U.S. Department
of Transportation

**Federal Railroad
Administration**

Effects of Coupler Height Mismatch on the Structural Integrity of Railroad Tank Car Stub Sills

**Office of Research and
Development
Washington DC 20590**

DOT/FRA/ORD-01/14

**Final Report
December 2001**

This document is available to the Public through the National Technical Information Service, Springfield, Virginia 22161

NOTICE

This document is disseminated under the sponsorship of the Department of Transportation in the interest of information exchange. The United States Government assumes no liability for its contents or use thereof.

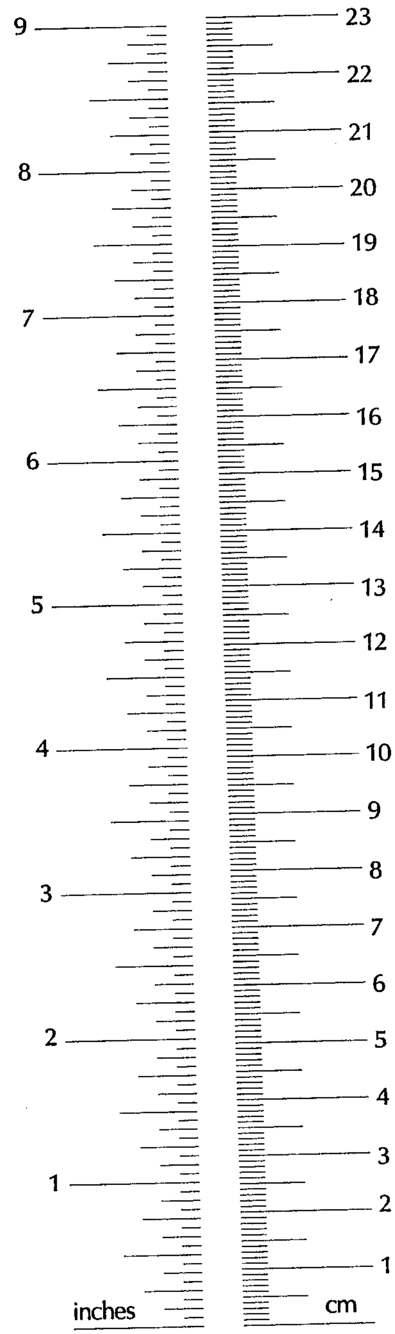
NOTICE

The United States Government does not endorse products or manufacturers. Trade or manufacturers' names appear herein solely because they are considered essential to the objective of this report.

METRIC CONVERSION FACTORS

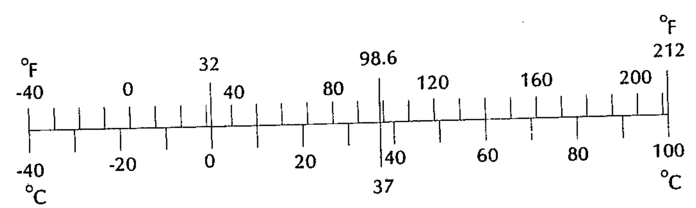
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.50	centimeters	cm
ft	feet	30.00	centimeters	cm
yd	yards	0.90	meters	m
mi	miles	1.60	kilometers	km
		AREA		
in^2	square inches	6.50	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.80	square meters	m^2
mi^2	square miles	2.60	square kilometers	km^2
	acres	0.40	hectares	ha
		MASS (weight)		
oz	ounces	28.00	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.90	tonnes	t
		VOLUME		
tsp	teaspoons	5.00	milliliters	ml
Tbsp	tablespoons	15.00	milliliters	ml
fl oz	fluid ounces	30.00	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.80	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.40	inches	in
m	meters	3.30	feet	ft
m	meters	1.10	yards	yd
km	kilometers	0.60	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.20	square yards	yd^2
km^2	square kilometers	0.40	square miles	mi^2
ha	hectares ($10,000\ m^2$)	2.50	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.10	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	36.00	cubic feet	ft^3
m^3	cubic meters	1.30	cubic yards	yd^3
		TEMPERATURE		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



* 1 in. = 2.54 cm (exactly)

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
DOT/FRA/ORD-01/14		

4. Title and Subtitle	5. Report Date
Effects of Coupler Height Mismatch on the Structural Integrity of Railroad Tank Car Stub Sills	December 14, 2001
	6. Performing Organization Code
7. Author(s) Anand Prabhakaran, Vinaya Sharma	8. Performing Organization Report No.
9. Performing Organization Name and Address Sharma & Associates, Inc. 5810 S Grant Street Hinsdale, IL 60521	10. Work Unit No. (TRAIS)
	11. Contract or Grant No. DTFR53-98-C-00012
12. Sponsoring Agency Name and Address USDOT/Federal Railroad Administration Office of Research & Development, RDV-32 1120 Vermont Ave., NW, Mail Stop 20 Washington, DC 20590	13. Type of Report or Period Covered 11/1998 to 11/2000
	14. Sponsoring Agency Code RDV 32

15. Supplementary Notes

16. Abstract

This project evaluated the safety implications of coupler height mismatches on the integrity of railroad tank car stub sills, through a series of static and impact tests. The test car was a loaded tank car instrumented with strain gages at critical locations. Static tests consisted of squeeze tests and vertical load tests. Impact tests using the test car as the struck car, and a 263 kip hopper car as the hammer car, consisted of impacts at speeds ranging from 2 mph to 8.5 mph for three (3) coupler mismatch levels: a) coupler heights at the same level, b) test car coupler 2" higher, and c) test car coupler 2" lower. The static tests showed that a 50,000 lb vertical force generated as much stress as a 680,000 lb longitudinal force, on average. This implies that moderately high longitudinal forces, in conjunction with moderately high vertical forces can cause significantly high stresses. The impact results indicate that significant coupler vertical forces are generated even in the absence of any coupler height mismatch. However, vertical forces and stresses in critical areas were generally higher when the struck car was 2" lower. Vertical forces induce 50 % of the maximum stresses seen at the tank head, and about 44 % of the maximum stresses seen at the head pad extension, under impact conditions. Therefore, ignoring the effects of vertical force will lead to an under-estimation of stress levels and fatigue levels in tank cars.

17. Key Words	18. Distribution Statement
Tank Car, Stub Sill, Impact, Coupler Height Mismatch, Draft Gears, Vertical Forces, Buff Loads, Draft Loads, Fatigue, Tank Car Safety, Track Stiffness Transitions, Impact Simulations	This document is available to the Public through the National Technical Information Service, Springfield, Virginia 22161

19. Security Classification (of the report)	20. Security Classification (of this page)	21. No. of Pages	22. Price
Non classified	Non classified	49	

Form DOT F 1700.7 (8-72)

TABLE OF CONTENTS

LIST OF FIGURES

Page #

EXECUTIVE SUMMARY

Revenue service tests of tank cars conducted by the Association of American Railroads (AAR) as part of the 'Freight Equipment Environmental Sampling Tests' (FEEST) program have measured high coupler vertical loads. Such high vertical loads have a significant effect on the operational safety of these cars, especially those carrying Hazardous Materials (HM). These high loads may induce cracks and/or accelerate crack propagation rates, causing premature stub sill separation, leading to compromised safety of operation. It is also possible that the cracks initiated at the stub sill / tank head welds may propagate into the tank shell causing catastrophic tank failure. It is suspected that coupler height mismatches are a major source of these coupler vertical forces. Many possible causes of coupler height mismatch have been identified, including train consist variations, train handling, track stiffness transitions, car/coupler geometry, curve geometry, variations in car load levels, use of multiple wear wheels, etc. The main objective of this project was to study the effects of coupler height mismatches on stub sill integrity. The second objective was to examine the role of draft gear characteristics and classification on the impact protection offered by draft gears. The objectives were accomplished using a full scale test sequence with static and impact testing of a tank car, followed by test data analysis to draw conclusions from the test effort.

The test car was a non-insulated GATX built tank car with a built date of 1/81 and a DOT classification of 111A100W1. The 'A' end of the car was instrumented with strain gages at critical locations on the stub sill, tank shell, reinforcement pad and the head brace, and the car was loaded with water to weigh 266,200 lb. The static tests consisted of longitudinal force tests in a squeeze fixture (at three different coupler heights), and vertical force tests with loads applied to the striker or the coupler head. Impact tests using the tank car as the struck car, and a 263,000 lb hopper car as the hammer car, consisted of two series of impacts at speeds ranging from 2 mph to 8.5 mph for the following coupler mismatch levels: a) Struck and striking car at the same level, b) Struck car 2" higher, and c) Struck car 2" lower. One impact series was performed using the 'premium' M-901 E draft gears that came with the test car, and the other using a pair of new M-901 G gears.

Significant vertical forces are generated during an impact event, even in the absence of any coupler height mismatch between impacting cars. On average, vertical force peaks of about 45,000 lb were observed at an impact speed of 7.5 mph even when the impacting cars were at the same level.

It was also observed these coupler vertical forces induce 50 % (average) of the maximum stress seen at the tank head, and 44 % (average) of the maximum stress seen at the head pad extension, under impact conditions. Ignoring the effects of vertical force will lead to an under-estimation of stress levels and fatigue levels in tank cars. Therefore, safety evaluations of stub sill-tank cars must include vertical force effects simultaneously with longitudinal force effects.

During an impact event, an initial upward vertical force is generated which is followed by a downward vertical force. The downward vertical force is generally more critical since it adds to the tensile stress levels created by the longitudinal buff force. The coupler height mismatch level between impacting cars has a significant effect on the vertical force levels during an impact. The downward vertical force tends to be higher when the struck car is lower than the hammer car. Also, stresses in critical areas of the stub sill-tank car interface are generally higher when the struck car is lower. These higher vertical force levels and higher stress levels at critical locations increase the risk of crack initiation and accelerated crack propagation, which could lead to premature stub sill separation or loss of lading, thereby seriously affecting operational safety. In light of this, it is recommended that the coupler height of a tank car be maintained no lower than the coupler heights of adjacent cars, whenever and wherever feasible.

The static tests established the criticality of vertical forces on the structural integrity of the stub sill-head brace area of tank cars. On average, a 50,000 lb vertical force can generate as much stress as a 680,000 lb longitudinal force, under static conditions. This implies that moderately high longitudinal forces, in conjunction with moderately high vertical forces can cause significantly high stresses, which may be near or above the yield point of the tank material.

It was observed during the tests that 'G' gears closed out at higher impact speeds (8.0 - 8.5 mph) than did the 'premium E' gears (7.5 - 8.0 mph). However, there were no significant differences in performance between the 'premium E' gears and the 'G' gears that were tested. While, the 'G' gears performed slightly better in some cases and the 'premium E' gears performed better in other cases, the test results did not offer a convincing argument for choosing 'G' gears over the 'premium E' gears. However, revenue service environment is different and is better characterized by lower force levels and lower run-in/run-out speeds. Hence, the above conclusion may not be applicable. It should be emphasized here that the 'G' gear that was tested was brand new, as compared to the 'premium E' gear which was already broken in. Generally, draft gears tend to perform better after they have been broken in. Also, the results could be very different when the comparisons are made with older/used E gears. The results could also be different if 286,000 lb tank cars are considered.

ACKNOWLEDGEMENTS

The authors wish to thank the Office of Research & Development of the Federal Railroad Administration, in particular Ms. Claire Orth and Mr. Jose S. Peña, for funding this research effort. We also wish to thank Mr. Jim Rader of the Office of Safety, Federal Railroad Administration for his valuable input into the program. Our sincere thanks are also due to GATX, especially Mr. William Garfield and Mr. Doug Mullins, for providing the test car. The authors also wish to thank Dr. Gerald J. Moyar and Mr. Samuel P. Halcomb for their invaluable guidance.

1. INTRODUCTION

1.1 Background

In an attempt to better represent the over-the-road (OTR)/revenue service environment of freight cars during the design process, the Association of American Railroads (AAR) conducted a series of tests wherein, instrumented freight cars (hoppers, tank cars, box cars, etc.) were inserted in revenue service train consists and monitored to evaluate the OTR load environment. These tests were conducted as part of the 'Freight Equipment Environmental Sampling Tests' (FEEST) program.

Revenue service tests of a tank car conducted under the FEEST program have measured high coupler vertical loads [1-3]. Such high vertical loads can have an adverse effect on the fatigue performance of tank car stub sills, leading to crack initiation, accelerated crack propagation, and premature stub sill separation, thereby seriously affecting operational safety. It is also possible that cracks initiated in the stub sill may propagate into the tank shell causing catastrophic tank failure. These safety concerns are particularly important for tank cars transporting hazardous materials.

It is suspected that coupler height mismatches are a major source of these coupler vertical forces, thereby contributing to stub sill failures, stub sill separations, lading release, etc. The coupler height of a freight car (from top of rail) can be between 31.5" & 33.5" if the car is loaded, or between 32.5" & 34.5" if the car is empty (Rule 16, E.3 of the AAR Field Manual). This implies that a static coupler height mismatch (measured when the cars are stationary in a yard) of up to 3" between adjacent cars is permissible in regular interchange service. These static variations in coupler height can result from variations in load levels, use of multiple wear wheels, coupler carrier wear plate conditions, truck spring variations, worn components, etc. Potentially, such variations could be amplified dynamically over the road due to train consist variations, train handling, track stiffness transitions, curve geometry, etc. Furthermore, OTR dynamic events can introduce coupler height mismatches between adjacent cars over-the-road, even if the cars were perfectly aligned during inspection in a train make-up yard.

Another variable that needs study is the effect of draft gear performance on the performance of stub sills, specifically in relation to coupler height mismatch. In addition, it was desired to study if draft gears validated under the M-901 G specifications would provide any significant benefits in terms of operational safety.

1.2 Objectives

The main objective of this project was to study the safety implications of coupler height mismatches on stub sill-tank cars. This was to be accomplished using a full scale test sequence with static and impact testing of a tank car, followed by test data analysis to draw conclusions from the test effort. Also, analytical studies were to be conducted to help pin-point the reasons/sources of coupler height mismatch. Another objective of the project was to analyze the role of draft gear design and capacity, especially as applied to tank cars with Gross Rail Loads (GRL) exceeding 263,000 lb in hazardous materials service.

This was to be accomplished as part of the test effort to determine the relative performance of draft gears specified under M-901 E and M-901 G specifications.

2. TESTING

The test plan was to instrument a test tank car so that strains (and consequently, stresses) at critical locations could be monitored. This was to be followed by a series of vertical and longitudinal static load tests to evaluate tank car response to these loads and for calibration purposes. This was to be followed by a series of specifically designed impact tests to evaluate tank car response to different levels of coupler height mismatch and different draft gear types. The test tank car was generously donated by GATX.

2.1 Test Car Setup

The test car is a non-insulated GATX built tank car (GATX 15693) with a built date of 1/81 and a DOT classification of 111A100W1. It has a capacity of 26,791 gallons, with a light weight of 70,200 lb and a load limit of 192,800 lb. The stub sill consists of 2 channels and a top plate. The sill connects to the tank through a head brace, and near the bolster (figure 1). The head brace is connected to the tank shell though a reinforcement pad (head pad extension). The front draft lugs and the striker are a single casting that is welded on to the web of the sill. The rear draft lugs and the center-bowl are a single casting that is bolted on to the web of the sill. The material of tank head, tank shell, and head pad extension is ASTM A515, Gr. 70 LR (yield point (yp) : 38,000 psi). The head brace material is ASTM A36 (yp : 36,000 psi) and the draft sill material is ASTM A572, Gr. 50 (yp : 50,000 psi).

For both static and impact tests, the test car was loaded with water to weigh 266,200 lb. The 'A' end of the car was instrumented with strain gages at critical locations on the stub sill, tank head, reinforcement pad and head brace. Gage locations were determined based on preliminary finite element analysis, consultations with industry experts, and an industry survey of fatigue crack locations on cars of similar design. Care was taken to place strain gages away from stress concentrations, so that strains in the parent material would be measured. Thirty single element and four rectangular rosette strain gages, forty two (42) channels total, were applied as part of the instrumentation. Figures 2 & 3 represent the gage locations (listed in table 1). The testing was conducted by Miner Enterprises, Inc., Geneva, Illinois.

2.2 Static Tests

To understand the effect of coupler height mismatch on tank cars, the performance of the test car under longitudinal and vertical loading had to be evaluated. For this purpose, the following tests were conducted.

- Static Compressive End Load (Squeeze Test):
  - Loading applied at standard height
  - Loading Center Lowered 2"
  - Loading Center Raised 2"
- Coupler Vertical Load: Down & Up
- Coupler Shank Test

The following passages describe these tests in greater detail.

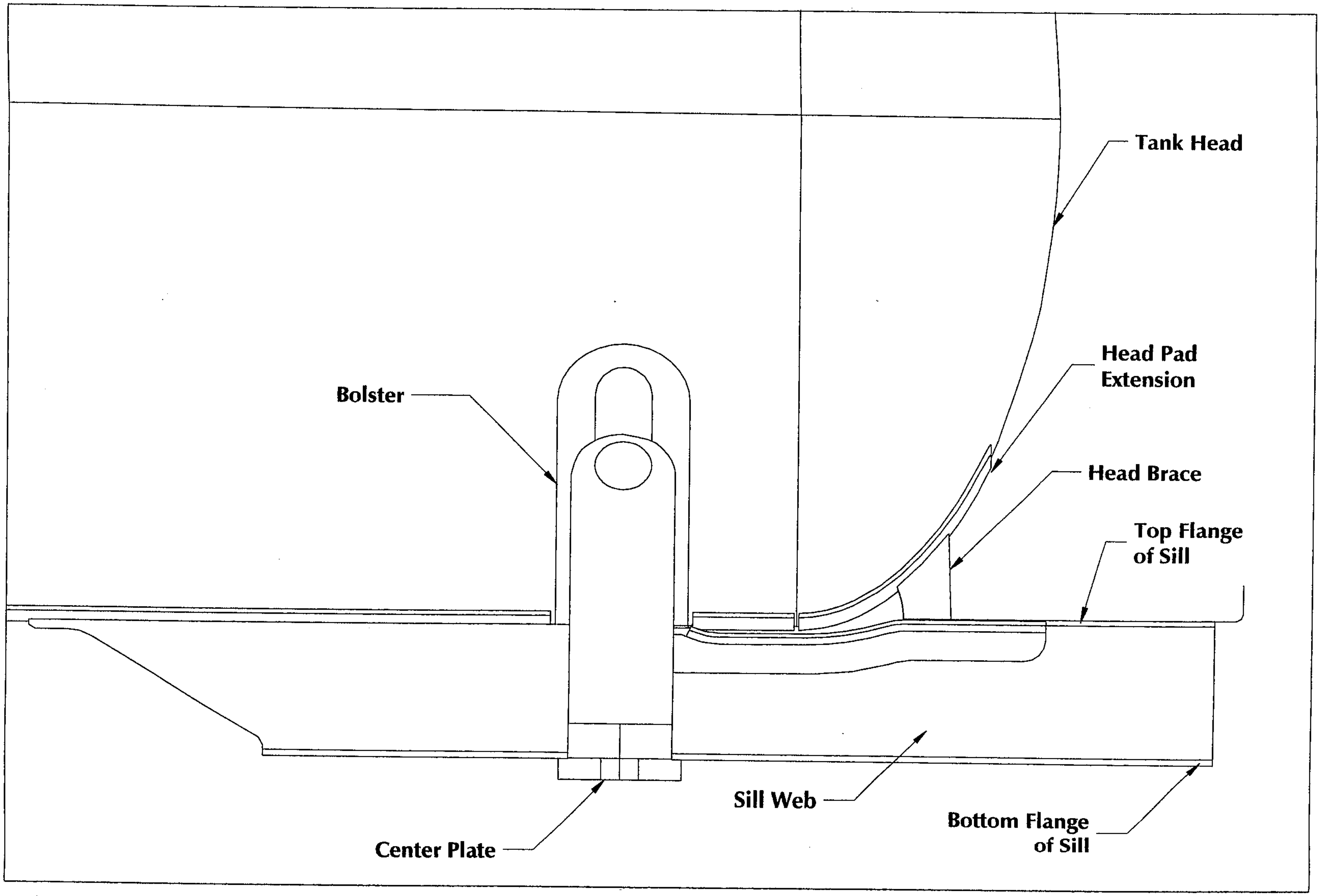


Figure 1. Schematic of test tank car

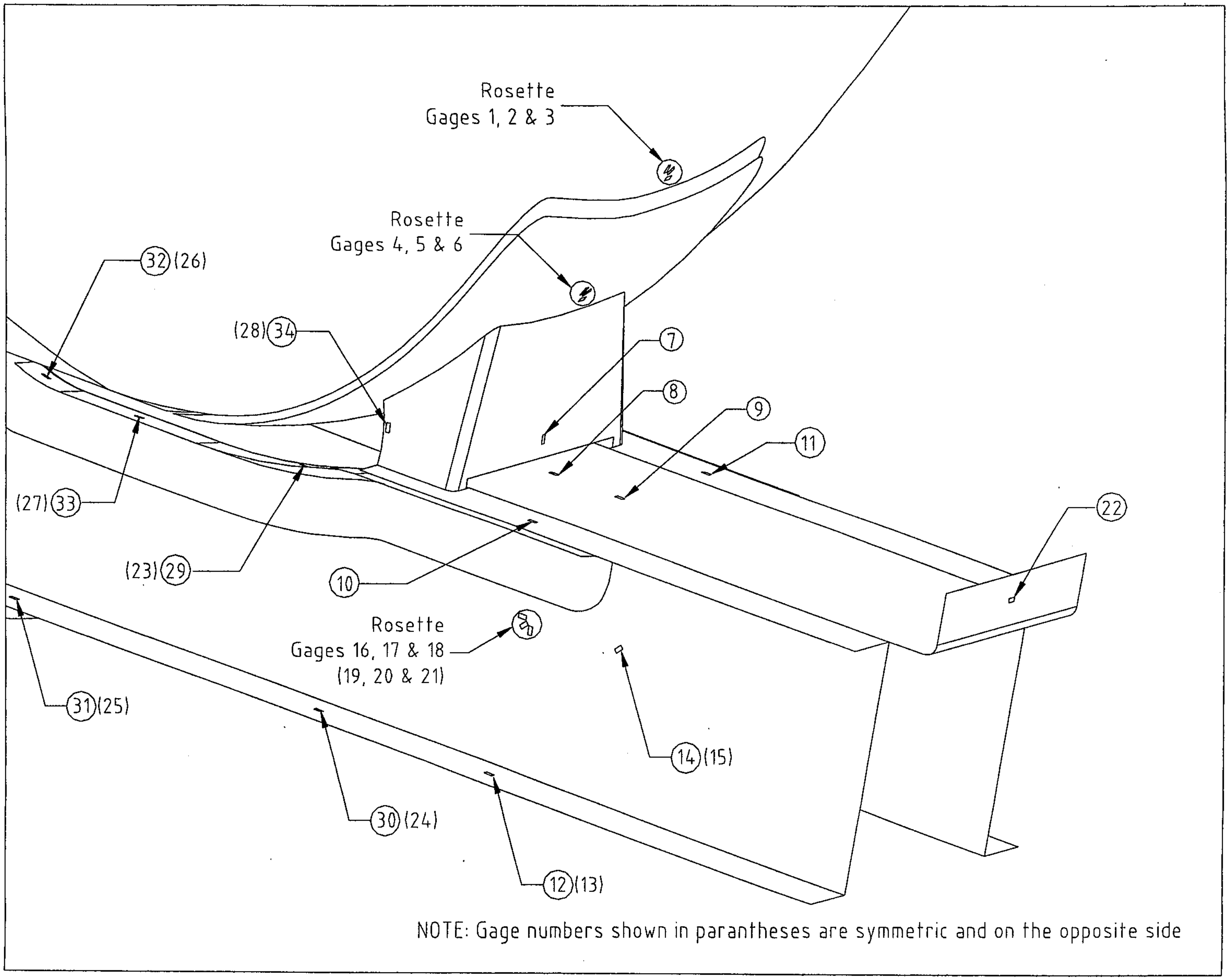


Figure 2. Gage locations on test car - View 1

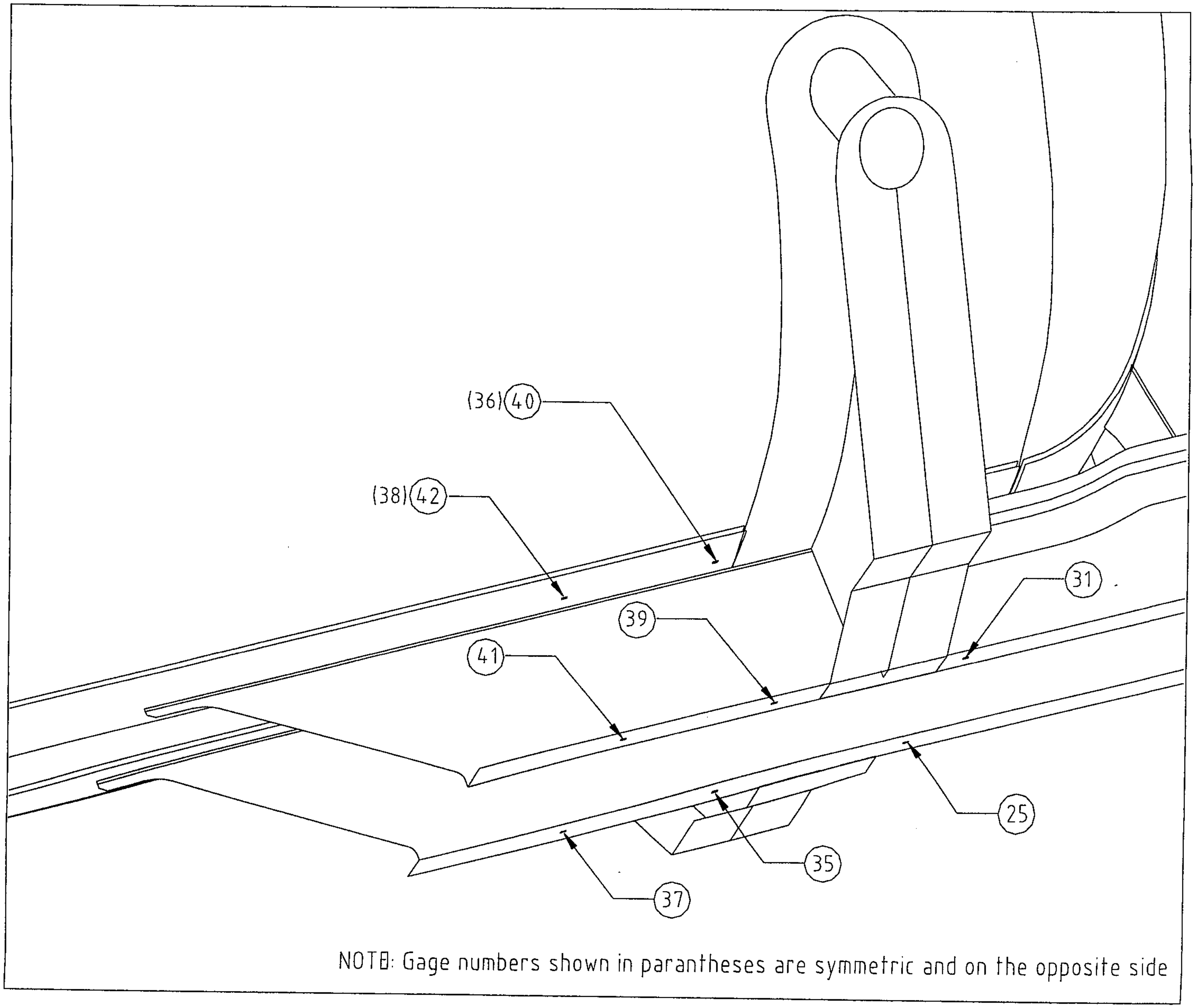


Figure 3. Gage locations on test car - View 2

Table 1. Strain Gage Locations

Gage No.	Location
1, 2, 3	Tank, above head pad extension - Rosette
4, 5, 6	Head Pad Extension - Rosette
7	Head Brace - Center - Vertical
8	Top of Sill - Along CL of car
9	Top of Sill - Along CL of car
10, 11	Top of Sill - Along the Edge - Right side, Left side
12, 13	Bottom of Sill - Along the Edge - Right side, Left side
14, 15	Shear Gage of Sill Web - Right side, Left side
16, 17, 18	Sill Web @ Neutral Axis - Rosette - Right Side
19, 20, 21	Sill Web @ Neutral Axis - Rosette - Left Side
22	Gage on Striker
23	Top Flange of Sill - Inboard of head-brace - Left Side
24	Bottom Flange of Sill - Inboard of head-brace - Left Side
25	Bottom Flange Of Sill - Near Bolster - Left Side
26	Top Flange of Sill - Near Bolster - Left Side
27	Top Flange of Sill - between head brace & bolster - Left Side
28	Edge of Head Brace - Left Side
29	Top Flange of Sill - Inboard of head-brace - Right Side
30	Bottom Flange of Sill - Inboard of head-brace - Right Side
31	Bottom Flange Of Sill - Near Bolster - Right Side
32	Top Flange of Sill - Near Bolster - Right Side
33	Top Flange of Sill - between head brace & bolster - Right Side
34	Edge of Head Brace - Right Side
35	Bottom Flange of Sill - Near Bolster - Left Side
36	Top Flange Of Sill - Near Bolster - Left Side
37	Bottom Flange of Sill - Inboard of Bolster - Left Side
38	Top Flange of Sill - Inboard of Bolster - Left Side
39	Bottom Flange of Sill - Near Bolster - Right Side
40	Top Flange Of Sill - Near Bolster - Right Side
41	Bottom Flange of Sill - Inboard of Bolster - Right Side
42	Top Flange of Sill - Inboard of Bolster - Right Side

2.2.1 Static Compressive End Load (Squeeze Test)
For this test both draft gears that came with the car were replaced with nullified draft gears. The nullified draft gears were constructed by welding a type 'F' follower plate to the top of a standard draft gear cylinder. A plunger fabricated from a type 'F' coupler shank was used to apply the test load to the solid draft gear. This combination of type 'F' coupler shank and follower plate simulated the AAR required ball end restraint. The truck springs were blocked to align the test car with the test fixture loading ram. The compressive load was provided by a system of five hydraulic cylinders controlled by five control valves, to ensure an even and progressive loading. Load was measured using a precision load cell located between the cylinder ram and the test car plunger. Figure 4 shows the tank car in the squeeze fixture.

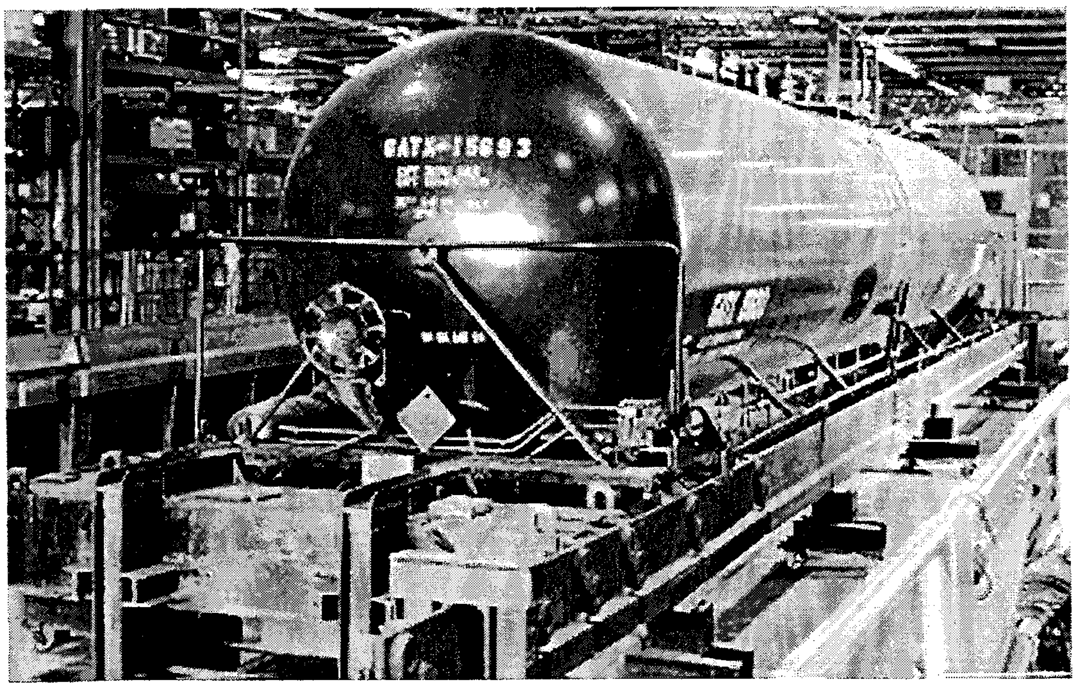


Figure 4. Tank car in the squeeze frame

These tests were performed at three different loading heights. For the standard height tests, the longitudinal squeeze load was applied directly at the center line of draft. For the next series, the loads were applied 2" below the centerline of draft, at the rear lugs. This was accomplished by adding shims to the lower portion of the draft gear housing base. For the third series the loads were applied 2" above the centerline of draft, at the rear lugs, accomplished by adding shims to the upper portion of the draft gear housing base.

In all cases, output from all forty two (42) strain channels, the load cell and the two displacement transducers were monitored. Test loads were applied in 100 kip increments up to 800 kips and removed in 100 kip increments to zero load. The 800 kip loading series was performed twice for each loading condition. Test data for this and other static tests were recorded using an Optim Corporation, Megadac 200, digital data acquisition system.

2.2.2 Coupler Vertical Load Tests

Coupler vertical load tests were performed on the loaded test car, with vertical loads being applied to the coupler head near the pulling face. Loads in the downward direction were applied using a hydraulic jack and a load frame and measured using a load cell located between the jack and the frame. Figure 5 shows the test setup. Loads were applied in 10 kip increments up to 50 kips and then released to zero load in 10 kip increments. This test sequence was performed twice, followed by another test series with loads applied in the upward direction. All loads were applied at the "A" end (instrumented end) of the test car. Output from the load cell, two displacement transducers, and all forty two (42) strain gage channels, was recorded at each load increment. One displacement transducer was located on the bottom of the striker. The second was located 36″ inboard of the striker.

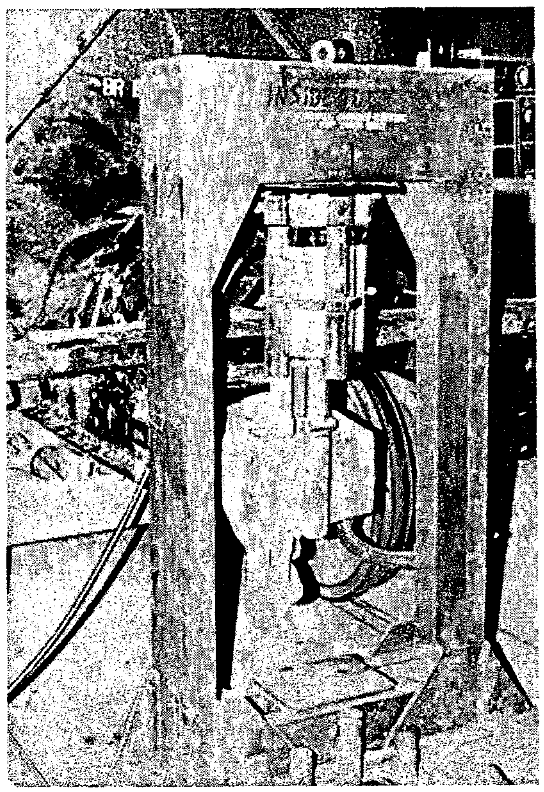


Figure 5. Coupler vertical load test

2.2.3 Coupler Shank Test

For this test, a vertical upward load was applied to the coupler shank immediately adjacent to the striker face, sufficient in magnitude to lift the fully loaded car free of the truck nearest to the applied load (approximately 125 kips). Test loads were applied in 25 kip increments and released to zero load after the car was free from the truck. All loads were applied at the "A" end using a hydraulic jack. Output from the load cell, displacement transducers, and all forty two (42) strain gage channels, was recorded at each load increment. The displacement transducers were located in the same positions as in the coupler vertical load tests.

2.3 Impact Tests

The objective of the impact tests was to measure tank car response under two different draft gear sets and three different coupler height mismatch cases at different impact speeds. The draft gear sets were:

- a set of premium M-901 E draft gears that came with the car
- a set of new M-901 G draft gears

For both sets of draft gears, impact testing with the tank car as the struck car was conducted at three different coupler height mismatch levels :

- Tank car & hammer car at the same level
- Tank car 2" higher, and
- Hammer car 2" higher

The hammer car was a hopper car loaded to weigh 263,000 lb. There were 2 backup cars behind the tank car weighing a total of 397,000 lb, with hand-brakes applied on the last car (figures 6,7). The tank car, hammer car and backup cars were all equipped with 'E' couplers. The hammer car was accelerated to different impact speeds using an inclined test track (impact ramp). Desired impact speeds were obtained by releasing the hammer car from different heights on the impact ramp. The 'A' end of the tank car was the struck end. The hammer car used a set of M-901 E draft gears for all the tests. The tank car and the backup cars were bunched together after each impact. Height mismatches between cars were achieved by replacing one of the inner truck springs of either the hammer car or the tank car (struck car) with a solid block.

Based on the results of the static tests, 34 channels of strain data were selected for measurement during the impact tests. In addition, coupler force (dynamometer coupler in the hammer car), draft gear travel (struck end of the tank car), and impact velocity were also measured. Also, 4 channels of acceleration data were recorded. Two accelerometers

Figure 6. Cars on impact test track

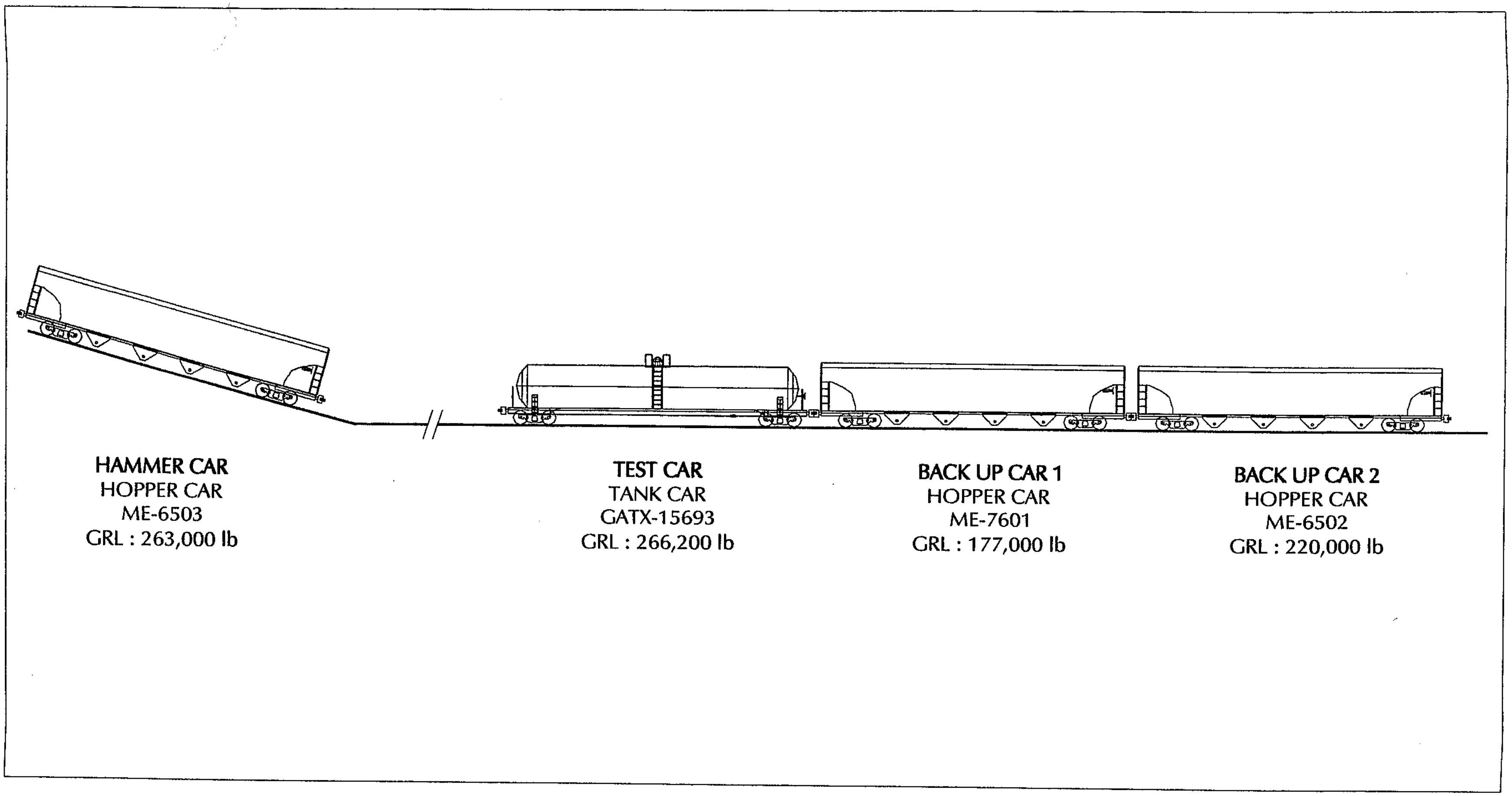


Figure 7. Schematic of impact test setup

Figure 8. Hammer car on impact ramp

were mounted on the striker and the head brace to measure accelerations in the vertical direction, and two were mounted on the web of the stub sill (one inboard & the other outboard of the bolster) to measure longitudinal accelerations.

For each test series, the impact tests started with an impact velocity of about 2 mph. Impacts were continued with increasing impact speed until the draft gears on the tank car bottomed out. This occurred at speeds between 7.5 and 8.5 mph. A total of 58 impacts were conducted.

3. Test Data Analysis

The large volumes of data generated in the course of both the static and impact tests were analyzed in detail to obtain a better understanding of the structural behavior of the tank car, especially the stub sill area. This section describes the data analysis procedures and the results derived from the test program.

3.1 Static Tests

The strain data from Miner was converted to stress data, including evaluation of principal stresses wherever rosettes were used. The static testing helped us calibrate two gages on the sill web (14 & 15) for measuring vertical forces under impact conditions while also establishing the significance of longitudinal & vertical forces on the structure of the tank car. The variation in stub sill - tank car stresses with variation in the height of application of buff forces was also studied using the results from the static tests. Results from the static testing were also used to calibrate the finite element model of the tank car that was developed. The following passages describe the test results.

3.1.1 Calibration

The primary calibration efforts were aimed at determining the ability of the gages on the web of the stub sill to measure vertical forces applied at the coupler head. Gages 14 & 15 (see table 1, figure 2) were mounted on either side of the stub sill near the neutral axis, outboard of the draft lugs, with a 45° inclination to the horizontal. This setup was designed to provide an estimate of the shear at the section, which would be representative of the vertical force level. Also, since the gages were outboard of the lugs (figure 11), they were not affected by the longitudinal force levels seen by the lugs.

Gages 14 & 15 were calibrated using test data from the static vertical force tests. Figures 9 & 10 show the linearity of gage behavior. The linear fit for both gages had a correlation coefficient greater than 0.99, indicating an excellent fit. The next step was to evaluate cross-talk levels of these gages (14 & 15) under longitudinal forces. The output of the gages under the longitudinal forces applied during the squeeze test was also studied. This curve was almost flat (very small slope) indicating the lack of response of the gages to applied longitudinal force. However, this small slope was taken into account when evaluating vertical force levels during the impact tests. The high level of correlation with applied vertical forces and negligible response to applied longitudinal forces indicates that these gages can be reliably used to measure vertical forces under impact conditions.

Gages on the tank head, head pad extension and head brace (locations A, B & C - figures 11 & 12) were also calibrated to evaluate their response to vertical and longitudinal forces. Unlike gages 14 & 15, these gages respond to both vertical & longitudinal forces. Also, the gages at locations A & B were rosettes. By knowing how these gages respond to vertical or longitudinal force, one can separate the vertical force effects from the longitudinal force effects during impact situations, where both force types are present simultaneously. The calibration graphs for location A (gages 1, 2, 3) and location B (gages 4, 5, 6) are shown in figure 13. All these graphs were fit using linear curves with high correlation coefficients.

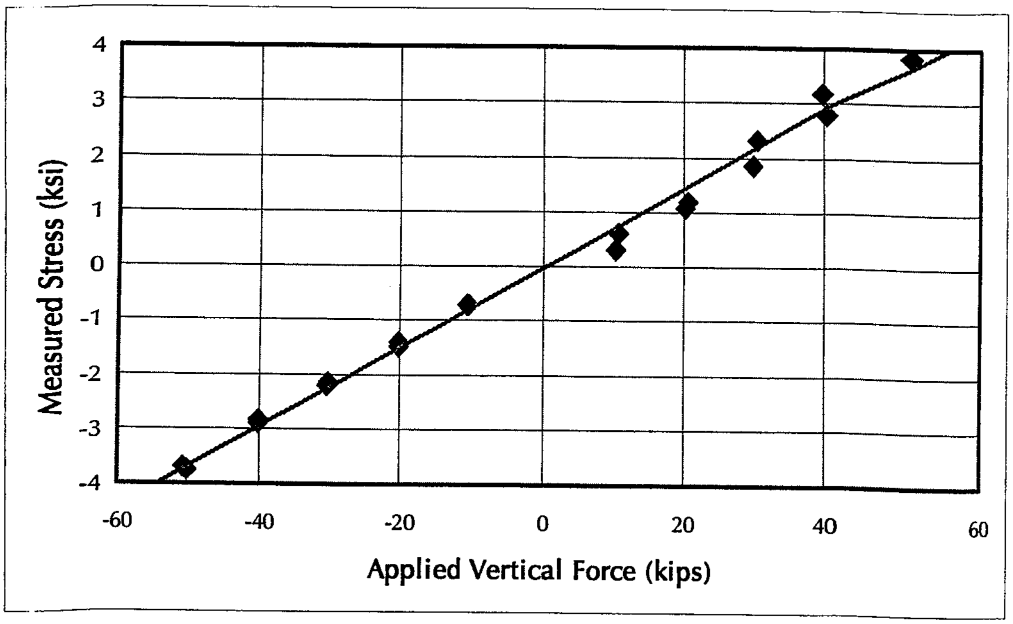


Figure 9. Vertical force calibration - Gage 14

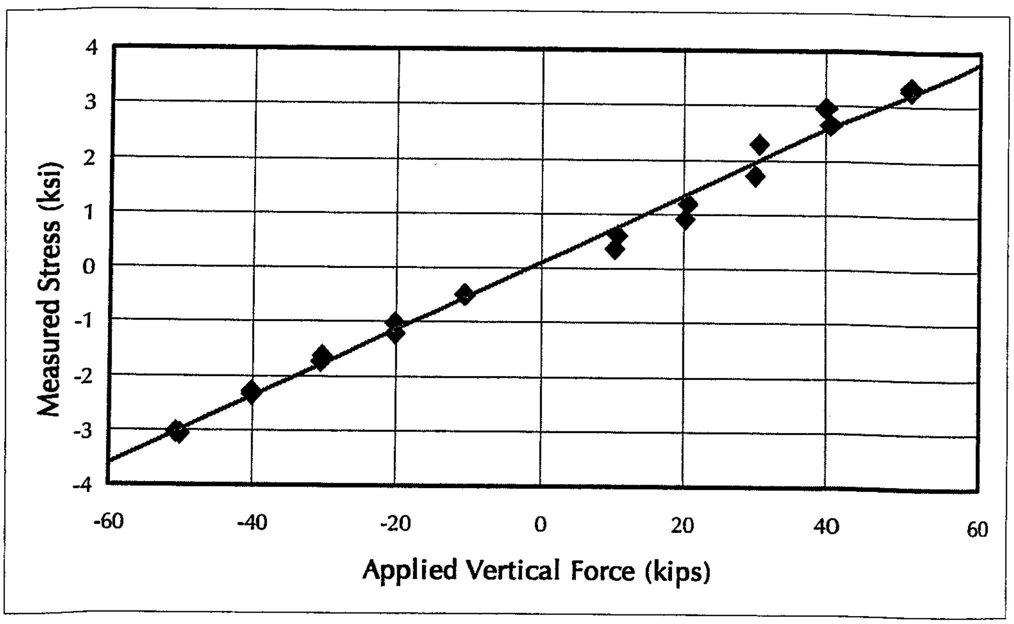


Figure 10. Vertical force calibration - Gage 15

3.1.2 Squeeze Test Results

For our discussion of test results, we will concentrate on stresses at three critical points on the car structure : the tank head (shell), the head pad extension, and the head brace. The three locations are marked as A, B and C respectively in figures 11 & 12.

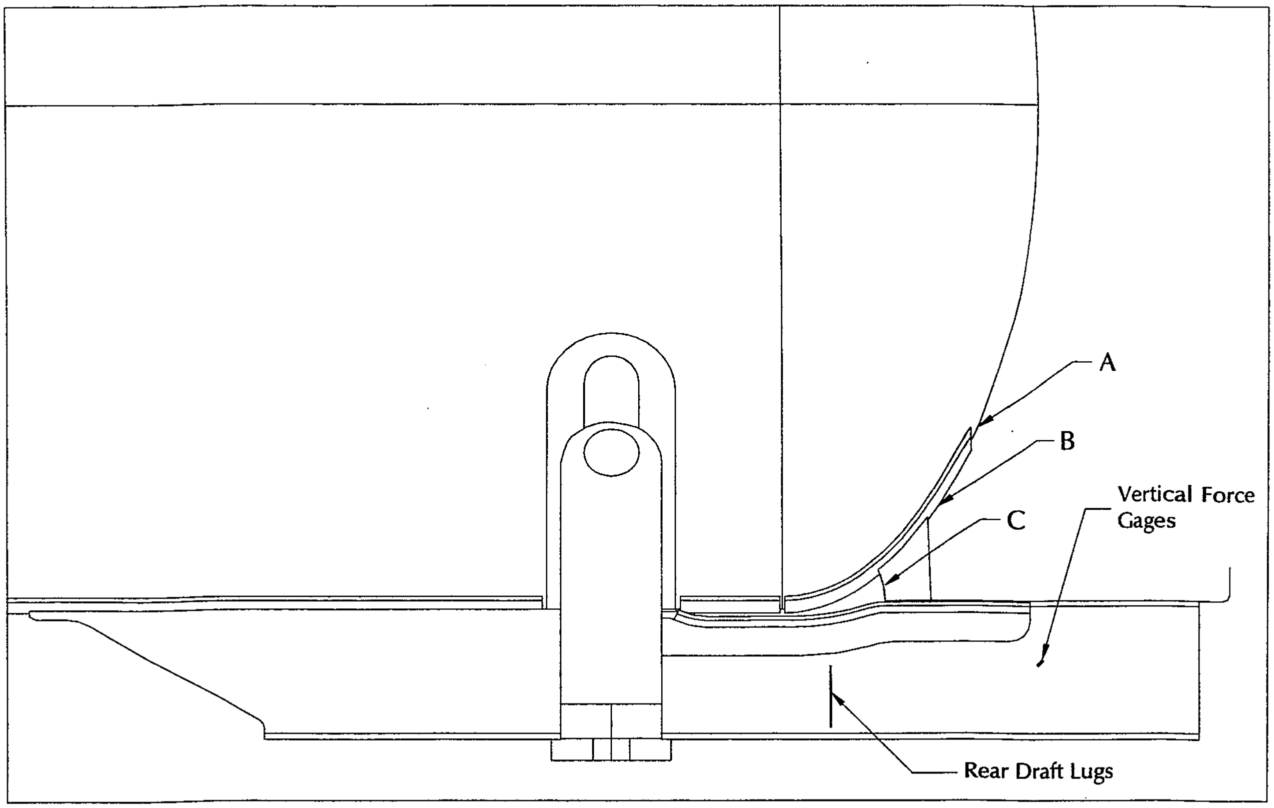


Figure 11. Critical locations - Elevation

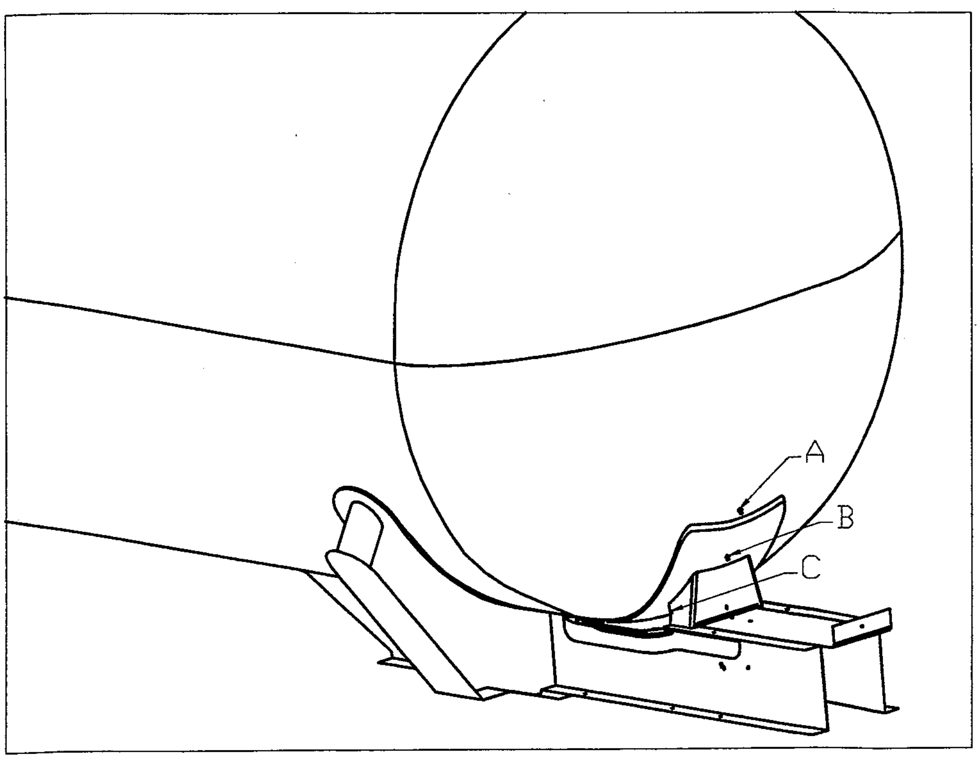


Figure 12. Critical locations - Isometric view

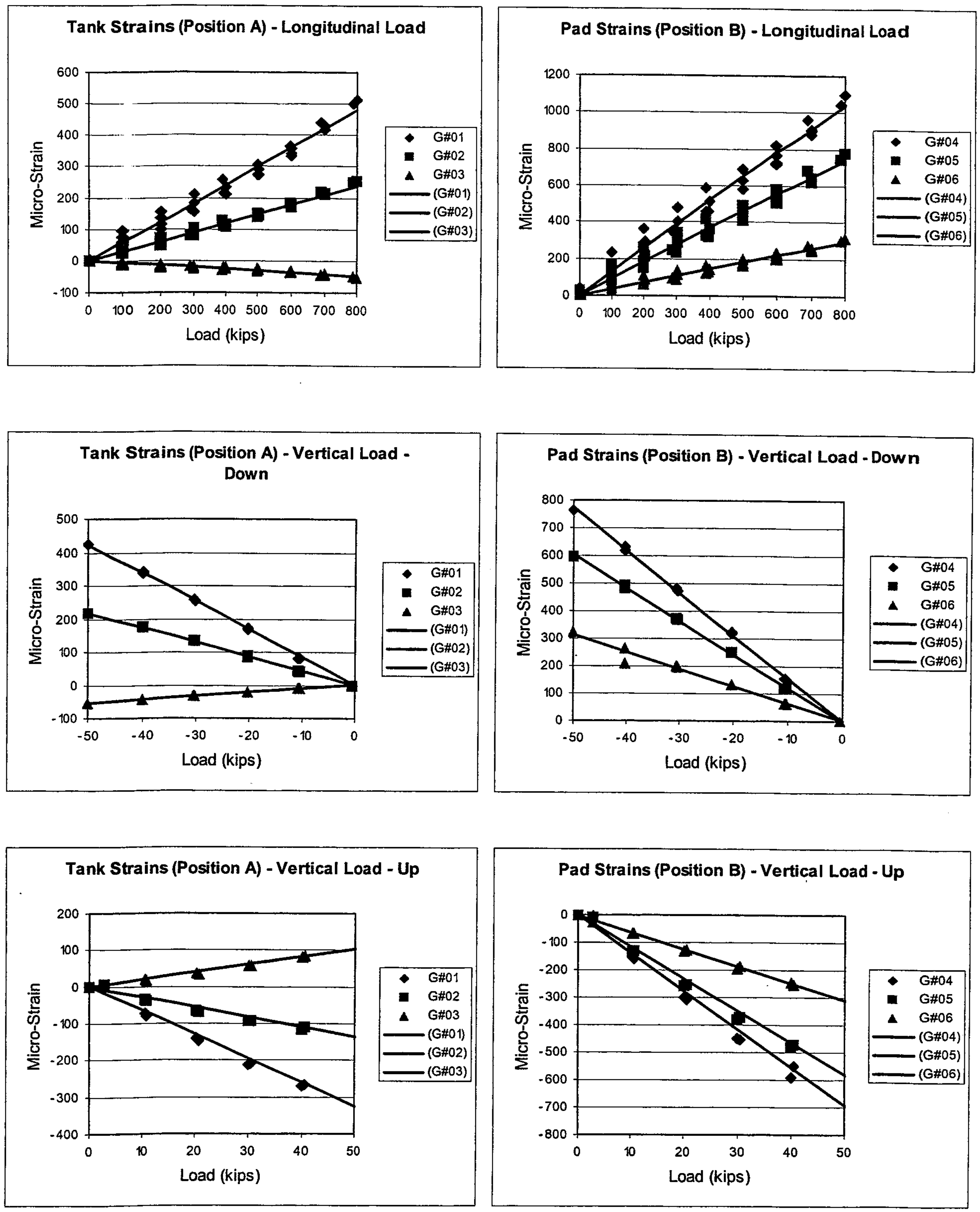


Figure 13. Calibration charts - Longitudinal & vertical forces - Locations A & B

The stresses measured on the tank head, the head pad extension and the head brace under the 800,000 lb squeeze load are tabulated in table 2. Tensile stresses are positive and marked (t), while compressive stresses are negative and marked (c). It is seen that fairly high stresses are induced in the stub sill under a 800,000 lb buff force. While such high buff forces are uncommon in a train, they are seen frequently in hump yards. The stress levels might increase even more when these high longitudinal forces occur simultaneously with high vertical forces.

It is also seen that the stresses in the tank head (A) and the head pad (B) vary significantly with change in the loading line. Lowering the loading line by 2" increased the stresses in the tank head and the head pad by 46 % and 44 % respectively (see table 2). However, the stresses in the head brace changed by only 8 %. Alternately, raising the loading line by 2" increased the stresses in the tank head and the head pad by 18 % and 13 % respectively, with the stresses in the brace changing by under 2 %. It should be noted here that these changes in the loading line were created by applying an off-center squeeze load directly on the rear draft lugs (see section 2.2.1). Such a loading scenario is not very likely under regular operating conditions. In normal operation, the loading offset will be at the coupler head, not at the draft lugs. However, this test represents an extreme case of offset and helps us better understand the effects of load offset.

Table 2. Stresses under a 800,000 lb squeeze load

Point of load application	Tank Head (A)	Head Pad Ext. (B)	Head Brace (C)
@ Center-Line (CL) of draft	16.4 ksi (t)	39.4 ksi (t)	-33.4 ksi (c)
2" below CL of draft	23.9 ksi (t)	56.9 ksi (t)	-30.7 ksi (c)
2" above CL of draft	19.3 ksi (t)	44.6 ksi (t)	-33.9 ksi (c)

3.1.3 Significance of Coupler Vertical Forces

To establish the significance of vertical forces, emphasis was placed on stresses at the same three critical points on the car structure : the tank head (shell-A), the head pad extension (B), and the head brace (C) (see figures 11 & 12).

Stresses measured under the vertical forces applied at the coupler head are shown in table 3. As indicated, there is some difference in stress levels, depending on whether forces are applied in the upward or downward direction. A maximum stress of 32.3 ksi (t) was observed at the head brace location under the downward 50,000 lb force. It is seen that forces in the downward direction are more severe (higher stresses), and cause tensile stresses in critical areas (locations A & B).

Given that squeeze loads (table 2) also cause high tensile stresses in locations A & B, a combination of buff loads and vertical (downward) forces has the potential to cause very high tensile stresses at these critical locations.

Table 3. Stresses for a 50,000 lb vertical load applied at the coupler head

Loading Direction	Tank Head (A)	Head Pad Ext. (B)	Head brace (C)
Downward	13.8 ksi (t)	29.5 ksi (t)	32.3 ksi (t)
Upward	-9.5 ksi (c)	-27.0 ksi (c)	-28.2 ksi (c)

The criticality of vertical forces is further emphasized when the magnitudes of stresses under vertical loads are compared to the stress magnitudes under horizontal loads. A 50,000 lb vertical force causes as much stress as a 670,000 lb longitudinal force (applied at the CL of draft) on the tank head. The same vertical force is equivalent to a 600,000 lb longitudinal force when stresses on the head pad extension are compared. It is also equivalent to a 770,000 lb longitudinal force when stresses on the head brace are compared. On average for the three locations, a 50,000 lb vertical force is equivalent to a 680,000 lb longitudinal force. This implies that high longitudinal forces, in conjunction with high vertical forces can cause significantly high stresses, which may be near or above the yield point of the tank material. This is discussed in more detail in section 3.2.3.

The static tests established the criticality of vertical forces on the structural integrity of the stub sill-head brace area. In addition, the importance of longitudinal forces applied off the centerline of draft were established. Also, the tests helped calibrate the gages on the sill such that the vertical forces generated under impact conditions could be estimated.

3.2 Impact Tests

Large volumes of data were collected during the impact tests. In addition to the 22 channels of strain data, coupler load and draft gear travel data collected by Miner, 12 channels of strain data and 4 channels of accelerometer data were collected using a data acquisition system belonging to Sharma & Associates, Inc. All this raw data was further processed using some data processing software routines that were developed in-house. These routines filter the data as needed and convert it into a useful and readable format, while also converting the measured strain data into stress data, including the calculation of principal stresses wherever rosettes were used. Zero-balancing of the data is also done during this process.

The impact tests were used to evaluate the effects of different variables on the stress levels seen by the tank car's stub sill. The studies included evaluations of the effects of 1) coupler height mismatch, 2) draft gear type, and 3) vertical forces. These studies are discussed in greater detail in the following passages.

3.2.1 Effect of Coupler Height Mismatch

As previously mentioned, impact tests were conducted at three different coupler mismatch levels; 1) Tank car & hammer car at the same level, 2) Tank car 2" higher, and 3) Hammer car 2" higher. The height mismatches were measured at the coupler head. The effects of coupler height mismatch on both vertical force levels at the coupler head and on stress levels on the stub sill were studied.

The vertical force levels at the coupler head were measured by using strain gages on the sill which were calibrated during the static testing (see section 3.1.1). Figure 14 shows a characteristic trace of coupler longitudinal force and vertical force(x 10) that were developed during a 7.5 mph impact. This particular impact was conducted with M-901 E gears in the tank car, with the couplers of the tank car and the hammer car being at the same level. It is seen that there is an initial upward vertical force followed by a downward vertical force during an impact. Negative values indicate that the vertical force is downward. This trend of an initial upward vertical force, followed by a downward vertical force is seen in all cases. In this particular case, an upward vertical force of about 43 kips, and a downward vertical force of about 56 kips were developed, in addition to a 632 kip longitudinal force.

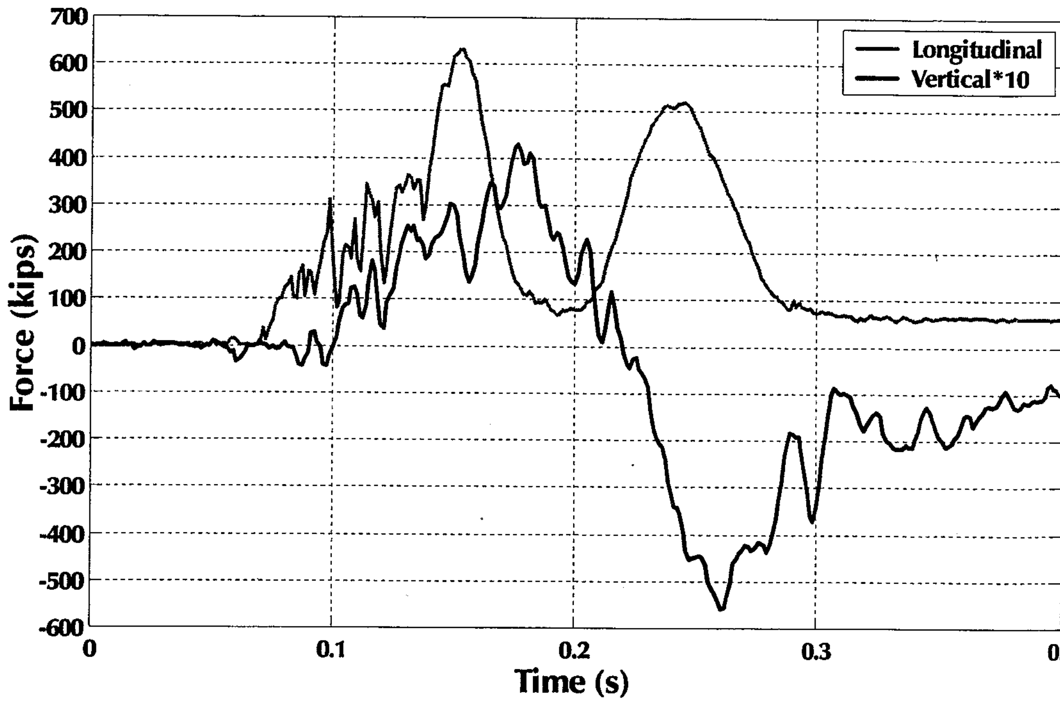


Figure 14. Sample vertical & longitudinal force trace - 7.5 mph - Couplers level

Figure 15 shows vertical force traces from 7.5 mph impacts for two different cases of coupler height mismatch. In this instance, the case where the tank car (TC) is 2" higher produces peak vertical forces of 26 kips and 15 kips in the upward & downward directions, respectively. The case where the TC is lower produces peak vertical forces of 46 kips and 38 kips in the upward & downward directions, respectively. Obviously, the case where the tank car is 2" lower gives higher vertical force peaks than when the tank car is 2" higher.

The above trend was seen for most cases with few exceptions. The condition where the tank car is lower almost consistently produces higher peak forces, especially in the downward direction. As previously mentioned (see section 3.1.2), downward vertical forces tend to be more critical than upward vertical forces. Essentially, stresses on the

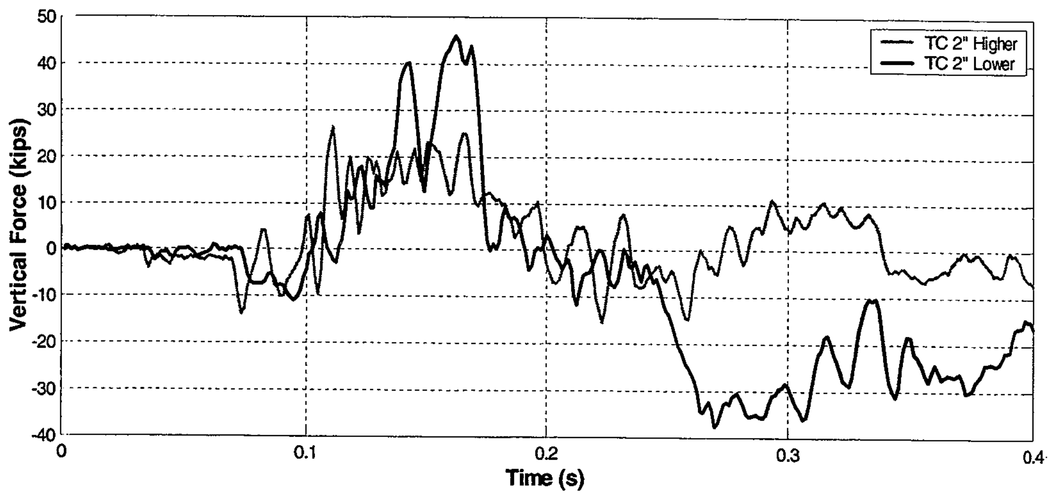


Figure 15. Vertical force comparison - 7.5 mph

tank head and head pad extension are higher under a downward vertical force, and tend to be additive to stresses from squeeze forces. Hence, under impact conditions where high squeeze (buff) forces exist, vertical forces in the downward direction are very critical. Figure 16 shows averaged peak vertical forces in the downward direction plotted against impact speed. From this chart, it is obvious that the struck car being lower is a more critical case, compared to when the cars are at the same level or if the struck car is higher.

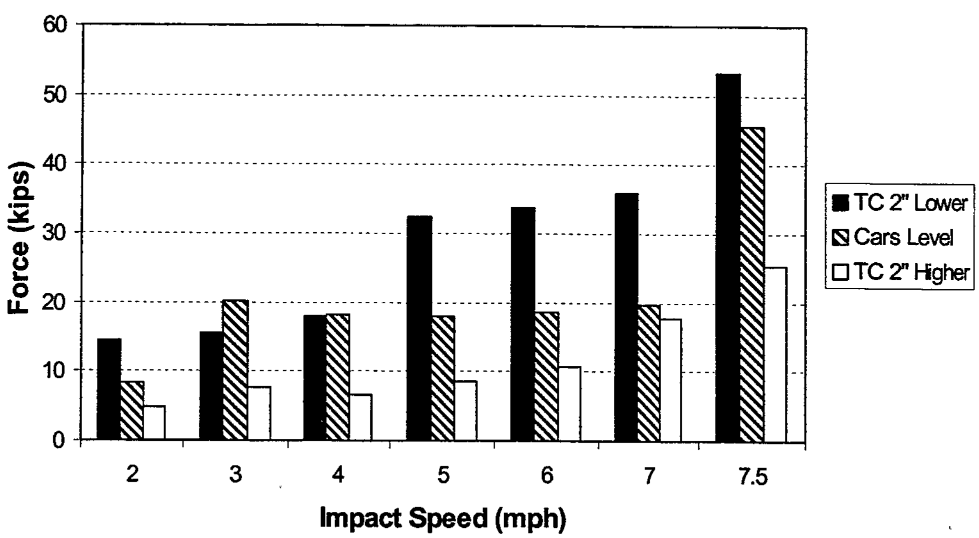


Figure 16. Peak vertical forces (averaged) - Downward

When considering forces in the upward direction, the results are not as straightforward (see figure 17). At speeds greater than 4 mph, slightly higher force peaks were produced when the cars were at the same level, than when the tank car was lower. In certain other cases (especially at lower speeds), higher peaks were produced when the tank car was higher. However, the force levels associated with these occurrences are small since the speeds are much lower. Given that forces in the upward direction are less critical, and

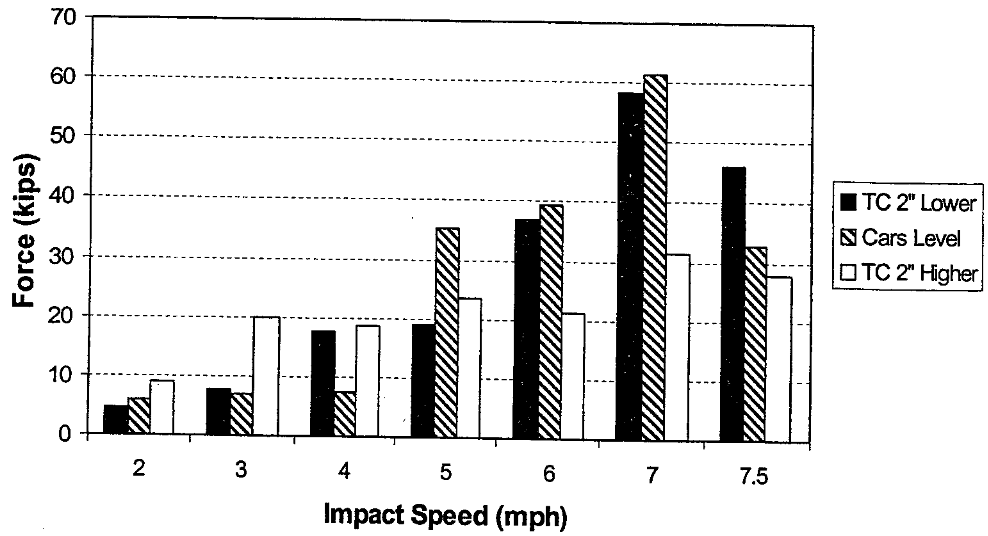


Figure 17. Peak vertical forces (averaged) - Upward

that at higher speeds, the condition where the tank car is lower produces forces that are comparable to or greater than those produced when the cars are at the same level, it is easy to conclude that the struck car being lower is the most critical case. **Therefore, it is good practice to ensure that the coupler heights of tank cars are not lower than the coupler heights of cars leading and trailing them.**

While vertical forces are higher when the struck car is lower, it was observed that high vertical forces are generated even in the absence of any coupler height mismatch between impacting cars. It is seen from figure 16 that, on average, vertical force peaks of about 45 kips were observed at an impact speed of 7.5 mph even when the impacting cars were at the same level.

These observations are indicative of the importance of coupler height mismatch levels on the vertical force levels seen by tank cars. In the next few passages, the effects of these forces on actual stresses measured will be discussed.

When the effects of coupler height mismatch on stress levels on the tank car structure were studied, it was seen that the tank car being lower produced higher stresses at critical car locations, especially at higher speeds. Once again, we will focus our attention on locations A, B & C. Figure 18 shows the measured stresses at the tank head (point A) at various impact speeds under the three different height mismatch conditions. The stress under consideration is the major principal stress (based on measurements from a strain gage rosette) at that point. It is seen that the tank car being lower causes the highest stress levels at all impact speeds. Percentage difference in tank head stresses under the three height mismatch conditions is shown in figure 19. It is seen that the stresses in the tank head are up to 15% higher when the tank car coupler is lower as compared to when the couplers are level during an impact. It is also seen that the stresses are less when the tank car is higher (at all impact speeds).

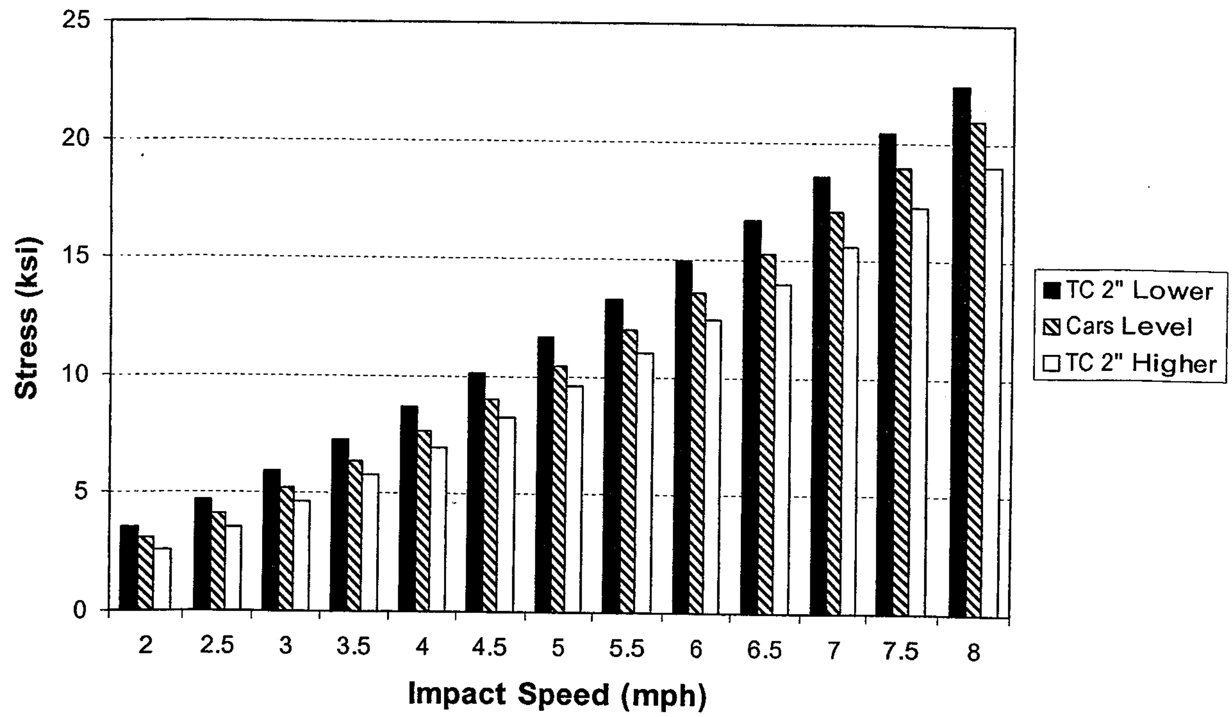


Figure 18. Principal stress at the tank head (A) - Averaged

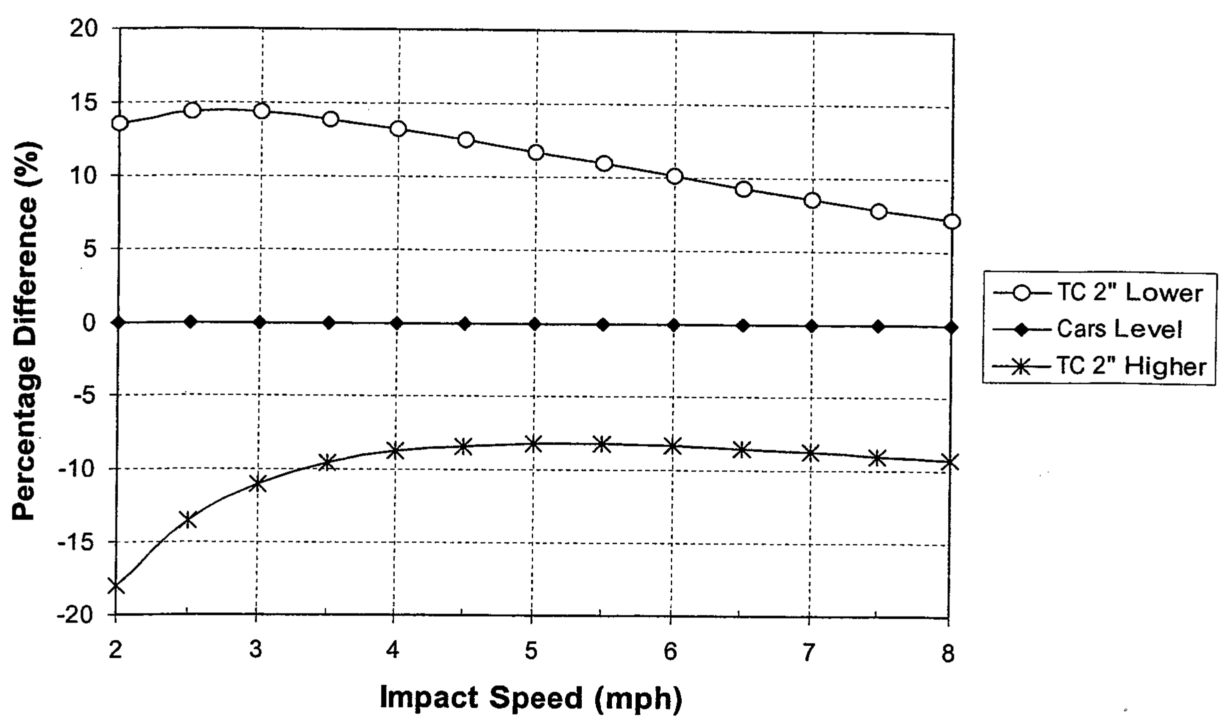


Figure 19. Variation in tank head (A) stresses

The stresses in the head pad extension (point B) show a similar trend (figure 20). Once again, the stress under consideration is the major principal stress at that point. It is seen that the condition where the struck car (tank car) is lower always produces higher stresses. Also, the case where the tank car is higher produces the lowest stresses. Percentage difference in head pad extension stresses under the three height mismatch conditions is shown in figure 21. Though percentage differences as high as 18 % are noted, at higher speeds the difference is about 5 %. While a 5 % difference might seem insignificant, it should be noted that these occur at very high stress levels, where the yield point of the material is being approached. At these high stress levels, the structural and fatigue implications of a 5 % increase in stress may be significant.

Figure 22 shows the stresses in the head brace under the three mismatch levels at various impact speeds. Here it is seen that the case where the tank car is lower produces the highest stress at all significant speeds. At lower speeds, the case where the cars are at the same level produces higher stresses. But these occur at low stress levels (under 4 ksi) and therefore are not significant. Figure 23 shows the percentage difference in head brace (point C) stress under different levels of height mismatch. When the tank car is lower, the stresses are up to 50% more, especially at higher speeds.

The above results indicate that the struck car being lower generally produces both higher vertical force levels and higher stress levels at critical locations compared to when the struck car is higher. This subsequently increases the risk of crack initiation and accelerated crack propagation, leading to premature stub sill separation or loss of lading, thereby seriously affecting operational safety. **In light of this observation, it is recommended that the coupler height of a tank car be maintained no lower than the coupler heights of adjacent cars, whenever and wherever feasible.**

3.2.2 Effect of Draft Gear Type

One of the objectives of this project was to evaluate the relative performance of a M-901 E draft gear as compared to the performance of a M-901 G draft gear under impact conditions. As described in chapter 2, impact testing on the tank car was conducted with two different sets of draft gears; the M-901 E gear set that came with the car and a set of new M-901 G gears. The M-901 E gears that came with the car were a set of 'premium' gears in good condition.

The AAR specified certification test routines for draft gears classified under M-901E or M-901G gears are different. M-901 E gears are certified using drop hammer tests, whereas M-901 G gears are certified using impact tests. M-901 E gears are required to have a minimum official capacity of 36,000 ft-lb, when tested under the standard 27,000 lb drop hammer. Also, they are required to have a capacity of at least 6,000 ft-lb at 1-5/16 in. travel. M-901 G gears are required to have a minimum rating impact velocity of 5 mph. Rating impact velocity is the velocity that produces a reaction of 500,000 lb. The impacts are to be conducted using 70-ton cars. In subsequent discussions, the M-901 E draft gears will be referred to as 'E' gears and the M-901 G draft gears will be referred to as 'G' gears.

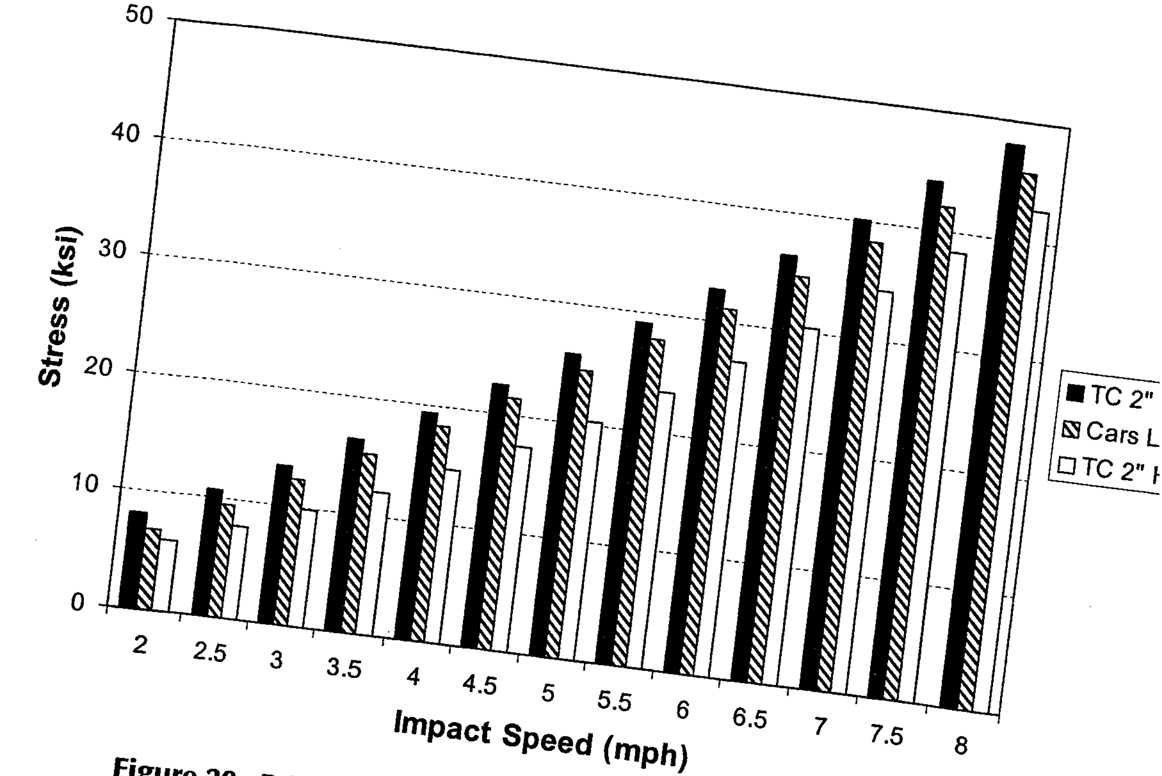


Figure 20. Principal stresses at the head pad extension (B) - Averaged

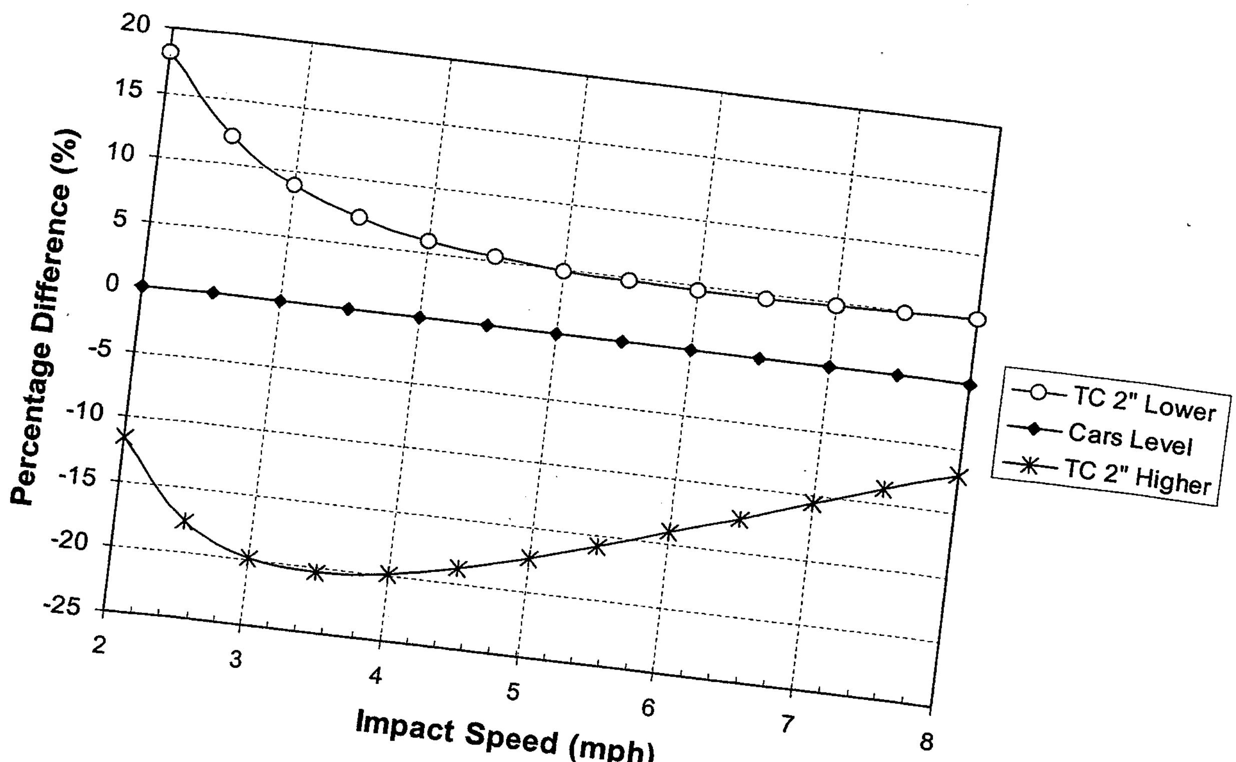


Figure 21. Variation in head pad extension (B) stresses

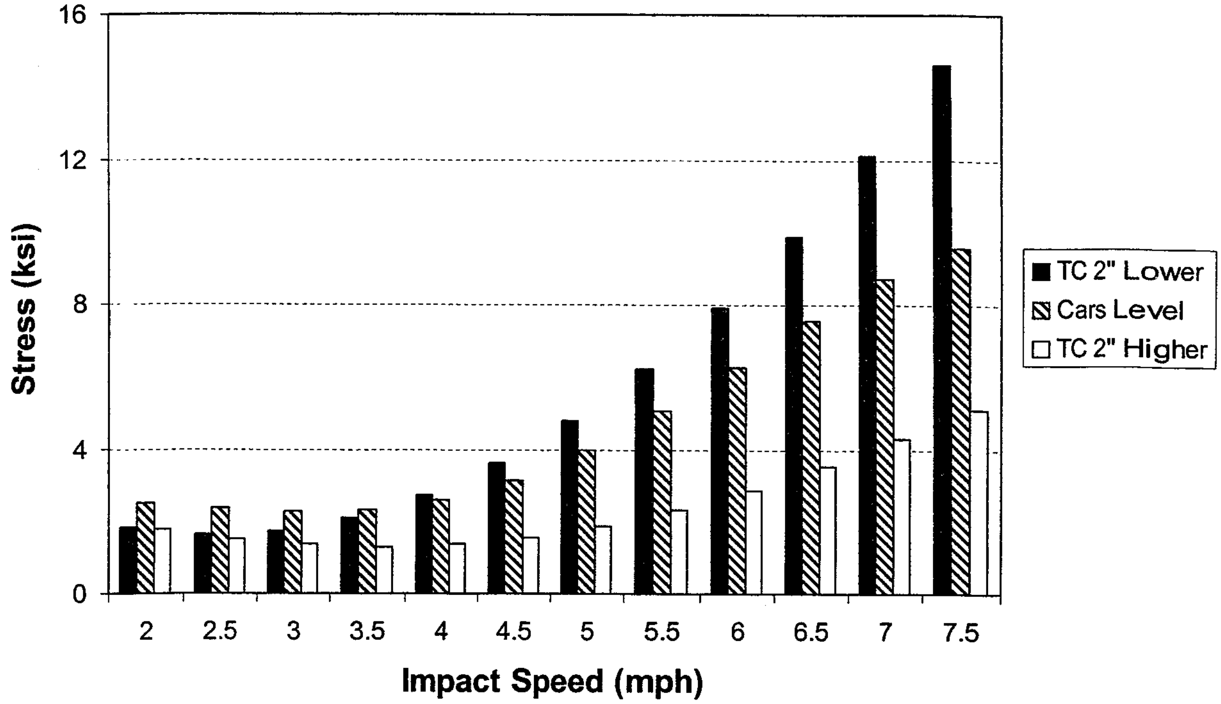


Figure 22. Stresses on the head brace (C) - Averaged

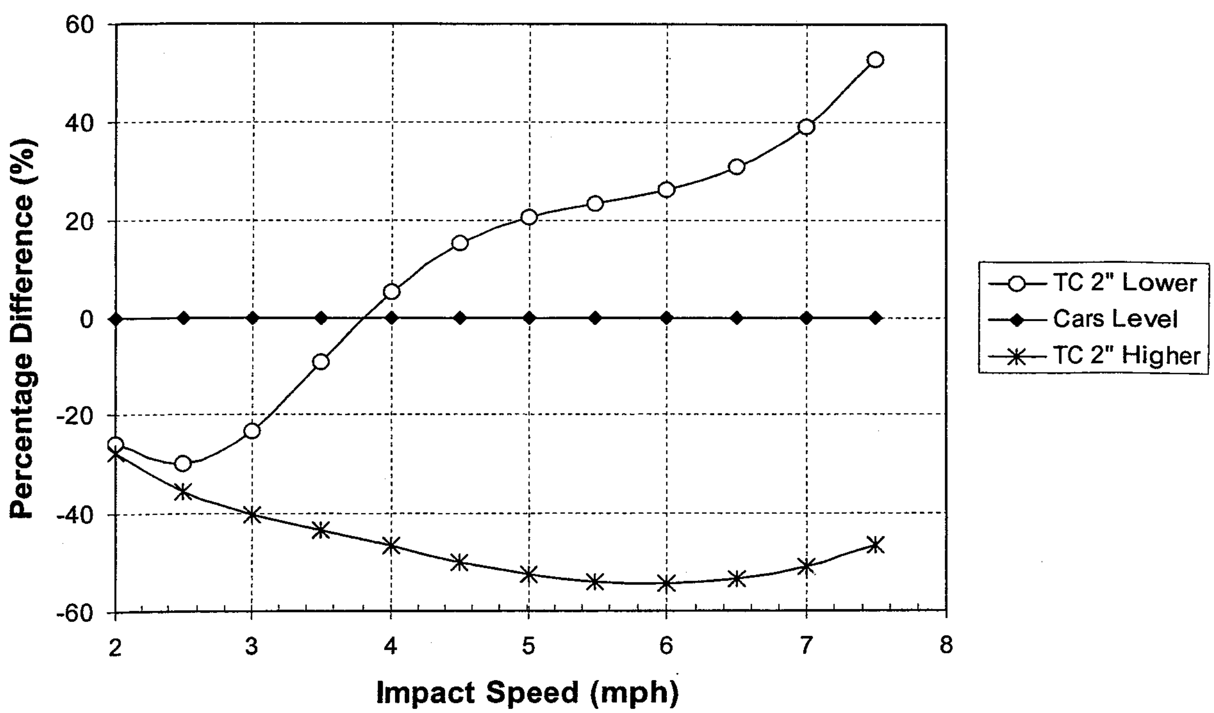


Figure 23. Variation in head brace (C) stress

The first observation made during the impact tests was that the 'G' gears closed out (solid) at slightly higher impact speeds than did the premium 'E' gears. It was seen that the 'E' gears closed out between 7.5 and 8 mph, whereas the 'G' gears closed out at between 8 and 8.5 mph. In the following passages, we will discuss the effects of different draft gears on coupler forces and stresses at critical locations.

Figure 24 shows the variation in longitudinal coupler forces with E gears as compared to G gears. It is seen that for most impact speeds, using G gears produces slightly higher impact forces. This is especially true at higher impact speeds. One exception to the case is that at around 2 mph, the G gear produced slightly lower (5 %) forces. Also, at speeds between 5 & 6 mph, the gears produced nearly identical coupler forces. Otherwise, the measured force data does not provide a convincing argument for choosing a G gear over a 'premium' E gear.

The variation of major principal stress at the tank head (A) with E and G gears is plotted in figure 25. It is seen from the plot that using G gears caused slightly higher (less than 6%) stresses on the tank head, except at lower speeds (less than 3 mph). The plot includes data from all three coupler height mismatch levels. Since, the overall trend seen in the plot is similar to the one seen if the data is separated into three mismatch levels, combining data from three mismatch levels into one graph is a valid representation of the test results. This also means that coupler height mismatch and draft gear type can be treated as independent variables in impact scenarios. This observation is valid at locations B and C also. Therefore, presenting the results purely as a function of gear type and impact speed is a valid representation of draft gear behavior.

A similar trend is seen when principal stresses on the head pad extension (B) are compared (figure 26). While there is no significant difference in stress levels at lower speeds (under 3 mph), stresses are slightly higher (less than 5 %) when G gears are used. Stresses measured at the head brace (C) show a slightly different trend (figure 27). While the stresses with the E gears are slightly lower at most speeds, at higher speeds (greater than 6.5 mph) the G gears offer better protection.

The results of this test effort do not offer a convincing argument for preferring M-901 G gears over 'premium' M-901 E gears for use in 263,000 lb hazardous material service, under impact conditions. However, revenue service environment is different and is better characterized by lower force levels and lower run-in/run-out speeds. Hence, the above conclusion may not be applicable. It should be emphasized here that the 'G' gear that was tested was brand new, as compared to the 'premium E' gear which was already broken in. Generally, draft gears tend to perform better after they have been broken in. The results could however be very different when the comparisons are made with 'non-premium'/used E gears. The results could also be different if 286,000 lb tank cars are considered. We will attempt to answer these questions through a follow-up project that is now underway.

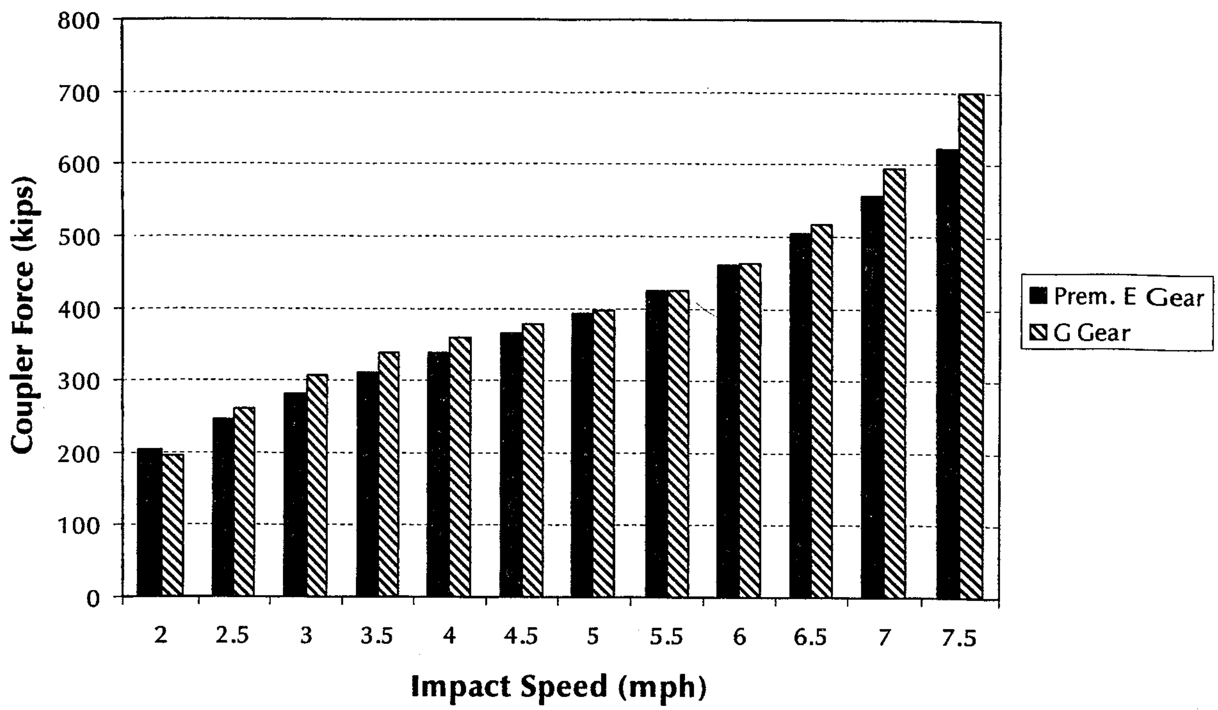


Figure 24. Variation of coupler forces with draft gear type

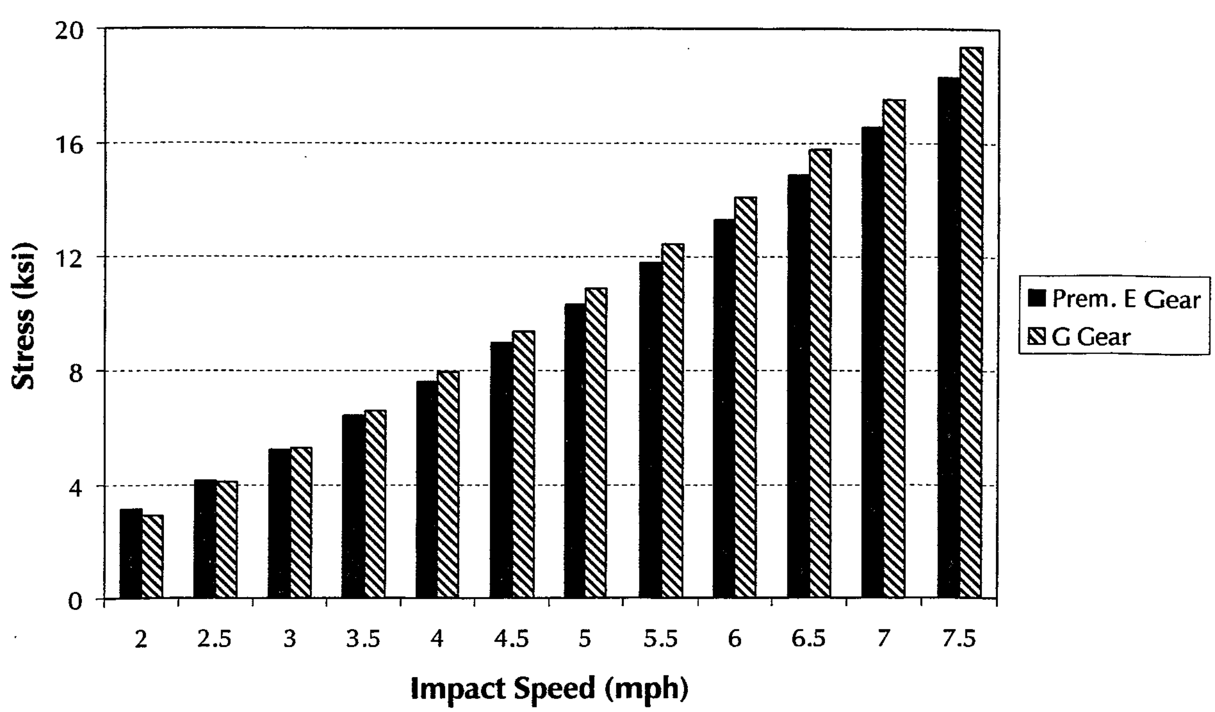


Figure 25. Variation of tank head (A) stresses with draft gear type

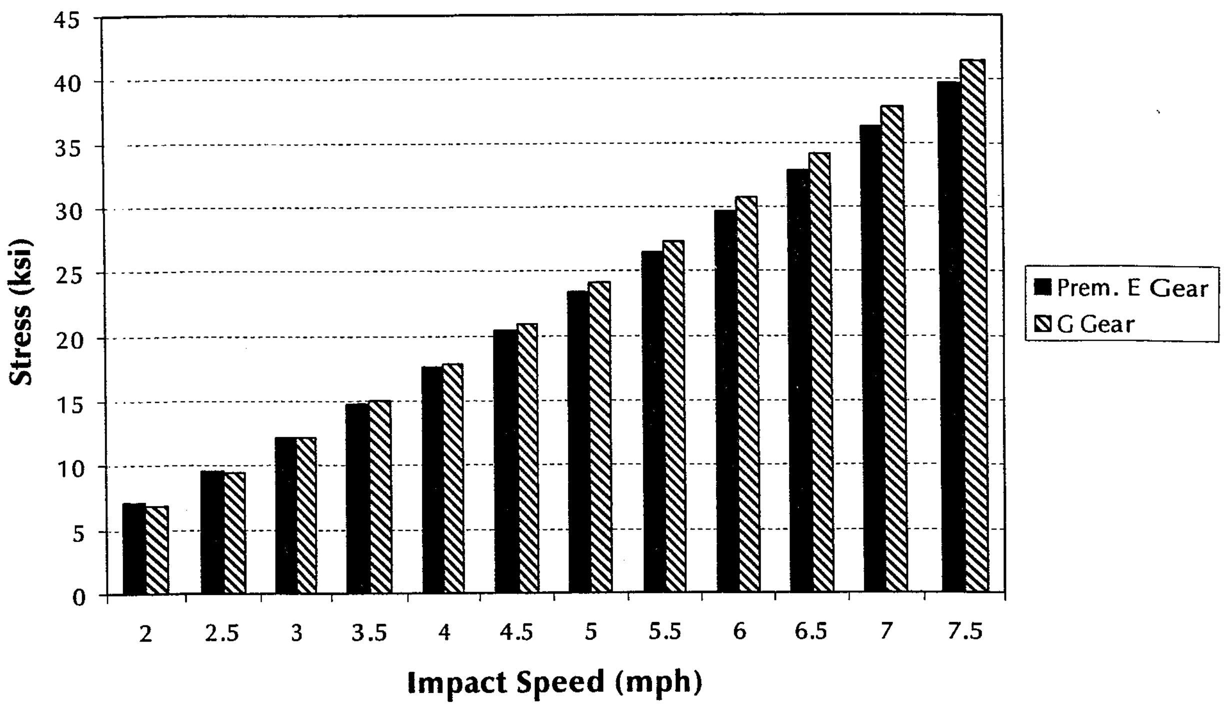


Figure 26. Variation of head pad extension (B) stresses with draft gear type

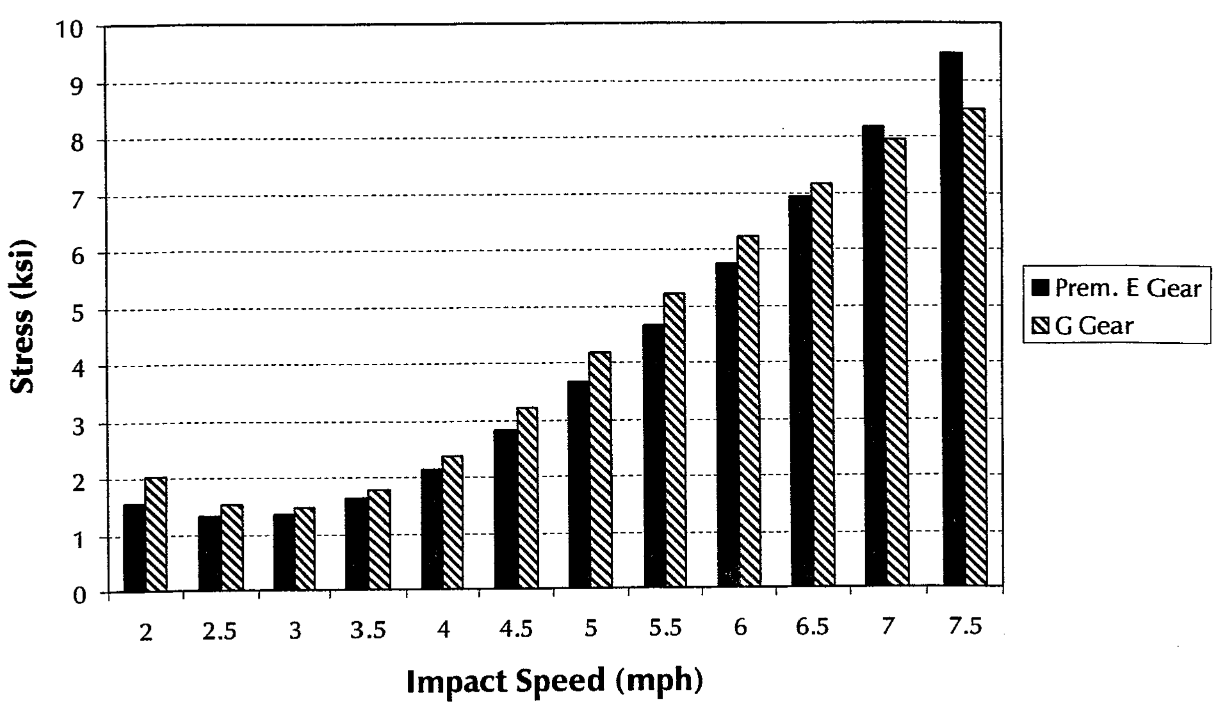


Figure 27. Variation of head brace (C) stresses with draft gear type

3.2.3 Effect of Vertical Forces

In earlier sections we have established the level of vertical forces that are seen during impact conditions. In this section, we have tried to evaluate the extent to which vertical forces contribute to the stress levels seen by tank cars. To establish how much a given vertical or longitudinal force affects the stress level at a point, it is necessary to establish calibration functions for these forces and stresses. Based on our static testing results (see section 3.1.1), we established static calibration factors at critical locations on the tank car for both vertical and longitudinal forces.

We then developed a mathematical model to evaluate principal stresses at points A & B based on these calibration factors and the measured longitudinal force (instrumented coupler) and vertical force (sill gages) histories. This is a simple model that evaluates stresses purely as a function of the force input. It does not take into full consideration draft gear characteristics, time lag between application and response, or other dynamic factors. However, the model still predicts stress histories that are comparable to the measured values in many cases. Figure 28 shows time histories from measured and evaluated principal stresses at the tank head (location A) for a speed of 4 mph. Figure 29 shows a similar history for a speed of 8 mph. As seen, the model predicts stress histories that are comparable to the measured histories in these cases. Good correlation between measured and evaluated stresses was seen in many cases. This was especially true at higher speeds. However, in some cases, especially at lower speeds the evaluated and measured values did not match up very well. This is because, the low speed performance of draft gears is highly dependent on the stick-slip behavior of the draft gear wedges which is not repeated reliably in impact tests.

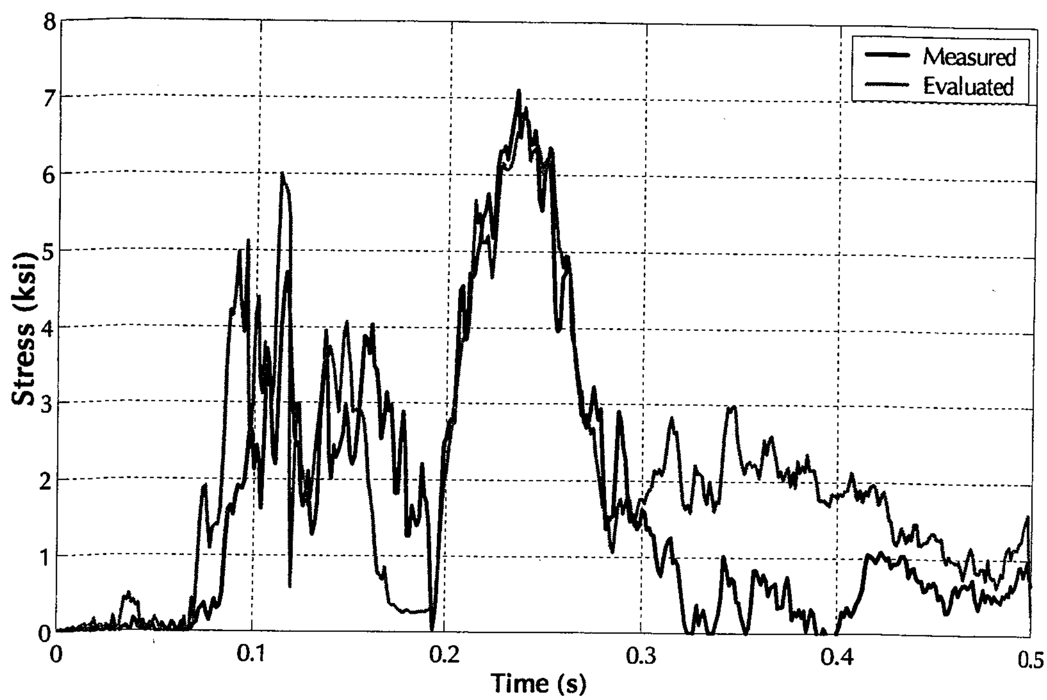


Figure 28. Comparison of stress time histories - Cars level - 4 mph impact

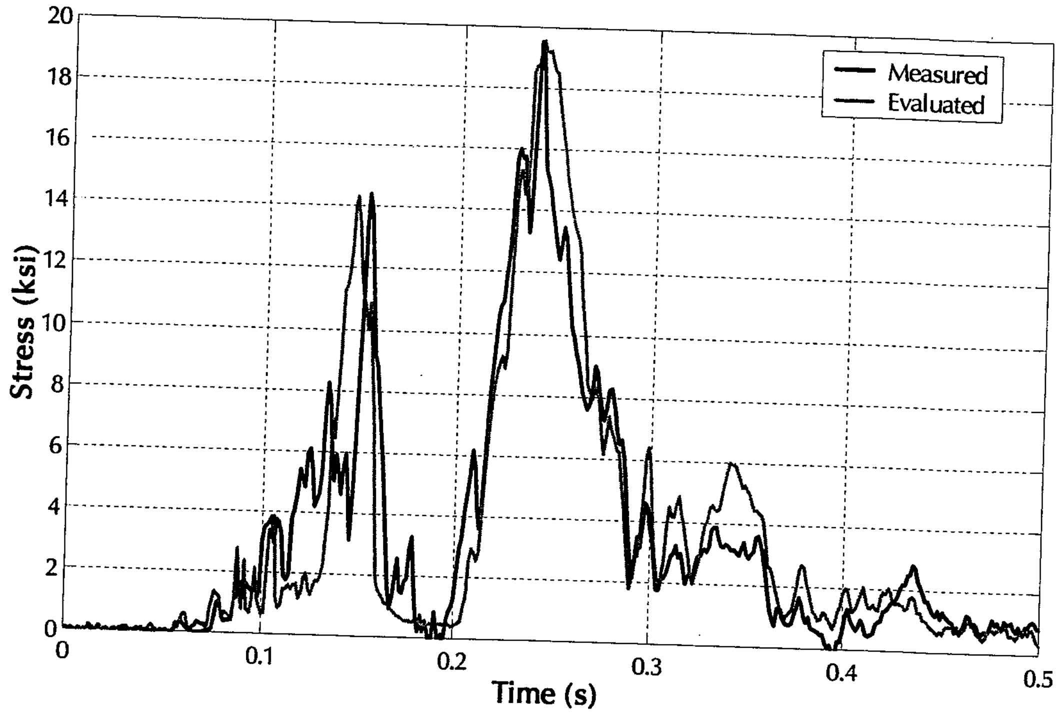


Figure 29. Comparison of stress time histories - Cars level - 8 mph impact

Since the above model predicts stress response to input vertical and/or longitudinal forces, it is possible to separate the vertical force effects from the longitudinal force effects. This was the main intended function of the model. In order to perform a valid comparison, we only chose the cases where the evaluated stress histories matched the measured histories reasonably well. Cases where the general trend of the predicted history was correct and the stress peaks were within 25 % of the measured values were picked. For these cases, the stresses expected at points A & B from the longitudinal force and the vertical force were evaluated.

Analysis of stresses at location A revealed that on average, vertical forces accounted for about 50 % of the stresses caused. For eighty percent of the cases analyzed, the percentage of stress attributable to vertical force was between 30 % and 70 %. At location B, vertical forces accounted for about 44% of the stresses caused, on average. As at location A, the percentage of stress attributable to vertical force was between 30 % and 70 % for over eighty percent of the cases analyzed. Additional analysis of this information did not reveal any speed based trends. As previously mentioned, the numbers presented here pertain mostly to impact data at speeds between 4 & 8 mph.

It can be reasoned from this exercise that it is essential to consider vertical coupler forces during any analysis or determination of the structural integrity of a tank car. Ignoring the effects of vertical force will lead to under-estimation of stress levels and fatigue levels in tank cars.

4. ANALYTICAL WORK

4.1 Shelf Coupler Stiffness

One of the concerns raised during this project was whether the structure and stiffness of top and bottom shelf couplers, mandated for use on hazmat tank cars, could cause more cars to derail, compared to the case where the cars had regular non-shelf couplers. Tank cars are required to use couplers with top and bottom shelves in order to prevent the coupler of an adjacent car from puncturing the tank head in the case of an undesired uncoupling, coupler override or an accident. A tank head puncture could result in uncontrolled lading release leading to serious consequences, especially in the case of hazardous materials.

While shelf couplers are beneficial in cases where potential uncoupling and coupler override might occur, there is a concern that the added torsional rigidity of such coupler connections might be detrimental in certain other cases, especially where cars have leaned over and derailed, for example, low speed derailments in classification yards. In such a scenario, there is a potential for a car that has derailed to 'take-down' an adjacent car with it, because the coupler connection was rigid. By the same argument, it is also possible that the rigidity of the coupler could prevent a car from tipping over, if it is held back by underailed cars on the other side. This issue does not lend itself to be resolved easily or through simple analyses, given the complexity of the coupling mechanism, design of the shelves and the varying circumstances leading to potential derailment. Therefore, a detailed study of this issue is beyond the scope of the work being presented here.

4.2 Effect of Stiffness Transitions at Grade Crossings

One of the questions that needed to be addressed as part of our project was whether stiffness transitions on railroad track, for example, at grade crossings, would contribute significantly to either vertical force levels or coupler mismatch levels between adjacent cars. To evaluate this, we developed a dynamic finite element model (using LS-DYNA3D) that modeled a single wheel of a freight car, including the corresponding mass and truck suspension, moving over a railroad track with a stiffness transition.

A schematic of the grade crossing model is shown in figure 30. As shown, the car mass corresponding to one wheel (one-eighth) is connected to a wheel through a standard 100-ton car suspension. The values for stiffness and damping of the truck suspension were derived from that of a standard 100-ton truck. The stiffnesses of the subgrades were set as needed to simulate various scenarios. A finite element plot of the model is shown in figure 31. Figure 32 is a close-up of the stress levels in the rail at the point of transition.

Three different stiffness variation scenarios were simulated (see table 4). A subgrade stiffness of 5,000 lb/in/in represents track in good condition. A subgrade stiffness of 2,500 lb/in/in represents railroad track in average-to-poor condition. Special scenarios such as track with concrete slab underlays are represented by the 25,000 lb/in/in stiffness. The simulations were run at two speeds 30 mph and 60 mph. Both forward and reverse directions were considered (stiff-to-flexible & flexible-to-stiff).

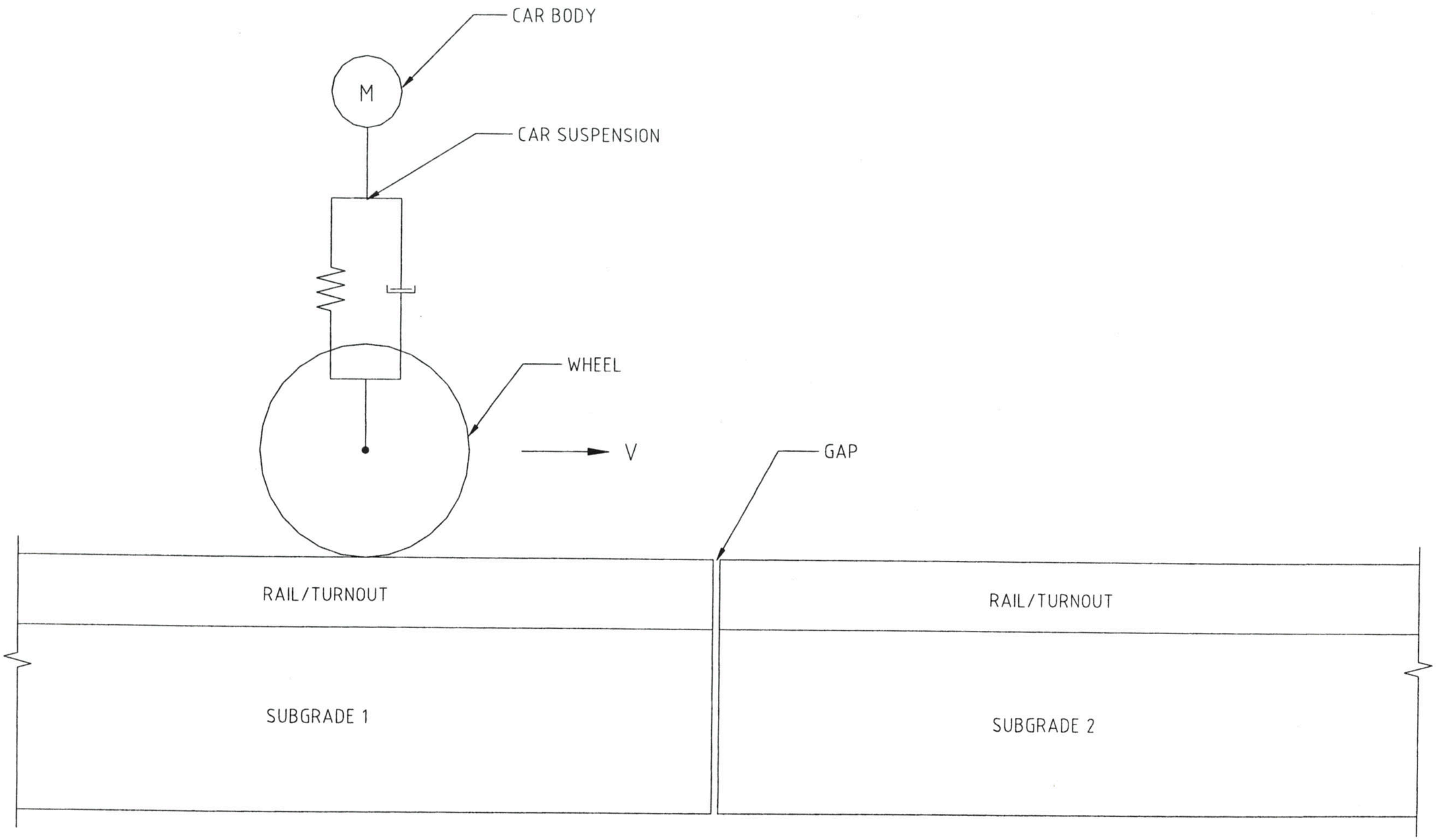


Figure 30. Schematic of grade crossing model

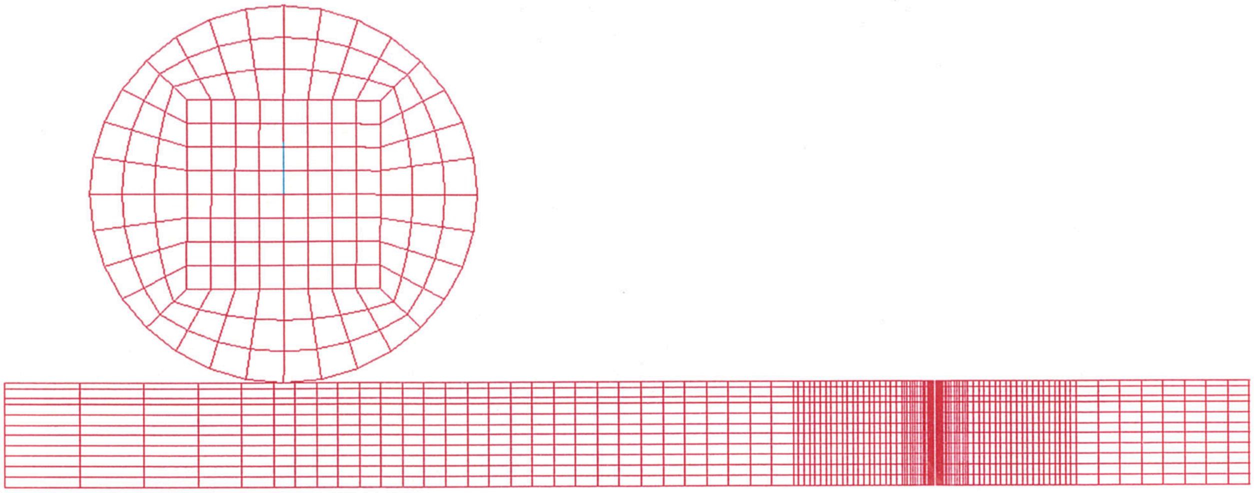


Figure 31. Finite element model – Grade crossing simulation

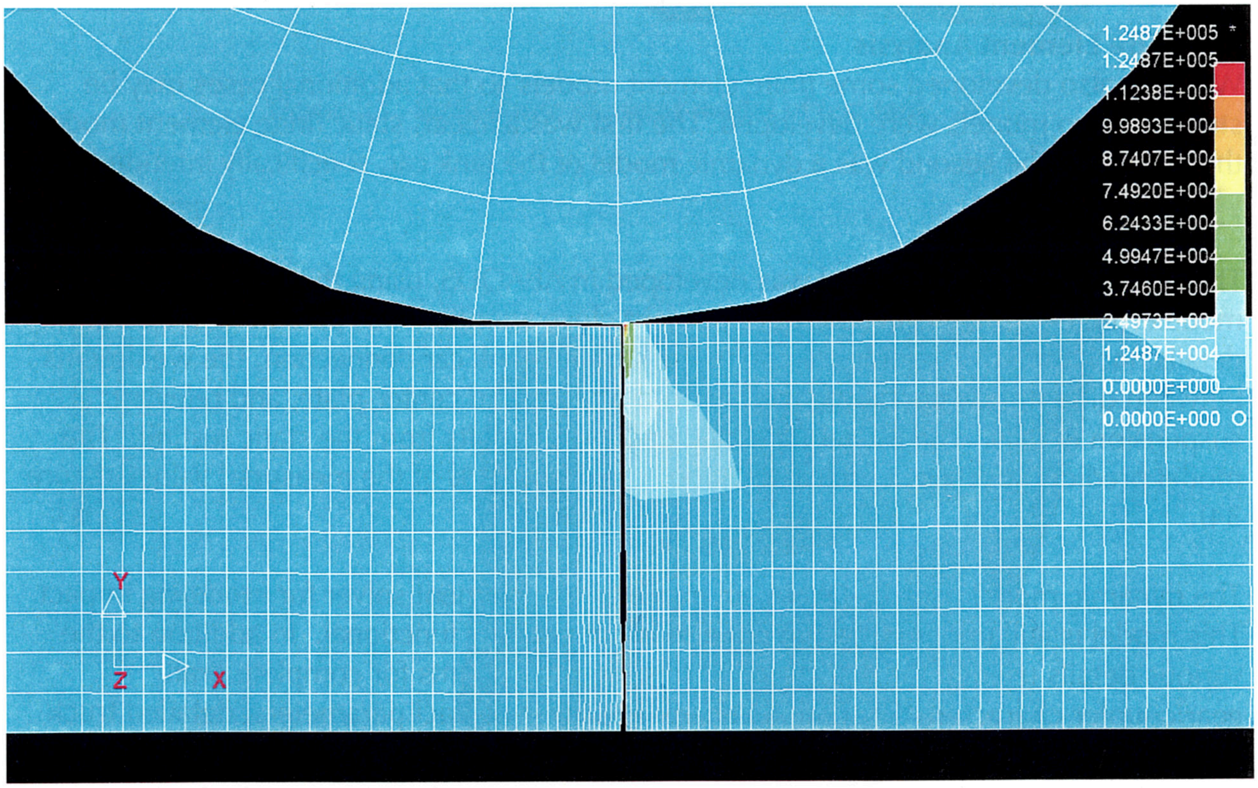


Figure 32. Maximum rail stress – Grade crossing simulation

Table 4. Grade crossing simulation scenarios

Subgrade Stiffness (lb/in/in)		Stiffness Ratio	Speed (mph)	Direction
Subgrade1	Subgrade2			
5,000	5,000	1	30, 60	-
5,000	25,000	5	30, 60	F, R
2,500	25,000	10	30, 60	F, R

For the cases simulated, the maximum vertical wheel displacement observed was less than one inch. Car body displacements were much smaller. The maximum car body displacement that was observed was only one-eighth of an inch. This indicates that most of the wheel displacement is absorbed by the truck suspension before it reaches the car body. In the scenarios being modeled any coupler height mismatch would have to come from differential car body displacements. Since the car body displacements are small, coupler height mismatches will be small also. It must be noted here that these analyses assume that the track is on the same level on either side of the transition. Differences in track height are likely to induce higher car body displacements.

4.3 Finite Element Analysis

We have also developed some finite element models so that additional cases may be studied. Two models were developed. The first was a linear, static finite element model of the tank car. The second was a dynamic model of the tank car, specifically intended for yard impact studies.

The static finite element model was developed in ABAQUS, using the I-DEAS modeler(figure 33). During the development close attention was paid to all the relevant details, including the complex shape of the stub sill-tank car connection, the weldments between different components, the bolted draft lugs and center-plate, and relevant suspension details. The model was validated using results from the static testing. The model predicts stresses at critical locations to within 10 % of the measured values (figure 34).

The dynamic model uses LS-DYNA, an explicit finite element solver for modeling impact. This model is still under development and is not yet fully functional. In addition to the various details used in the static model, this model accounts for complete draft gear/coupler characteristics, coupler shank-to-striker contact parameters, detailed truck suspension characteristics, and the masses and stiffnesses of the hammer car & backup cars. So far the model has been predicting peak coupler forces with good accuracy. Once the model is further tuned using some impact test data, it can be used to study many different impact scenarios.

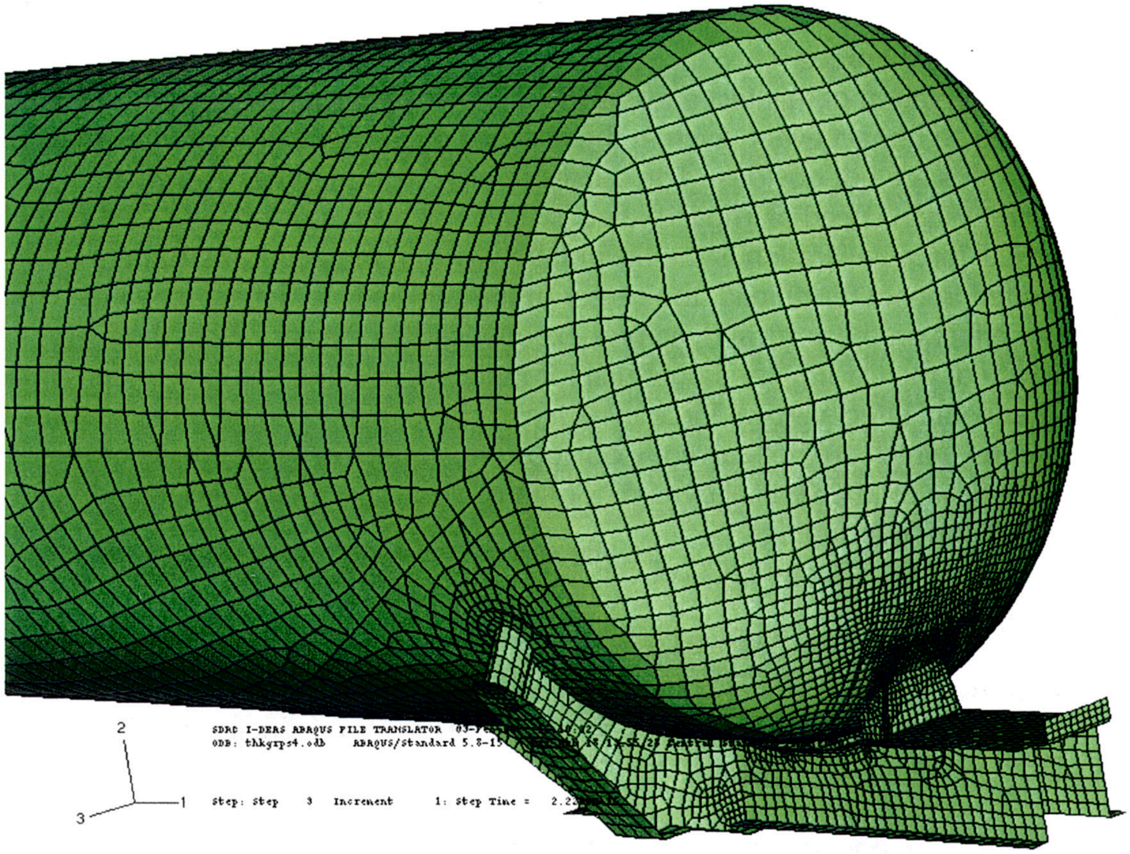


Figure 33. Static finite element model of test tank car

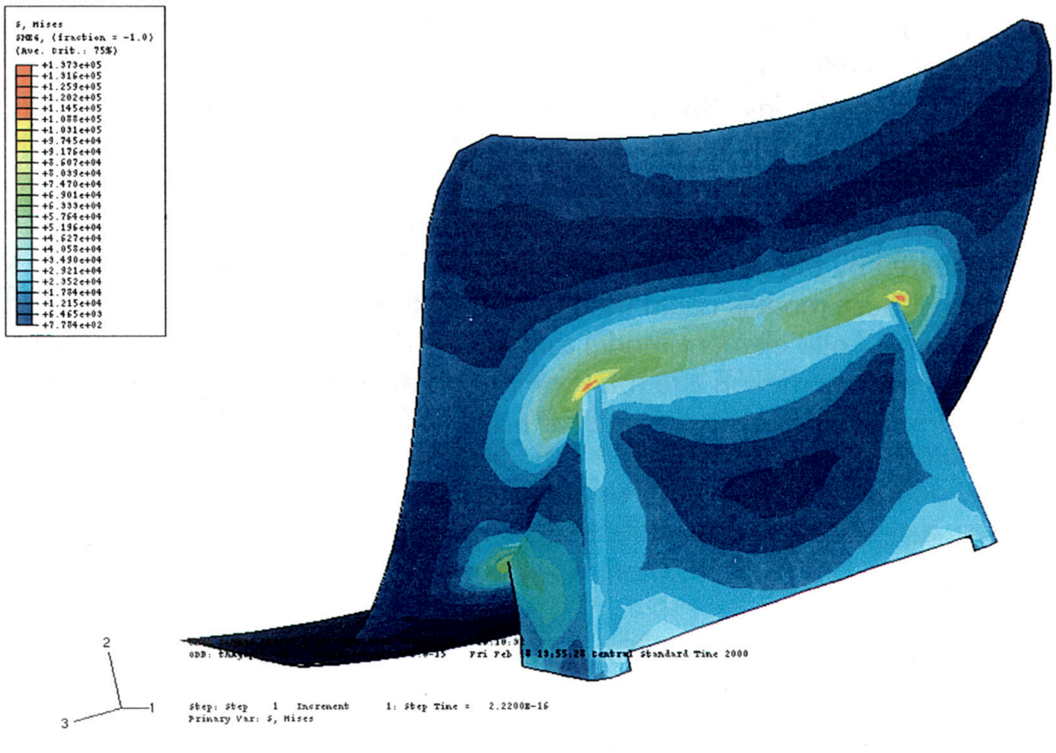


Figure 34. Stress pattern at the head brace and reinforcing pad under a squeeze load

4.4 Source of Vertical Forces

An important observation of this test effort is the fact that significant vertical forces are generated during an impact event, even in the absence of any coupler height mismatch. The reasons for this are not obvious. The causes of high vertical forces and what parameters (other than speed of impact) affect it have not been studied yet. One potential cause for vertical force generation is the flexural behavior of the tank car structure itself. Under an applied impact (buff) load, the center of the car tends to deflect upwards. This could have an effect on the vertical dynamics experienced by the car. Also, the high center-of-gravity of the car implies a potential for inducing pitching motions under impact conditions that could affect vertical force dynamics. There might also be other reasons for this phenomenon, which are not currently obvious. It is not clear what effects other variables might have on the vertical dynamics of tank cars. These variables include :

- Truck spring groupings
- Impacting car structure, CG of car
- Coupler type
- Striker-to-coupler shank clearances
- Fluid dynamics (sloshing of lading, etc.)
- Vertical vibration characteristics of impacting cars (frequencies, modes)
- Draft gear characteristics

This high vertical force phenomenon was observed on a stub sill-tank car. So far, there is no testing or analysis data that indicates whether the observed phenomenon is restricted to stub sill-tank cars or is present to varying degrees in other freight car designs (e.g., through sill or side sill designs).

Given the criticality of vertical coupler forces on stub sill integrity, it is imperative to study this process in detail in order to gain a better understanding of the vertical dynamics of a tank car under impact conditions. Such a study would help us identify the reasons for such high forces and also identify methods of minimizing the force levels. This would distinctly improve the safety of tank car operations. The study could be started with the development of simple dynamic models that would at least identify the source of this phenomenon, followed by the development of verifiable detailed models to further extend our understanding. These models could then form the basis for developing potential remedies. The dynamic finite element model that we are developing will provide more understanding of this phenomenon and may point to possible design improvements that could be made to improve the safety of tank car operations.

5. CONCLUSIONS

1. Significant vertical forces are generated during an impact event, even in the absence of any coupler height mismatch between impacting cars. On average, vertical force peaks of about 45,000 lb were observed at an impact speed of 7.5 mph even when the impacting cars were at the same level.

2. Under impact conditions, vertical forces induce 50 % (average) of the maximum stress seen at the tank head (location A), and about 44 % (average) of the maximum stress seen at the head pad extension (location B). Ignoring the effects of vertical force will lead to under-estimation of stress levels and fatigue levels in tank cars. Therefore, safety evaluations of stub sill-tank cars must include vertical force effects simultaneously with longitudinal force effects.

3. During an impact event, an initial upward vertical force is generated which is followed by a downward vertical force. The downward vertical force is generally more critical because it adds to the tensile stress levels created by the longitudinal buff force.

4. The coupler height mismatch level between impacting cars has a significant effect on the vertical force levels during an impact. The downward vertical force tends to be higher when the struck car is lower than the hammer car. Also, the stresses in critical areas of the stub sill-tank car interface are generally higher when the struck car is lower. Differences (in stress level) over 50% (compared to when the cars are level) were noted at the head brace, when the struck car was 2" lower than the hammer car. These higher vertical force levels and higher stress levels at critical locations increase the risk of crack initiation and accelerated crack propagation, which could lead to premature stub sill separation or loss of lading, thereby seriously affecting operational safety. In light of this, it seems prudent to keep coupler height mismatch to a minimum, whenever and wherever feasible. We also recommend that the coupler height of a tank car be maintained no lower than the coupler heights of adjacent cars.

5. The static tests established the criticality of vertical forces on the structural integrity of the stub sill-head brace area. On average, a 50,000 lb vertical force can generate as much stress as a 680,000 lb longitudinal force, under static conditions. This implies that moderately high longitudinal forces, in conjunction with moderately high vertical forces can cause significantly high stresses, which may be near or above the yield point of the tank material.

6. Under impact conditions, there were no significant differences in performance between the 'premium E' gears and the 'G' gears that were tested. **However, the 'G' gears closed out at higher impact speeds (8.0 - 8.5 mph) than did the premium 'E' gears (7.5 - 8.0 mph).** While, the 'G' gears performed slightly better in some cases and the 'premium E' gears performed better in other cases, the test

results did not offer a convincing argument for choosing 'G' gears over the 'premium E' gears. However, revenue service environment is different and is better characterized by lower force levels and lower run-in/run-out speeds. Hence, the above conclusion may not be applicable. It should be emphasized here that the 'G' gear that was tested was brand new, as compared to the 'premium E' gear which was already broken in. Generally, draft gears tend to perform better after they have been broken in. Also, the results could be very different when the comparisons are made with 'non-premium'/used E gears. The results could also be different if 286,000 lb tank cars are considered.

7. Simulations of car movement over track stiffness transitions such as grade crossings indicate only small levels of coupler height mismatch under these scenarios.

6. RECOMMENDATIONS FOR FUTURE WORK

1. This project evaluated the safety implications and effects of coupler height mismatch on the structural integrity on tank cars under impact conditions. It would be useful to extend this project so that the effect of coupler height mismatch in revenue service can be studied. This can be done by instrumenting a series of cars (to measure force levels & stress levels) with different levels of height mismatch, using them in revenue service and studying the data collected. This data would also help in quantifying the longitudinal and vertical load cycles seen by a tank car in revenue service.

2. Traditionally, fatigue analysis of freight cars has been done under the assumption that longitudinal and vertical loads occur independently. This study has demonstrated that under impact conditions, both longitudinal and vertical forces occur simultaneously in tank cars. Therefore, it is essential to conduct detailed fatigue evaluations, especially at the stub sill-tank interface, that account for the simultaneous application of these forces. This might necessitate the development of fatigue evaluation techniques and force histograms that permit the simultaneous application of different force types. On the same note, it would also be useful to conduct a detailed survey of classification yard operations that would indicate the usual range of impact speeds seen by tank cars. This will assist in better quantifying the expected loading history of tank cars, and thus ensure more reliable safety evaluations.

3. The impact tests conducted during this project were restricted to tests with new M-901 G and premium M-901 E draft gears. However, the majority of draft gears in service are softer, non-premium, M-901 E gears that have been in service for many years. These gears may not provide the same level of protection to the tank car as the ones already tested. The differences in performance may be more severe in the case of 286,000 lb service. Therefore, it is necessary to conduct a detailed test and analysis program to evaluate the protection levels offered by soft 'E' gears in 263,000 lb and 286,000 lb service.

4. An important observation of this test effort is the fact that significant vertical forces are generated during an impact event, even in the absence of any coupler height mismatch. The causes of high vertical forces and what parameters (other than speed of impact) affect it have not been studied yet. However, given the criticality of vertical coupler forces on stub sill integrity, it is imperative to study this process in detail in order to gain a better understanding of the vertical dynamics of a tank car under impact conditions. Such a study would help to identify the reasons for such high forces and also identify methods of minimizing the force levels, thereby improving the safety of tank car operations. The study could be started with the development of simple dynamic models that would at least identify the source of this phenomenon, followed by the development of verifiable detailed models to further extend our understanding. These models could then form the basis for

developing potential remedies. Once such models have been developed and validated, they could also be used to study impact scenarios under various combinations of car weights, draft gear characteristics, coupler height mismatch levels, etc. in a far less expensive manner than full scale testing. As mentioned in section 4.3, we are already in the process of developing some basic models. We hope to extend the development process and produce a model that has been validated using results from the impact tests. The model will then provide a better understanding of impact phenomena (including high vertical forces) and may point to possible design improvements that could be made to improve the safety of tank car operations.

5. There is a concern that the added torsional rigidity of double shelf couplers might be detrimental in cases where cars have leaned over and derailed. In such a scenario, a car that has derailed might 'take-down' an adjacent car with it because the coupler connection was rigid. The effects of shelf coupler rigidity on the derailment propensity of tank cars needs further study.

7. References

1. Sharma, V., Sneed, W.H., and Punwani, S.K., 1984, "Freight Equipment Environmental Sampling Test-Description and Results", ASME Rail Transportation Spring Conference Proceedings, pp. 59-70.
2. Cogburn, L.T., 1995, "Stub Sill Tank Car Research Project - Results of a 15,000 mile Over-the-Road Test", Report No. FRA\ORD\95-11, DOT/Federal Railroad Administration, Washington, DC.
3. "Fatigue Design of Freight Cars", Specifications for design, fabrication and construction of freight cars, AAR specification M-1001, Association of American Railroads.