaction force at the base of the tank container become more or less independent of the system natural frequencies.

Conversely, the reaction force measured between the primary system and the input (draft gear force) is much less dependent on the system natural frequencies.

4.2.3 Acceleration Versus Natural Frequency

Graph 5 and 6 show the resultant acceleration for model 51-58 and model 40-48, respectively. Graph 5 reflects the effect of varying the tank container mass on the resultant accelerations of the primary and secondary masses, while Graph 6 reflects the effect of varying the tank container "support" stiffness. In both series of simulations, the effect on the rail car rigid body acceleration is negligible. Conversely, the effect on the tank container rigid body acceleration is again significant when the secondary system natural frequency approaches the natural frequency of the primary system. In general, therefore, the acceleration measured at the corner castings will not necessarily be equal to the reaction force divided by the tank container mass.

4.2.4 Estimating The Rail Car Acceleration

An issue that was raised by a number of correspondents canvassed during the project, was that the mass of the tank container as tested was often considerably less than the rated mass. This is due to the fact the tank containers are designed to carry liquids with specific gravity greater than water (which is the invariably used test mass). It is not practical or safe to stipulate the tank containers be tested at their rated capacity since this would involve testing with hazardous liquids.

A number of test agencies have attempted to address this problem by using a simple "correction" factor. For example, the EDC test procedure (EDC/TES/023/000 par. 6.4.3) stipulates that if the test mass is less than the rated mass then the new test level acceleration must be calculated with the formula:

$$\frac{\text{Rated Mass} \times 4}{\text{Test Mass}} \tag{8}$$

Where: "4" equals the nominal test value acceleration in gravitational units.

Note that this correction factor is applied to the acceleration measured at the corner castings and does not take into account the mass of the rail car which will play a significant role in the system dynamics. Reviewing the test results for the 2DOF model indicates that the mass of the tank container has much less influence on the acceleration (of the rail car/corner casting) than expected.

In the limiting case of a rigid equipment there will be only one predominant natural frequency given by (Ref.7):

$$\omega = \sqrt{\frac{K1}{M1 + M2}}$$

(9)

The transmitted acceleration of the primary system (rail car/corner casting) will be proportional to the velocity step input and primary stiffness:

$$\frac{d^2(X1)/dt^2}{\left(dU/dt\right) \times \sqrt{K1}} = \frac{1}{\sqrt{(M1+M2)}}$$

(10)

If the mass of M2 (the tank container) is increased or decreased by an amount equal to ΔM, then the new acceleration value should be related to the old acceleration value as shown by:

$$\left(\frac{d^2(X1)}{dt^2}\right)_{corrected} = \left(\frac{d^2(X1)}{dt^2}\right)_{measured} \times \frac{1}{\sqrt{1 + \frac{\Delta M}{M1 + M2}}}$$

(11)

Where: ΔM = (Rated Mass) - (Test Mass)

Graph 7 shows the results of estimating the acceleration values for increasing M2 values based upon the original acceleration value and the modified "correction" factor. The results show a close correlation between the actual and estimated values with the estimated value always on the conservative side.

As the ratio of ΔM /M1+M2 increases, the estimated value grows faster than the actual value; hence equation (6) tends to over-predict the increasing acceleration (as the tank container mass is decreased). This tendency is shown in Graph 8. Note, however, that even when ΔM is nearly 15%, the difference between the actual and estimated values is less than 5%.

Although the corrrection factor postulated in Equation (11) is an accurate predictor of peak acceleration response in the time domain for certain limited cases (i.e. rigidly mounted tank containers), it is not clear how to best to apply this factor in the frequency domain. For example, applying the correction factor to the measured acceleration time history data to produce the "corrected" acceleration time history data and then reducing the "corrected" acceleration data to the SRS would produce a SRS curve shifted down in value in relation to the original or "uncorrected" SRS. This would likely produce the correct effect in the vicinity of the first natural frequency but would also have the effect of reducing the acceleration values at all other frequencies as well. The net result, in terms of matching a “corrected” SRS curve to a specified test level would be a “correct” test severity level in the vicinity of the first natural frequency but an overtest in terms of the remainder of the spectrum. The amount of “overtest” is still expected, however, to be less than the “overtest” amount resulting from applying equation (8).

4.3 Non-linear, MDOF Model

4.3.1 Model Description

A multi-degree of freedom (MDOF) model shown in Figure 6, was developed based on CSTT’s impact testing facility. The motivation for specifically modelling CSTT’s test set-up was that acceleration and impact force data existed for an impact test of an empty box container. Thus, the model could be validated against the experimental data, thereby lending confidence to the results of subsequent simulations involving tank containers. The simulations were performed using the ADAMS Version 8.2 dynamics software package.

The two objectives of the MDOF model were:

1. Evaluate the effect of various input parameters such as impact velocity, draft gear type, tank container stiffness and mass, and impact velocity on impact force and acceleration time histories.
2. Reduce the modelling output response to the tank container Shock Response Spectrum to determine the effect of the input parameters on the SRS.

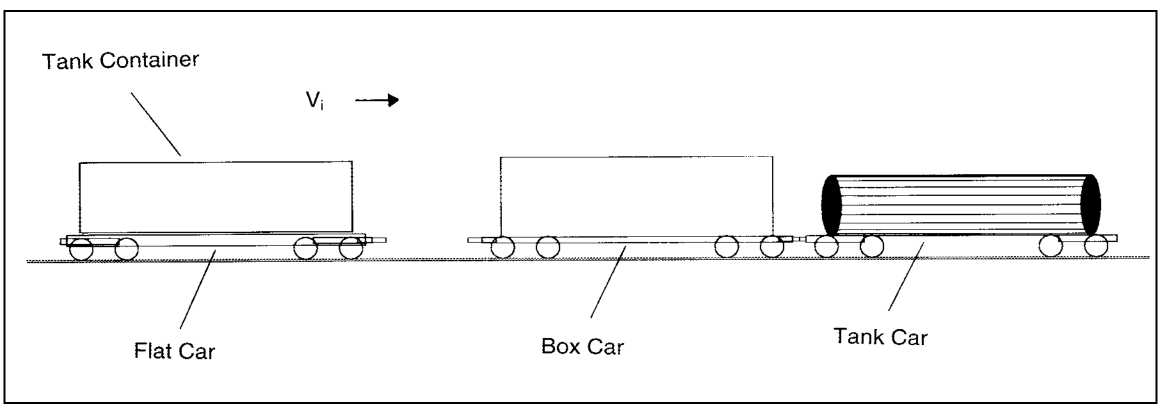


Figure 9. MDOF System Model For Rail Impact Test

CSTT conducts impact tests on tank containers utilizing a modified 55 ft flat car weighing 44,900 lb, an empty box car weighing 47,200 lb and a loaded tank car weighing 232,000 lb. The tank container to be tested is mounted on the flat car and the loaded hammer car is winched up a ramp for release. Impact velocity is set based on the height of the flat car and container on the ramp. The anvil cars are coupled with the box car ahead of the tank car.

The flat car was modelled as a rigid body mass with Westinghouse NY-11F standard draft gear at the impact end of the car. The box car and tank car were also modelled as rigid bodies with the box car having NY-11F standard draft gear and the tank car equipped with Mark 50 draft gear. The standard draft gear was modelled as a stiff non-linear spring with its force-displacement curve based on manufacturer's data. The box and tank cars were coupled together. The flat car and box car had their facing draft gear modelled to permit rebound and neither car had brakes set, i.e. the cars were free to move after impact. Simulations modelling end-of-car (EOC) cushioning on the flat car used a draft gear force-displacement curve based on a typical 10 inch stroke hydraulic draft gear.

The tank container was modelled as a mass-spring system with the front end pinned to the flat car and the back end constrained to slide in the longitudinal direction. The longitudinal stiffness of a standard (frame) tank container, k_{STD} was estimated to be 1.5 x 10^6 lb/in based on a finite element model developed by CSTT. The longitudinal stiffness of a beam tank container, k_{HI} was estimated to be 6.0 x 10^6 lb/in based on hand calculations by CSTT. The stiffness of an advanced suspension type tank container, k_{LO} was estimated to be 1.0 x 10^5 lb/in. The weight of the loaded standard tank container was 62,900 lb of which 10,050 lb was the tare weight. The weight of a heavy tank container was estimated to be 107,125 lb.

The fluid in the tank container was assumed to have a specific gravity of 1.0 with 3% volumetric ullage. The fluid's sloshing motion was modelled as a spring-damper system similar to a tank car fluid model previously developed by CSTT for Transport Canada.

The impact velocity, v_i of the hammer car was set as an initial condition and was varied between 4, 6 and 8 mph.

4.3.2 Initial Time-History Results

The model validation simulations using a box container mounted on the flat car with an impact speed of 8.4 mph showed very good correlation with the container acceleration, corner casting force and coupler force time domain test data. The agreement between the model and the test results indicated that the rail impact model was fundamentally correct.

4.3.3 Initial SRS Results

The shock spectrum response results are presented in Appendix D and represent the reduced tank container acceleration response data.

The SRS results were shown to be highly sensitive to impact velocity as well as draft gear type. Figure D.1 clearly shows the effect of impact velocity on the container shock spectrum response of a standard container on a flat car with standard draft gear. As the impact velocity was increased from 4 to 6 to 8 mph, the SRS curves for the container correspondingly climbed. Therefore, increasing the impact energy increases the resulting impact shock to the tank container.

Simulations were also performed with end-of-car cushioning on the flat car. The hydraulic draft gear was shown to reduce the shock response of the tank container as compared to standard draft gear.

The 8 mph simulation with standard draft gear is representative of impact testing performed in accordance to CSA B620 Proc. A4.2 and represents a worst case service-type impact. The combination of the modelled impact velocity and standard draft gear arrangement for the hammer and anvil cars results in higher shock response than would be predicted if either the impact velocity were lowered and/or end-of-car cushioning were employed.

The effect of different SRS damping ratios ranging from 0 to 20 percent was also examined. Figure D.2 shows that 0 percent damping generated the highest response and was, therefore conservative. However, this amount of damping used in the SRS may not be physically representative of real systems and therefore, may be too conservative. Further investigation and discussion are warranted. Overall, the results show that the SRS approach is not only feasible, but also practical.

4.3.4 Final Results

A last set of simulations were performed with the view of evaluating the SRS based on acceleration input as opposed to the acceleration response of the tank container. The results presented in Appendix D are the final summary results of the MDOF modelling effort. The simulations all involved impact speeds of 8 mph and employed standard draft gear on the flat car. Again these conditions represent a worst case service-type impact according to CSA B620 Proc. A4.2.

Figure D.3 shows the container input (flat car) acceleration time-histories for the four different types of containers. Note that the peak accelerations are in the range of 8.5 to 10.0 g while the container stiffness covers a range from 1.0×10^5 lb/in to 6.0×10^6 lb/in and the container weight varies from 62,900 lb to 107,125 lb. The highest acceleration is 18% higher than the lowest acceleration. It is worth noting that the peak accelerations of the flat car with the standard and the high mass container are the same which is consistent with TES' modelling results. However, the flat car response with the heavy container definitely exhibits higher transient effects.

Figure D.4 shows the corner casting force time-histories for the standard, low stiffness, high stiffness and high mass tank containers. Note that the high mass simulation has an appreciably higher peak corner casting force. The peak force of the high mass container is 300,000 lb while the peak force of the standard weight containers is in the range of 205,000 to 220,000 lb.

The calculated tank container acceleration responses are plotted in Figure D.5. The acceleration in g's was determined by dividing the peak corner casting force by the container weight. The peak tank container acceleration was 3.5 g for the standard and beam (high stiffness) containers, 3.3 g for the low stiffness container and 2.8 g for the high mass container, for a range of 2.8 to 3.5 g. The highest calculated acceleration is 25% higher than the lowest calculated acceleration.

The results of Figures D.3 and D.5 show that the peak input accelerations to the tank container are less affected by container design than the peak response accelerations calculated from corner casting force.

Figures D.6 and D.7 show the SRS plots for all four containers with 5 and 10 percent damping applied, respectively. These curves give an idea of the approximate magnitude of the SRS curve in the possible future tank container impact specification. It should be noted that SRS curves based on test data would be much smoother due to the large number of vibrational modes in real systems. The SRS data was processed for the period of 0.85 sec to 2.0 sec. The effect of the sampling period on shock response spectrum will have to be evaluated in the future.

In Figure D.6 (5% damping), the highest amplitude is approximately 180 m/s^2 or 18.3 g at 10 Hz in the high mass container. The high acceleration in the 4 to 10 Hz frequency range is due to transient vibration in the high mass container affecting the flat car. The highest shock amplitude for the standard container is approximately 175 m/s^2 or 17.8 g at 100 Hz. Note that 100 Hertz was arbitrarily selected as a cut-off frequency for the SRS plots.

In Figure D.7, (10% damping), the highest amplitude is approximately 170 m/s^2 or 17.3 g at 95 Hz in the low stiffness container. Note again the increased response of the high mass container in the 4 to 10 Hz range. The highest shock amplitude for the standard container is approximately 165 m/s^2 or 16.8 g at 100 Hz.

5. CONSULTATION - SURVEYS AND SITE VISITS

5.1 Survey Responses

In addition to published documents in the form of existing standards and reference texts, the responses to the survey questionnaires also provide a valuable source of information. Comments, opinions and information were generously provided by the following agencies:

Test Agencies:

a. Centre For Surface Transportation Technology (CSTT);
b. Centre National D'Essais De Tergnier (CNEST);
c. Forschungs-und-Technologie-Zentrum; and
d. Engineering Development Centre (EDC).

Tank Container Manufacturers:

a. Brenner Transportation Tanks For Chemical, Food & Dairy Products;
b. Consani Engineering;
c. Containers and Pressure Vessels Limited; and
d. Welfit-Oddy.

The original survey responses (with hand written English translation where applicable) are presented in the Appendix A of this report in addition to the representative questionnaire sent out to selected Tank Container manufacturers and the representative questionnaire sent out to selected Test Agencies.

5.2 Site Visits

A number of site visits were made to various test agencies and tank container manufacturers. These visits were invaluable in providing insight and documentary evidence of how tank containers are constructed and how they are tested. Highlights from the site visits are listed in chronological order.

Visit Testing Laboratories: SABS
A visit to the South Africa Bureau of Standards (SABS) was made by T E S on July 24, 1997. The following persons were in attendance:

1. Alicja Ondracek (SABS);
2. Iain Bennie (SABS);
3. Johan Mans (SABS);
4 Mike Evans (Welfit Oddy); and
5. Murray Sturk (T E S)

A video clip was shown of a rail impact test conducted at the Engineering Development Centre. Also, a tour was given of the SABS facilities used for conducting the static tank container tests (ISO 1496/3). The latest ISO tank container meeting (ISO Tc 104/SC2) was held in Beijing, China in October 1997. Alicja Ondracek is the South African national chairperson for the ISO working group.

Visit Testing Laboratories: Engineering Development Centre (EDC)
A visit was made by T E S to EDC of SPOORNET on July 24th 1997. A presentation was made by BKS (University of Pretoria) on their proposed test methodology for the rail impact test. The test method relies on reduction of the measured acceleration to the shock response spectrum. Details on how the spectrum would be presented and used were not completed; however, the general opinion was the shock response spectrum was a good starting point in developing a new test standard. EDC presented a copy of their current test method, EDC/TES/023/000/1991-07 (copy enclosed). The instruments used to collect data are accelerometers; EDC does not use load cells because they believe it radically alters the configuration of the tank container/wagon interface.

Visit Manufacturing Facilities: Henred Fruehauf
A visit was made by T E S/SABS to the manufacturing facility of Henred Fruehauf, represented by Malcolm Elston, Engineering Manager, on July 25th 1997. Mr. Elston had no major objection to the Shock Response Spectrum test method approach as long as the data analysis could be encoded in a “black box” arrangement which could not be altered by any one manufacturer to their advantage over another manufacturer. Mr. Elston also pointed out that it is very difficult to load a test container to its rated capacity since this requires a test liquid with very high specific gravity for those containers having low volume (e.g. 12,000 litres) but high capacity (36 tonnes). The issue was therefore on how the SRS would take into account the variance between the test mass and the rated capacity.

Visit Manufacturing Facilities: Welfit Oddy
Also on July 25th 1997, a visit was made by T E S to the S.African manufacturing facility of Welfit Oddy. A tour of the facility was given by Fred Belanger and Mike Evans.

Visit Manufacturing Facilities: Consani Engineering.
On July 28th 1997, a visit was made by T E S to the S. African manufacturing facility of Consani Engineering, represented by Solly Essop and Eric Brandes. Mr. Brandes pointed out that the shock impact to the tank container during test could vary considerably depending upon the fit-up or interface between the corner castings of the tank container and the test wagon, with the load being taken by either two or four of the four corners of the tank container. Mr. Brandes suggested a “surrogate” design test wagon would ensure repeatable input loads to the tank container during test. Mr. Brandes also questioned whether the presence of pressure within the tank container during testing would affect the test. Low volume, high capacity tank containers are built in order that relatively dense liquids, e.g. trimethyl lead, S.G.= 2.0, can be transported.

Mr. Brandes noted that a dynamic test simulated by static loading would permit in-house testing.

Visit to Manufacturing Facilities: CPV

A visit to Containers and Pressure Vessels Limited (CPV), represented by Robert Fossey, was made by T E S on July 30th 1997. Mr. Fossey, Technical Director of CPV, is also the ISO TC/SC2/WG4 Convenor. With regard to the request for information made at the March 29, 1996 ISO/TC/SC2/WG4 meeting, no significant test data has since been transmitted to Mr. Fossey. The lack of test data was attributed to the scarce number of rail impact tests conducted since March 29, 1996.

The following issues were discussed in conversation between Mr. Sturk and Mr. Fossey:

1. Fatigue tests;
2. Rated capacity of tank containers compared to typical load mass;
3. Surging of liquid within tanks;
4. Tank container design (beam versus frame tanks);
5. Tank symmetry;
6. In-house testing of tank containers;
7. Existing rail environment (i.e. North America versus Europe); and
8. Proposed test methods.

Visit Testing Laboratories: Tergnier, France

Richard Thomas and Murray Sturk visited the rail testing and railway refurbishing facilities at Tergnier, France on September 11th 1997. Mr. Sturk and Mr. Thomas were warmly received by the following personnel:

e. Genevieve Baudoin (Centre D'essais De Vitry);
f. Dominique Done (Service d'Agrément du Matériel Combiné;
g. Frédéric Dufetrelle (Chef UP Technique);
h. Jacky Vasset (Directeur Adjoint); and
i. Monsieur Legrand.

On the morning of the 11th 1997, Mr. Sturk and Mr. Thomas were able to view the facilities at Tergnier and evaluate some of the test equipment and apparatus presently in use in the rail impact testing. In the afternoon, a lengthy discussion ensued regarding the efficacy and validity of using load cells versus accelerometers. Also, the relative merits and shortcomings of using peak acceleration versus shock response spectrums were also discussed. Unfortunately, there was no data in electronic form available to present to T E S for analysis.

Visit Testing Laboratories: Minden Germany
Richard Thomas and Murray Sturk visited the railway facilities at Minden, Germany on the 15th and 16th of September 1997 and met with the following personnel:

a. Reinhard Damzog;
b. H. Rothman,
c. Herr Schwinder;
d. Herr Becker; and
e. Herr Sévin (translator).

Mr. Thomas and Mr. Sturk were able to view a mock rail impact demonstration, not involving a tank container, and the associated transducers and instrumentation. There were lengthy discussions which took place over the course of two days involving the anticipated problems of using the shock response spectrum. The authorities from Minden asked excellent questions regarding how the test spectrum would be developed and how it could be implemented during an actual test. Electronic data of an actual rail impact test was also provided to T E S for analysis. This data has already been analyzed (reduced to Shock Response Spectrum) and forwarded back to Minden for comment.

5.3 Test Data

Rail impact test data was requested from several test agencies. To date, the only test data received by T E S/CSTT in electronic format (suitable for data reduction) was that of a single tank car, as opposed to a tank container. The data, which contained the unfiltered acceleration time history of the impact event, was reduced to a Shock Response Spectrum which is presented in Appendix B.

6. RESULTS

6.1 Evaluation of Existing Standards

Table 1 in Section 2 lists the important similarities and differences in the existing rail impact test standards and procedures. This table reflects the wide disparity in instrumentation, test apparatus, and test procedure, in the current existing standards/procedures. It was not possible to extract from the existing standards a common or equivalent level of shock severity merely from evaluation of the standards themselves as they are presently written.

6.2 Survey Responses

There was a general lack of consensus among the respondents on whether the rail impact shock should be characterized in terms of reaction forces, peak acceleration values, or shock response spectrum. All of the respondents, however, expressed an interest and willingness to discuss alternative test methods if the test methods could be demonstrated as being reliable and repeatable.

Several respondents also mentioned the fact that the tank containers are normally not pressurized prior to testing. None of the existing rail impact test standards explicitly specify pressurization of the tanks as part of their test procedure. Yet, in practice, some tanks are designed to withstand internal pressure to several bar. As noted in Section 2.2.3, the MAWP specified on IMO Type 1 and Type 5 tank containers is typically 4 bar or greater which could produce circumferential stresses in the order of 100 Megapascals (at 4 bar gauge pressure). Also, the resultant loads on the end caps of the container shell due to internal pressurization can exceed by several times the inertial loads imposed by movement of the fluid contents. A consultation with two test agencies revealed there is no practical reason why the tanks could not be pressurized prior to test; in fact, a pressure test of the tank container shell is often performed just prior to the rail impact test.

6.3 Modelling Results

6.3.1 Linear, Viscous Damped, 2DOF Model

Reaction Force At Input To Rail car:
The reaction forces measured at the hypothetical draft gear were directly proportional to the input (velocity step) and heavily dependent on the rail car characteristics (K1,M1,C1). The reaction force was only moderately influenced by the tank container properties (M2, K2, C2).

Reaction Force Between The Rail Car and Tank Container:
The linear, viscous damped 2DOF model showed that the tank container stiffness had a large influence on the resultant reaction force between the rail car and tank container. The reaction force can therefore be considered to be dependent upon the tank container design. Although actual information was not available on the modal analysis of specific designs, tank containers can be expected to exhibit a broad range in stiffness due to varying construction techniques.

Acceleration Response of Rail Car:
Tank container stiffness and tank container mass had negligible effect on the resultant acceleration response of the rail car. A "correction" factor used in predicting the acceleration of the rail car when loaded with under mass tank containers was over conservative by approximately 5% when the difference in tank container mass, ΔM, divided by the total mass, (M1 + M2), was approximately 15%.

6.3.2 Non-linear, MDOF System

Initial Time-History Results
The box container impact model showed good correlation with the container acceleration, corner casting force and coupler force time-history data obtained from testing conducted at CSTT. These preliminary results gave confidence in the rail impact model. Simulations with a standard tank container model showed that hydraulic draft gear significantly reduced container acceleration, corner casting forces and coupler forces.

Initial SRS Results
Simulations were performed varying impact velocity and draft gear type. The tank container acceleration data were converted to SRS format. Increasing impact velocity was found to significantly increase the shock response of the tank container. End of car cushioning was found to reduce the shock response of the tank container. The damping ratio used in the reduction algorithm had an effect on the SRS curves.

Final Results
The final simulations evaluated the effect of tank container mass and stiffness on the flat car (i.e. tank container input) acceleration and corner casting force time histories. The flat car peak acceleration was found to be less dependent on container design that the calculated tank container peak acceleration response. The corner casting force was found to be dependent on container mass. The SRS plots provide an approximation of the response of a standard container on a flat car equipped with standard draft gear subjected to an 8 mph impact, as per CSA B620 Proc. A4.2.

7. CONCLUSIONS

1. As shown in Table 1, there are presently in existence several international rail impact test standards, all having variations in procedure, test apparatus and evaluation criteria. Due to these differences it is not possible to extract from the existing standards a common or equivalent level of shock severity merely from evaluation of the standards themselves as they are presently written.
2. If the rail impact shock is to be characterized from the measurement of a single variable, then longitudinal acceleration measured at the corner castings represents the closest approximation to the “true” input (i.e. independent of tank container design).
3. The two (2) most influential parameters affecting the rail impact shock imparted to the tank container, as ascertained from simplified modelling study, are the impact velocity and the longitudinal stiffness of the rail car, in that order.
4. The difference in acceleration of the corner castings as a result in the difference in mass between a tank container loaded to its maximum rating and a tank container loaded with water can be estimated from a simple expression, or “correction factor”, involving the rated mass, the test mass, and the test car mass.
5. Reducing the measured acceleration time-history data to the response domain presents the shock data in a format which is readily comparable and understandable with minimum extraneous information obscuring the “input”.

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8. RECOMMENDATIONS

Although it is not possible at this time to provide a test level based upon the existing body of test data, a proposed test procedure, as presented in Appendix E, has been prepared which has the capability of producing repeatable and reproducible test results. The proposed test procedure is predicated on the following concepts:

1. Longitudinal acceleration measured at the corner castings is the variable most representative of the “true” input which is independent, to the maximum extent possible, of tank container design, and test apparatus;
2. The best method of characterizing the shock input, based upon available measured acceleration time-histories, as proposed in 1. is reduction of the unfiltered time signal to the shock response domain;
3. The tank container should be pressurized prior to test to its Maximum Allowable Working Pressure (MAWP.); and
4. A simple algebraic expression can be used to provide a conservative correction factor for test mass below the nominally required rated value.

In order to establish the level or curve for the Shock Spectrum Test Level, the authors propose that a database of rail impact test data, collected from ongoing rail impact tests, be developed. The database would contain an adequate number of test runs to perform a statistical analysis (a brief description of what is required for statistical analysis of shock events is given in most references on the subject, two of which are listed in this report (Ref.4,8)). The quantity of data required for statistical analysis will be dependent upon many parameters including the variance in the data itself. A rough order of magnitude estimate is that twenty test runs for a given test car configuration and impact speed should be sufficient to characterize the test configuration (i.e. test car/draft gear system). It is feasible that the results from twenty impact tests (involving tank containers) could be made available from both EDC and CNEST before the end of 1998.

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APPENDIX A

SURVEY RESPONSES

**(*Not available in electronic format/*
Non disponible en format électronique)**

APPENDIX B

RAIL IMPACT TEST DATA - REDUCED TO SRS

(*Not available in electronic format/*
Non disponible en format électronique)

APPENDIX C

LINEAR, 2DOF SYSTEM MODEL - SIMULATION RESULTS

**(*Not available in electronic format/
Non disponible en format électronique*)**

APPENDIX D

NON-LINEAR, MDOF SYSTEM MODEL - SIMULATION RESULTS

(*Not available in electronic format/
Non disponible en format électronique*)

APPENDIX E

TANK CONTAINERS - LONGITUDINAL RAIL IMPACT TEST DRAFT STANDARD

(*Not available in electronic format/ Non disponible en format électronique*)

APPENDIX F

SHOCK RESPONSE SPECTRUM
SOURCE CODE

FORTRAN Source Code

```
      program SRS2A

c
c     SRS2A
c     Shock Response Spectra - version 2, mod "A"
c
c     This test program is used to calculate the maximax shock
c     resonse spectrum (SRS) at several natural frequencies.
c     Input is absolute acceleration, SRS output is derived
c     from the peak relative acceleration.
c
c     Reference:
c     "Principles and Techniques of Shock Data Analysis"
c     R. Kelly & G. Richman
c     The Shock and Vibration Information Center, SVM-5, 1969
c
c
c     MAX_POINTS   =  maximum size of data arrays
c
c     INPUT_FILE   =  name of input acceleration data file
c     DELTA_T      =  time between consecutive points in INPUT_ACC
c                     (1.0 / Sampling_Rate)
c     DAMPING      =  damping factor
c     FREQ_LOW     =  analysis frequency range, lower bound
c     FREQ_HIGH    =  analysis frequency range, upper bound
c     FREQ_STEP    =  analysis frequency, interval between bins
c     OUTPUT_FILE  =  name of output SRS data file
c
c     INPUT_ACC    =  input array, acceleration data
c     NPNTS        =  size of INPUT_ACC
c
c     W_N          =  undamped natural frequency (in radians)
c     W_D          =  damped natural frequency
c     SUM          =  accumulator
c     NBINS        =  number of frequency bins processed
c
c     WORK_EXP     =  work array, exponential term
c     WORK_SIN     =  work array, sine term
c     WORK_COS     =  work array, cosine term
c
c     OUTPUT_DISP  =  output array, calculated relative displacement
c     OUTPUT_ACC   =  output array, calculated absolute acceleration
c     OUTPUT_SRS   =  output matrix, frequency bin / maximax SRS
c

      implicit none

c...  Parameters

      integer*4  max_points
      parameter (max_points = 5000)

      real*4  pi, twopi
      parameter (pi = 3.141592654)
      parameter (twopi = 2.0 * pi)
```

```
c...  Variables

      real*4  input_acc(max_points), work_exp(max_points),
     &        work_sin(max_points), work_cos(max_points),
     &        output_disp(max_points), output_acc(max_points),
     &        output_srs(max_points,2)

      real*4  t1, t2, t3,
     &        freq_low, freq_high, freq_step, freq_bin,
     &        delta_t, w_n, w_d, damping, sum

      integer*4  i, k, n, ierr, nbins, npnts

      character*80  input_file, output_file

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c
c...  Start of main


c
c...  Open parameter file and get run info
c
c     Parameter file should contain:
c       Line 1 = Input data filename
c       Line 2 = Intersample time (delta time)
c       Line 3 = Damping (0% to 99%)
c       Line 4 = Frequency analysis band:
c                  t1 = lower bound
c                  t2 = upper bound
c                  t3 = step
c       Line 5 = Output data filename

      open (unit=1, file='SRS2A.IN', form='FORMATTED',
     &      mode='READ', status='OLD', iostat=ierr)
      if (ierr .ne. 0) then
        write (*,*) '%Error: cannot open input parameter file'
        goto 8000
      endif

c
c...  Get input data filename

      read (1,'(a)',iostat=ierr) input_file
      if (ierr .ne. 0) then
        write (*,*) '%Error reading input data filename'
        close (unit=1)
        goto 8000
      endif

c
c...  Get intersample time

      read (1,*,iostat=ierr) t1
      if (ierr .ne. 0) then
        write (*,*) '%Error reading delta time parameter'
        close (unit=1)
        goto 8000
      endif
```

```
      if (t1 .le. 0.0) then
        write (*,*) '%Error: invalid delta time parameter'
        close (unit=1)
        goto 8000
      endif

      delta_t = t1

c
c...  Get damping

      read (1,*,iostat=ierr) t1
      if (ierr .ne. 0) then
        write (*,*) '%Error reading damping parameter'
        close (unit=1)
        goto 8000
      endif

      if (t1 .lt. 0.0  .or.  t1 .gt. 99.0) then
        write (*,*) '%Error: invalid damping parameter'
        close (unit=1)
        goto 8000
      endif

      damping = t1 / 100.0

c
c...  Get frequency band parameters

      read (1,*,iostat=ierr) t1, t2, t3
      if (ierr .ne. 0) then
        write (*,*) '%Error reading frequency band parameters'
        close (unit=1)
        goto 8000
      endif

      if (t1 .lt. 0.1  .or.  t1 .gt. 250.0) then
        write (*,*) '%Error: invalid lower frequency bound'
        close (unit=1)
        goto 8000
      endif

      if (t2 .lt. t1  .or.  t2 .gt. 250.0) then
        write (*,*) '%Error: invalid upper frequency bound'
        close (unit=1)
        goto 8000
      endif

      if (t3 .lt. 0.1) then
        write (*,*) '%Error: invalid frequency step parameter'
        close (unit=1)
        goto 8000
      endif

      freq_low  = t1
      freq_high = t2
      freq_step = t3

c
c...  Get output data filename
```

```
      read (1,'(a)',iostat=ierr) output_file
      if (ierr .ne. 0) then
        write (*,*) '%Error reading output data filename'
        close (unit=1)
        goto 8000
      endif

c
c...  Close parameter file

      close (unit=1)

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c
c...  Open input data file and read acceleration data

      open (unit=2, file=input_file, form='FORMATTED',
     &      mode='READ', status='OLD', iostat=ierr)
      if (ierr .ne. 0) then
        write (*,*) '%Error opening input data file'
        goto 8000
      endif

      n = 0

1000  read (2,*,iostat=ierr) t1
      if (ierr .lt. 0) goto 1500        ! end of file
      if (ierr .ne. 0) then
        write (*,*) '%Error reading from input data file'
        close (unit=2)
        goto 8000
      endif

      n = n + 1
      if (n .gt. max_points) then
        write (*,*) '%Error: input data buffer overflow'
        close (unit=2)
        goto 8000
      endif

      input_acc(n) = t1

      goto 1000


1500  close (unit=2)

      if (n .lt. 2) then
        write (*,*) '%Error: too few points in input data file'
        goto 8000
      endif

      npnts = n

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c
c...  Open output data file and write header

      open (unit=4, file=output_file, form='FORMATTED',
```

```
     &      mode='WRITE', status='NEW', iostat=ierr)
      if (ierr .ne. 0) then
        write (*,*) '%Error opening output data file'
        goto 8000
      endif

      write (4,*,iostat=ierr) '      Frequency       Amplitude'
      if (ierr .ne. 0) then
        write (*,*) '%Error writing to output data file'
        close (unit=4)
        goto 8000
      endif

      close (unit=4)

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      write (*,*) delta_t
      write (*,*) damping
      write (*,*) freq_low, freq_high, freq_step
      write (*,*) npnts

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c
c...  Loop to generate data at each frequency bin

      nbins = 0
      freq_bin = freq_low

5000  if (freq_bin .gt. freq_high) goto 6000

        nbins = nbins + 1
        write (*,*) '%Processing frequency bin: ', freq_bin, ' Hz'

c
c...    Convert natural frequency from Hz to radians

        w_n = twopi * freq_bin

c
c...    Calculate damped natural frequency

        w_d = w_n * sqrt(1.0 - damping**2)

c
c...    Generate work arrays

        t1 = -damping * w_n * delta_t
        t2 = w_d * delta_t

        do i=1,npnts
          t3 = float(i-1)
          work_exp(i) = exp(t1 * t3)
          work_sin(i) = sin(t2 * t3)
          work_cos(i) = cos(t2 * t3)
        enddo

c
c...    Calculate relative displacement
```

```
        t1 = -delta_t / w_d

        do i=1,npnts
          sum = 0.0
          do k=1,i
            n = i - k + 1
            sum = sum + ( input_acc(k) * work_exp(n) * work_sin(n) )
          enddo
          output_disp(i) = t1 * sum
        enddo

c
c...    Calculate absolute acceleration

        t1 = delta_t * 2.0 * damping * w_n
        t2 = ((2.0 * damping**2) - 1.0) * w_n**2

        do i=1,npnts
          sum = 0.0
          do k=1,i
            n = i - k + 1
            sum = sum + ( input_acc(k) * work_exp(n) * work_cos(n) )
          enddo
          output_acc(i) = (t1 * sum) + (t2 * output_disp(i))
        enddo

c
c...    Scan for acceleration maximums

        t1 = abs(output_acc(1))

        do i=2,npnts
          t1 = max(t1, abs(output_acc(i)))
        enddo

        output_srs(nbins,1) = freq_bin
        output_srs(nbins,2) = t1

c
c...    And loop for next frequency bin

        freq_bin = freq_bin + freq_step
        if (nbins .lt. max_points) goto 5000

        write (*,*) '%Warning: SRS buffer full'

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c
c...  Save SRS results

6000  open (unit=4, file=output_file, form='FORMATTED',
     &      mode='WRITE', access='APPEND', status='OLD',
     &      iostat=ierr)
      if (ierr .ne. 0) then
        write (*,*) '%Error opening SRS results file'
        goto 8000
      endif

      do i=1,nbins
```

```
          write (4,*,iostat=ierr) output_srs(i,1), output_srs(i,2)
          if (ierr .ne. 0) then
            write (*,*) '%Error writing to SRS results file'
            close (unit=4)
            goto 8000
          endif
        enddo

        close (unit=4)

c
c...   Exit

8000   stop
       end
```