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CARGO RESPONSE TO RAILCAR IMPACT AND TIEDOWN LOAD ANALYSIS*

by

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ABSTRACT

An analytical study that investigated the loads produced during coupling of railcars carrying heavy shipping containers is described. The structural model of the impact event is represented by a lumped parameter technique. Each discrete mass lump possesses longitudinal, vertical, and rotational degrees of freedom. The resulting computer simulation provides for nonlinear railcar coupler stiffness and linear damping forces in the coupler and container tiedowns. Results include response to parametric variations in container weight, impact speed, and tiedown stiffness. Container dynamic response and tiedown loads are found to depend heavily on these parameters. Also, railcar bending and subsequent vertical motion are shown to be important contributors to these responses. When experimentally substantiated, the model can serve as a useful tool in the design and evaluation of shipping container tiedown structure.

INTRODUCTION

When massive energy materials containers are moved by rail, considerations must be given to forces produced by railcar coupler impacts. These forces may be large and can affect the container, its tiedown structure, and the railcar. Few rational dynamic analyses exist for the problem of dynamic interaction between these three systems. Current design practice employs static analysis procedures, using empirical load factors, to design tiedown systems, but no dynamic analysis has confirmed the applicability of these static analysis methods. There has been a need to develop a computer model that will handle the vertical, rotational, and longitudinal motions of both the container and the railcar. This is especially important for the heavier spent nuclear fuel containers that are transported by railcar.

In this paper an analytical model of railcar impact during coupling is developed using a lumped structural parameter technique.[1] The model is appropriate for horizontally laded

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flexible containers and most flatbed railcars. A vertically laded rigid container can also be simulated. A digital computer simulation of this model, RICTL, (Railcar Impact Container Tiedown Loads), has been developed by the Los Alamos Scientific Laboratory (LASL). This simulation presently has provision for a nonlinear coupler as the only nonlinearity. However, the program is written so that nonlinear subroutines can be easily added for the tiedown configuration and for other longitudinal members of the railcar. With RICTL in its present form, we have studied the effects of variations in certain analytical model parameters. These parameters include container weight, impact speed, and tiedown stiffness (and corresponding damping). The effects of varying these parameters include the response accelerations of the railcar and the container and tiedown loads.

This paper contains the general method of analysis and the application to a specific railcar carrying three different cargo weights with variations in coupling speed and tiedown parameters, a discussion of the results of these analytical studies, and the general conclusions of the study.

A. Method of Analysis

The structural dynamic method of analysis used to study the problem of coupling impact for a railcar carrying an energy material shipping container is based on the lumped parameter technique of structural modeling. Figure 1 shows a side view schematic of a specific railcar[2] and coupler, a shipping container with its tiedown configuration, and the mass lumping schematic with degrees of freedom assigned to each lumped mass. Mass 11 represents the initially stationary train that the railcar strikes. The container geometry is based on the NFS-M100 cask system, but with three weights--13.6, 27.2, and 63.5 metric tonnes. Seven mass lumps represent the initially moving railcar, and three mass lumps represent the shipping container. The mass lumps are interconnected by massless springs and/or dampers as indicated by the members designated ℓ_{ij} between mass lumps i and j where $i,j = 1,2,\ldots N$. The nonlinear coupler is designated by stiffness $k_{7,11}$ and damping $c_{7,11}$ as shown in Fig. 1. The impact velocity (V_I) is given as the initial velocity in the direction shown for each of the ten masses that constitute the railcar-container structural dynamic system.

The lower schematic of Fig. 1 indicates the motion degrees of freedom for the system. The designation u_i refers to the positive horizontal displacement of the i^{th} mass; v_i refers to the positive vertical displacement of the i^{th} mass; and θ_i refers to the positive rotational motion of the i^{th} mass. There are 29 degrees of freedom. The dynamic equations of motion in matrix form are:

$$[M]\,\{\ddot{x}\} + [C]\,\{\dot{x}\} + [K]\,\{x\} = \{F\} \tag{1}$$

with initial conditions

for $\dot{x}_j = \dot{u}_i$, $(i = 1, 2, 3 \ldots 10)$

$\dot{x}_j = V_I$ and

all other $\dot{x}_j = 0$.

Also, in the eleventh equation of Eqs. (1), $F_{11} = -R_T$, the sliding friction force of the initially stationary train being struck (Fig. 1). All other $\{F\} = 0$. The vector $\{x\}$ refers to the appropriate absolute motion of each of the 29 degrees of freedom; [M] refers to the mass and mass-moment-of-inertia matrix; [K] refers to the stiffness matrix; and [C] refers to the damping matrix. Equations (1) constitute a set of 29 second order nonlinear differential equations, which are solved with the digital computer program RICTL.

The mass matrix [M] is a diagonal matrix of 29 elements derived from railcar and container mass and mass-moment-of inertia data. Table I includes the numerical data used for the specific railcar and cargo considered. Details of lumping methods are included in Ref. 3.

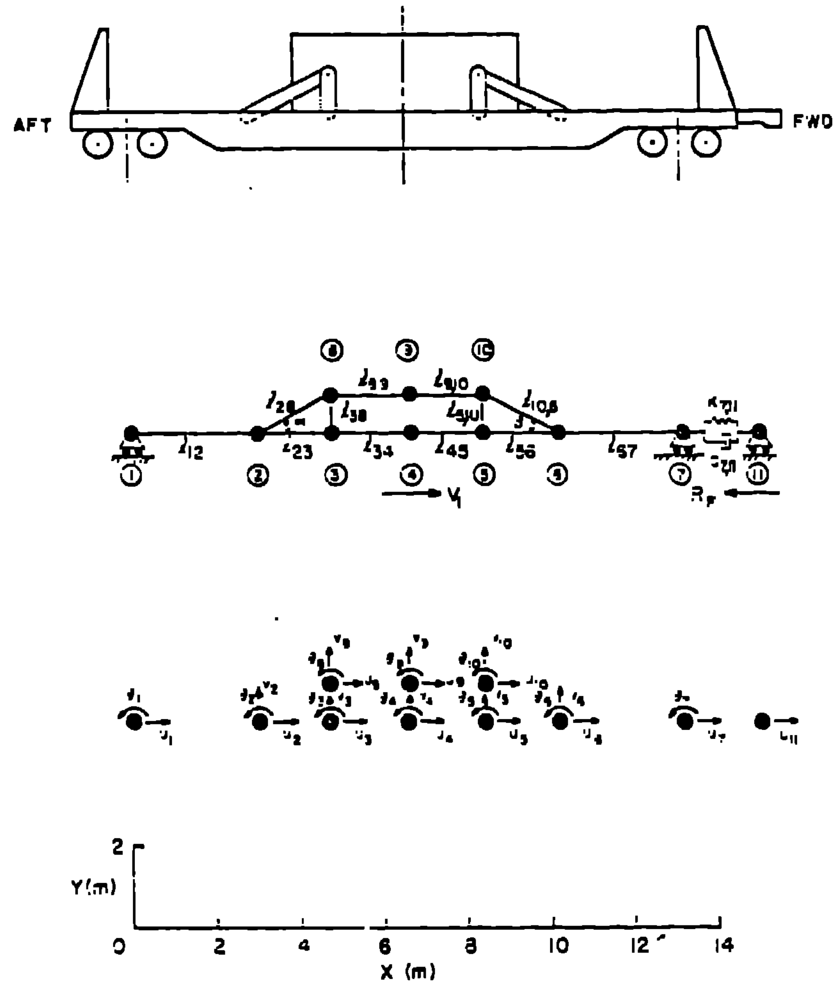

Fig. 1. Structural modeling technique for railcar, container, and coupler.

The stiffness matrix [K] is derived in the standard way[1] by first obtaining a transformation matrix [β] from absolute to relative coordinates. The axial and bending component stiffnesses [k] are as given for beams,[1] and the resulting stiffness matrix [K] was formulated by the matrix product

$$[K] = [\beta^T]\ [k]\ [\beta]\ . \qquad (2)$$

TABLE I

ANALYTICAL MODEL DATA

Railcar data from Appendix L, "Stress Analysis for 50 Ton 53 Ft. 6 In. Bulkhead Flat Car," Published by Association of American Railroads, January 1966.

Length between trucks = 13.182 m (519 in)
Container weights of 13.6, 27.2, 63.5 tonnes (15, 30, 70 tons)

Weights of railcar

(1)	Distributed weight	= 15,292 N/m (87.32 lb/in)
(2)	Bulkheads (2)	= 16,414 N each at x = 0, x = 13.182 m (3,690 lb each at x = 0, x = 519 in)
(3)	Weight of trucks (2)	= 31,805 N each at x = 0, x = 13.182 m (7,150 lb each at x = 0, x = 519 in)

Moduli (Pa)

Railcar $E_R = 2.0684 \times 10^{11}$ (30,000,000 psi)
Container $E_C = 2.0684 \times 10^{11}$ (30,000,000 psi)

Tiedown damping (est. $\zeta = 0.05$) - $(\frac{N}{m/s})$

Impact velocity (km/h) $V_I = 8, 11.3, 14.5, 17.7$

Stationary train - Weight = 226.8 tonnes - 2.224×10^5 N (500,000 lb)

- Resistance force (coeff. of friction of 0.2) = 4.448×10^5 N (100,000 lb)

The damping matrix [C] was formulated by first assuming linear viscous damping in the tiedown members and in the coupler. These are the only damping forces included. They were formed in a relative damping matrix $[\gamma]$, and then [C] was formulated by the matrix product

$$[C] = [\beta^T]\,[\gamma]\,[\beta] \quad . \tag{3}$$

The nonlinear spring ($k_{7,11}$) associated with the draft gear of the coupler was as given by Magnuson and Wilson as a standard Miner RF-333 draft gear. A subroutine was written that updates the $k_{7,11}$ element in [K] based on the current (time dependent) relative displacement in the coupler ($u_{7,11} = u_{11} - u_7$).

A Runge-Kutta numerical integration routine was used to solve Eqs. (1). A constant time step increment was used. The results of computer runs indicate that a small enough time step for convergence does not require excessive computer time to simulate the impact event for sufficient duration to obtain maximum motions and loads.

The constraint forces (including tiedown loads) were computed from the following relationships for every time step:

$$\left\{X_{rel}\right\} = [\beta]\{X\} \; , \tag{4}$$

$$\left\{\dot{X}_{rel}\right\} = [\beta]\{\dot{X}\} \; , \tag{5}$$

$$\{F\} = [k]\,X_{rel} + [\gamma]\,\dot{X}_{rel} \quad . \tag{6}$$

where $\{F\}$ is the constraint force vector, and

$\{X_{rel}\}, \{\dot{X}_{rel}\}$ = the relative displacement and velocity of the ends of each member carrying a constraint force.

B. Discussion of Results

The analytical results were obtained using the RICTL code. The study was parametric in nature with container weight, impact velocity, and tiedown stiffness and damping as parameters. Complete results of the study are presented in Ref. 3. The results of interest presented here include peak values of the time profiles of

(1) Horizontal deceleration of the railcar at the point of impact,
(2) Vertical response acceleration of the railcar at struck-end container attachment,
(3) Horizontal deceleration of the container c.g.,
(4) Vertical acceleration of the container c.g.,
(5) Struck-end tiedown load horizontal component,
(6) Struck-end tiedown load vertical component, and
(7) Struck-end tiedown load bending moment.

The tiedown loads were expressed in terms of the total load of the two struck-end tiedowns (which produced the maximum tiedown loads) and involved the axial load in each of the tiedown members $\ell_{5,10}$ and $\ell_{10,6}$. These axial loads were converted to equivalent horizontal and vertical loads and a bending moment of a single member attached rigidly between Mass 10 and Mass 5.

Figure 2 shows the variation in peak acceleration with the container weight for the variables of interest. The significant result shown in Fig. 2 is that, for heavy containers, the vertical acceleration of the railcar is nearly as large as the horizontal deceleration. This result emphasizes the relative importance of vertical dynamics compared with horizontal dynamics. This implies that bending effects of the railcar are important in considering tiedown loads. Figure 3 shows the peak tiedown load variation with container weight. This plot confirms the fact that bending effects in the tiedowns need to be considered. Although the net vertical component is small compared with horizontal tiedown loads, the bending moment is certainly significant.

Figures 4 and 5 show variation of the same peak load variables as Figs. 2 and 3 with impact speed as the parameter. The largest container weight, 63.5 tonnes, was used for these studies. The curves all indicate that the peak loads vary linearly, or approximately linearly, with impact speed. This is an entirely predictable result, since the impulse of impact is proportional to the momentum, which varies linearly with impact velocity. Variations from linearity are due to the peak loads being based on vibratory responses that peak at different relative times during the impact event. In the low speed case (3 km/hr), the coupler load peaks at .006 s when the coupler stiffness is high (10.51×10^7 N/m). In the higher speed cases, the coupler load peaks at .038 s when the coupler stiffness is low (2.1×10^7 N/m) during deflection of the draft gear. Thus the nonlinear stiffness characteristic of the coupler accounts for the deviation from linearity of some of the peak response loads.

The peak horizontal deceleration of the railcar is linearly related to impact velocity because it occurs 40 µs after initial impact for all impact speeds. At this time the coupler stiffness is still in the linear range, and the draft gear is undeflected. The container response acceleration peaks are linearly related to impact velocity because, although they occur at times when the draft gear is deflecting, they occur at the same time (.012 s for horizontal response and .017 s for vertical response) for each impact velocity.

Figures 6 and 7 show the peak load variations of the 63.5 tonne container configuration as a function of container tiedown axial stiffness. The tiedown stiffness embodies the same values of stiffness in each of the four members $\ell_{2,3}$, $\ell_{1,9}$, $\ell_{5,10}$, and $\ell_{10,6}$. Also, since the

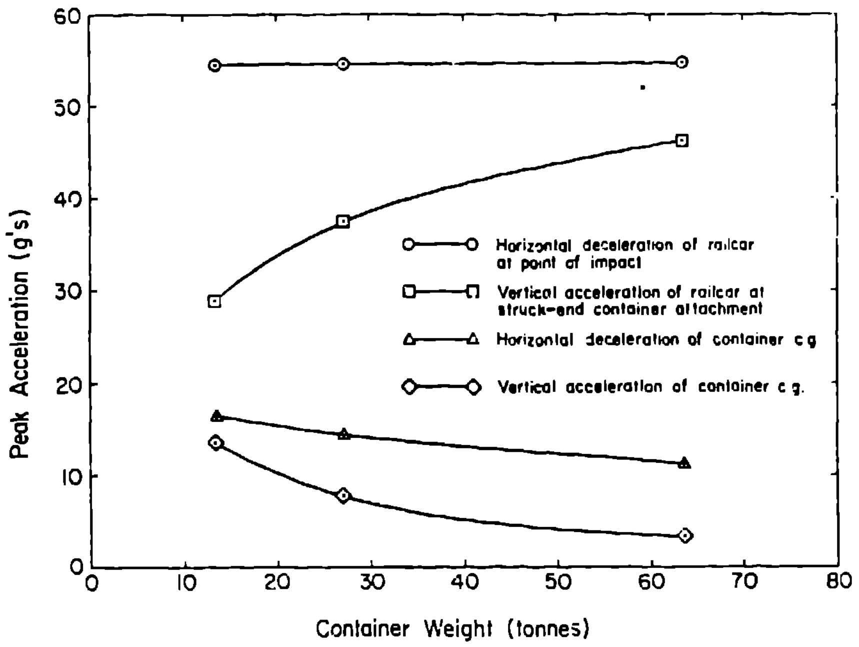

Fig. 2. Peak acceleration during impact at 17.7 km/h (11 mph).

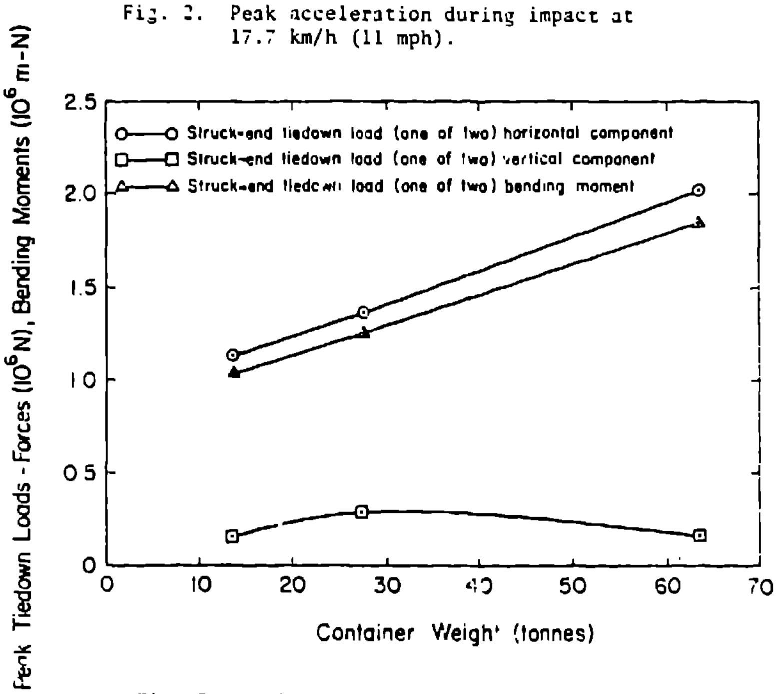

Fig. 3. Peak tiedown loads during impact at 17.7 km/h (11 mph).

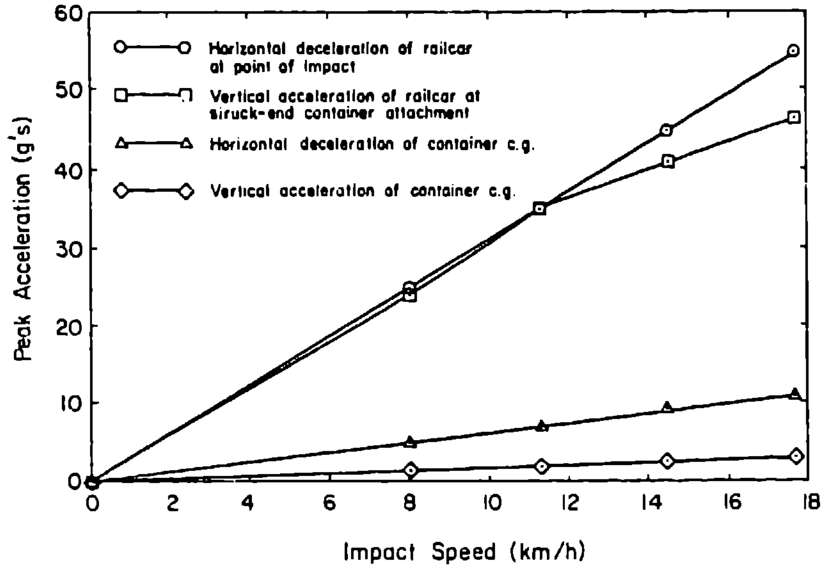

Fig. 4. Peak acceleration during impact of railcar carrying 63.5 tonne container.

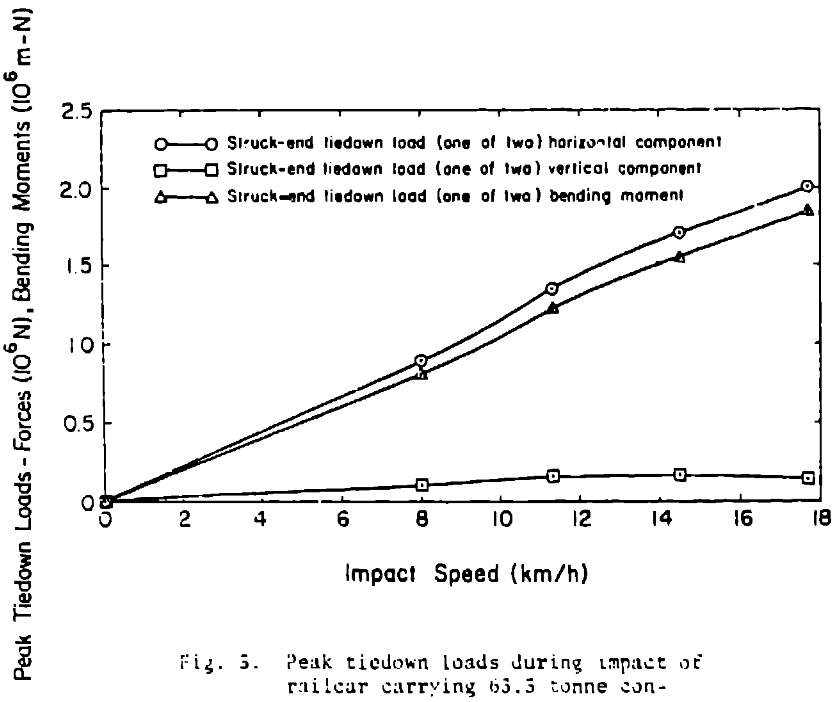

Fig. 5. Peak tiedown loads during impact of railcar carrying 63.5 tonne container.

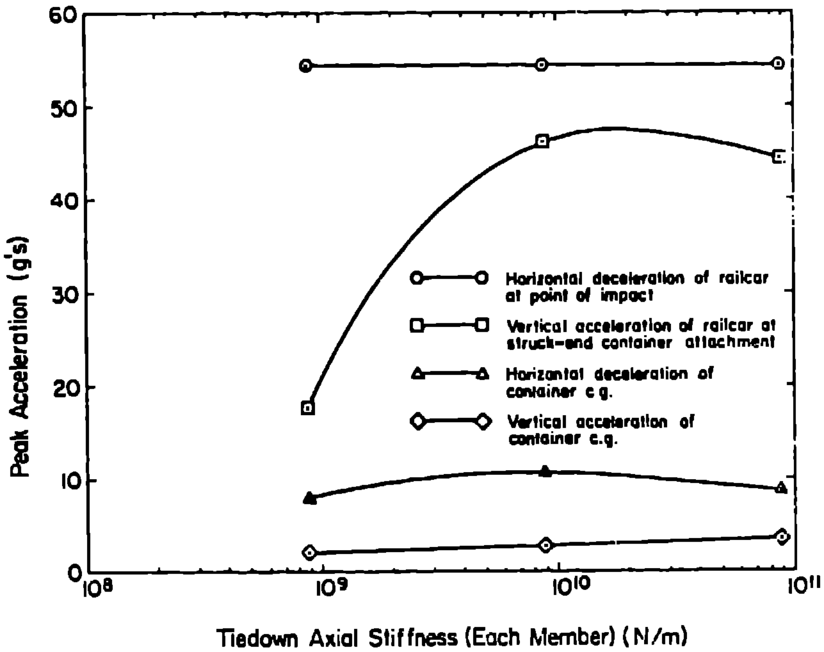

Fig. 6. Peak acceleration during impact at 17.7 km/h (11 mph) of railcar carrying 63.5 tonne container.

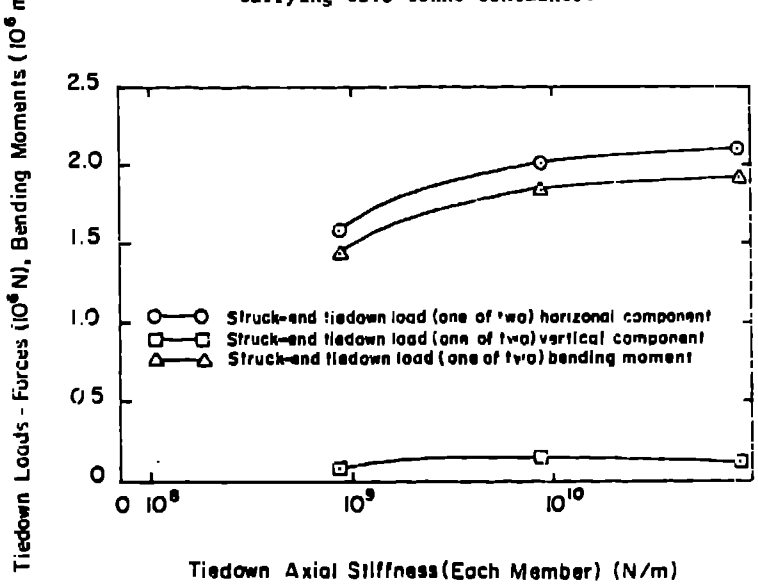

Fig. 7. Peak tiedown loads during impact at 17.7 km/h (11 mph) of railcar carrying 63.5 tonne container.

tiedown damping was assumed to vary with the tiedown stiffness, the damping constants were varied by the same ratio as the stiffness. The most notable result is that the response variables that depend on the tiedown interconnections to transmit loads are markedly reduced when the tiedown stiffness is reduced. The railcar vertical acceleration plot of Fig. 6 and the tiedown load plots of Fig. 7 show this effect.

Time profiles were computed for the heaviest weight container-railcar configuration striking an initially stationary train at the highest coupling impact velocity (17.7 km/hr or 11 mph). For brevity, these profiles are not included, but they were computed to show the oscillatory behavior of the impact event. The impact deceleration has an apparent dominant frequency of 125 Hz, which is revealed initially in the tiedown load of member L_{11s}, the tiedown having the dominant loading. This result appears to be approximately in accord with the result obtained by Magnuson and Wilson in their model that has only longitudinal degrees of freedom for the ATMX car tiedown structure (dominant frequency of 100 Hz).[4] The magnitude of peak deceleration that we obtained (54.5 g's) also compares with the Magnuson and Wilson results for cargo weights between 178,000 N and 445,000 N, which gave 52 g's and 58 g's respectively for a spent fuel cask system with 3.2 mm travel space. The configuration of Magnuson and Wilson and our configuration are not e the same, so a direct comparison is not possible. However, the closeness of the two resu for longitudinal motion variables is of interest.

CONCLUSIONS

The results indicate that the RICTL simulation of railcar coupling impact dynamics produces results that are plausible when compared with results of other impact analyses.[4,7] The analysis method applied to the design of cargo tiedowns should be a significant improvement over current static design methods that employ equivalent empirically derived load factors in the design of tiedowns.[5] Also, vertical motion and bending are found to be important effects to be considered in the design of tiedown structure. More cannot be said about the adequacy of the simulation until a degree of correlation is established between the RICTL model and actual impact tests.

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