CARGO TANK MANUFACTURERS ASSOCIATION
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268889 RSPA-04-17074-1 P-1433

November 1, 2003

U.S. Department of Transportation
Research and Special Programs Administration
400 Seventh Street NW
Washington, DC 20590-0001

Attention: Edward Mazzullo, Office of Hazardous Materials Standards

Subject: Petition of Rulemaking, DOT 331, 406, 407 & 408 tanks

Gentlemen:

This is a petition for rulemaking concerning the subject cargo tanks which have essentially identical requirements for structural integrity. It is petitioned that the vertical dynamic stress factors for the normal transport condition be changed from .35 to .25G and for the extreme transport condition from .7G to .45G. The justification for this request is contained in the **REPORT OF STUDY ON THE DYNAMIC ANALYSIS OF DOT 407/412 TYPE CARGO TANK MOTOR VEHICLES** dated May 24, 2003 and done under U.S. Department of Transportation Order No. TRS56-02-P-70045 of September 25, 2002. This report analyzes the results of over-the-road tests on a particular semi trailer type cargo tank which was driven over very rough roads under full load, half load and empty conditions to determine the G forces associated with each mode of response of the tank to dynamic forces experienced. This report contained some recommendations for rule change and further study and this petition covers Recommendations a) and b) under regulation change.

The exact changes recommended to the regulations are presented in two attached documents entitled **DOT 400 SERIES TANKS, PROPOSED REVISIONS TO 49CFR178.345-3 STRUCTURAL INTEGRITY** and **DOT 331 CARGO TANKS, PROPOSED REVISIONS TO 49CFR178.337-3 STRUCTURAL INTEGRITY.** The economic justification for these changes is given in a report entitled **ECONOMIC IMPACT OF PROPOSED CHANGES TO DOT CARGO TANK REGULATIONS CONCERNING DOT 400 SERIES TANKS** prepared by M.R. Ward, P.E. and dated May 24, 2003. The proposed changes result in certain tanks being lighter and therefore able to carry 1% more product. This means that 1% fewer tanks can carry the same amount of load as the heavier tanks since these tanks almost always operate fully loaded. Making a conservative assumption that there are 3000 such tanks operating at a conservative annual cost of $100,000 per tank, the savings amount to $3,000,000 per year minimum (one percent of $300 million).

We hope this information is adequate to initiate the rulemaking process on these items. Please advise if you need more information.

Respectfully,

M.R. Ward —

M.R. Ward, P.E.
CTMA Technical Manager

RTL, Inc.
RR#1 Box 91
Hollister Ranch
Gaviota, CA 93117
(805) 567-0280

Enclosures

MRW/cs

DOT 331 CARGO TANKS
PROPOSED REVISIONS TO 49CFR178.337-3 STRUCT. INTEGRITY

Description of recommended changes:

a) In 178.337-3©(1)(iii)C and 178.337-3©(1)(iv)A, change the factor .35 to .25 for vertical accelerative force for normal operating loadings.
b) In 178.345-3©(2)(iii)C and 178.345-3©(2)(iv)B, change the factor .7 to .45 for vertical accelerative force for extreme dynamic loadings.

Rationale: A recent over the road test of a cargo tank motor vehicle in which the dynamic response was subjected to a harmonic analysis to determine the distribution of accelerations over a wide frequency range indicated that the factors in the present regulations are well above what contributes to design load for both operating and extreme conditions. A conservative analysis of the data obtained from the test program indicates the following:

a) The overall G factor measured without harmonic analysis is more than twice the value obtained when corrected for frequency.
b) Extreme load conditions occur very seldom and the cumulative number of such loadings over the life of a tank is a maximum of 22,000 assuming 3 occurrences per day every day for a 20 year life. This is well below the frequency requiring fatigue analysis.

Cost/Benefit Considerations: This change in requirements affects tanks that are small diameter, long and thin and generally made of stainless steel. If the changes are adopted, the tank wall thickness of some of these tanks can be reduced. Since these tanks generally operate with full loads if the weight is reduced, the tank can carry more and fewer lighter tanks can carry the same amount of total product. This can lead to an operating savings. This change is the same as recommended for DOT 400 series tanks which have the same structural integrity requirements and is made to keep both sections of the regulations consistent with each other.

Revised Text of Regulations

49CFR178.34537-3©(1)(iii)C – Change to read – The tensile or compressive stress generated by the bending moment resulting from normal operating vertical force equal to .25 *(formerly .35)* times the vertical reaction at the suspension assembly of a trailer; or the horizontal pivot of the upper coupler (fifth wheel) or turntable; or anchoring and support members as applicable. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the cargo tank wall.

49CFR178.337-3© (1)(iv)(A) - Change to read – The vertical shear stress generated by a normal operating accelerative force equal to .25 *(formerly .35)* times the vertical reaction at each suspension assembly of the trailer; or the horizontal pivot of the upper coupler (fifth wheel) or turntable; or anchoring and support members of the truck as applicable. The vertical reaction must be calculated based on the static truck as applicable. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the cargo tank wall.

49CFR178.337-3©(2)(iii)© - Change to read – © The tensile or compressive stress generated by the bending moment resulting from an extreme vertical acceleration force equal to .45 *(formerly .7)* times the vertical reaction at the suspension assembly of a trailer, and the horizontal pivot of the upper coupler (fifth wheel) or turntable; or the anchoring and support members of a truck; as applicable. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the tank wall.

49CFR178.337-3©(2)(iv)(B) – Change to read – (B) The vertical shear stress generated by an extreme vertical acceleration equal to .45 *(formerly .7)* times the vertical reaction at the suspension assembly of a trailer; and the horizontal pivot of the upper coupler (fifth wheel) or turntable; the anchoring and support members of a truck, as applicable. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the tank wall.

DOT 400 SERIES CARGO TANKS
PROPOSED REVISIONS TO 49CFR178.345-3 STRUCT. INTEGRITY

Description of recommended changes:

a) In 178.345-3©(1)(iii)B and 178.345-3©(1)(iv)B, change the factor .35 to .25 for vertical accelerative force for normal operating loadings.
b) In 178.345-3©(2)(iii)B and 178.345-3©(2)(iii)B, change the factor .7 to .45 for vertical accelerative force for extreme dynamic loadings.

Rationale: A recent over the road test of a cargo tank motor vehicle in which the dynamic response was subjected to a harmonic analysis to determine the distribution of accelerations over a wide frequency range indicated that the factors in the present regulations are well above what contributes to design load for both operating and extreme conditions. A conservative analysis of the data obtained from the test program indicates the following:

a) The overall G factor measured without harmonic analysis is more than twice the value obtained when corrected for frequency.
b) Extreme load conditions occur very seldom and the cumulative number of such loadings over the life of a tank is a maximum of 22,000 assuming 3 occurrences per day every day for a 20 year life. This is well below the frequency requiring fatigue analysis.

Cost/Benefit Considerations: This change in requirements affects a large class of tanks that are small diameter, long and thin and generally made of stainless steel. If the changes are adopted, the tank wall thickness of these tanks can be reduced about .020". Since these tanks generally operate with full loads if the weight is reduced, the tank can carry more and about 1 percent fewer thinner tanks can carry the same amount of total product. This can lead to an operating savings alone of about $1000 per year per tank. With a population of 3000 such tanks, annual savings amount to about $3 million, not an insignificant sum. Other tanks are affected as well but not quite to the same degree so that total savings would be higher than this.

Revised Text of Regulations

49CFR178.345-3©(1)(iii)(B)(4) – Change to read – The tensile or compressive stress generated by the bending moment resulting from normal operating vertical force equal to .25 *(formerly .35)* times the vertical reaction at the suspension assembly of a trailer; or the horizontal pivot of the upper coupler (fifth wheel) or turntable; or anchoring and support members as applicable. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the cargo tank wall.

49CFR178.345-3© (1)(iv)(B) - Change to read – The vertical shear stress generated by a normal operating accelerative force equal to .25 *(formerly .35)* times the vertical reaction at each suspension assembly of the trailer; or the horizontal pivot of the upper coupler (fifth wheel) or turntable; or anchoring and support members of the truck as applicable. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the cargo tank wall.

49CFR178.345-3©(2)(iii)© - Change to read – The tensile or compressive stress generated by the bending moment resulting from an extreme vertical acceleration force equal to .45 *(formerly .7)* times the vertical reaction at the suspension assembly of a trailer, and the horizontal pivot of the upper coupler (fifth wheel) or turntable; or the anchoring and support members of a truck; as applicable. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the tank wall.

49CFR178.345-3©(2)(iv)(B) – Change to read – The vertical shear stress generated by an extreme vertical acceleration equal to .45 *(formerly .7)* times the vertical reaction at the suspension assembly of a trailer; and the horizontal pivot of the upper coupler (fifth wheel) or turntable; the anchoring and support members of a truck, as applicable. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the tank wall.

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ECONOMIC IMPACT OF CERTAIN PROPOSED CHANGES TO REGULATIONS FOR DOT 400 SERIES CARGO TANKS

A recent report of a road test on a DOT 407/412 type cargo tank recommends certain changes in the structural integrity requirements for 400 series tanks. This report analyzes the economic impact of some of these changes on the country.

Two recommendations are related and call for revising 49CFR178.345-8 to reduce dynamic load G values to .25G instead of .35G for vertical forces in transit when combined with lateral and longitudinal forces, and to .45G from .7G when vertical forces only are considered. Vertical forces due to dynamic loads are controlling in the design of fully loaded long thin wall small diameter low pressure (40psi and below) tanks, such as 48″ to 64″ diameter and 40 feet long made of stainless steel up to about .188″ thick. There is a fairly large population of these tanks. An analysis indicates that the required thickness of these tanks can be reduced by .020″. In a 60″ diameter by 40′ tank, a weight savings of over 500 lbs. occurs. This is about one percent of the payload and, since these tanks are usually fully loaded, a tank with 1% less weight can haul 1% more product. This means that 1% fewer tanks (including tractors) can satisfy constant demand and transportation costs would be lowered by 1% also. A cargo tank of this size probably costs at least $100,000 per year to operate and a 1% savings would amount to $1000 per year per tank. There are probably at least 3000 such tanks in service presently, perhaps more. The yearly savings for 3000 tanks is $3 million, not an insignificant sum. These recommendations do not involve the ASME Code and can be adopted without changing the basic Code.

Two other recommendations involve ASME since one proposes that weld efficiencies for circumferential buttwelds in shells can be 100% for structural stresses due to longitudinal bending while retaining established Code efficiencies for pressure stresses. In the same types of tanks as considered above, the longitudinal bending stress in tension is often controlling and amounts to over 98 percent of the total. For most DOT 407 and 412 tanks, ASME mandated weld efficiencies are 70%, so conservatively, these tank thicknesses can be reduced by 25 percent if this recommendation can be implemented. In some cases, compressive stresses may become controlling is thicknesses are reduced that much. The other recommendation is to have ASME increase allowable compressive stresses which will allow tensile stresses to be controlling at lower thicknesses than at present. Service experience with MC 307 and 312 tanks, which were not designed for 1.7G vertical forces and 70% higher longitudinal bending moments, indicates that allowable bending stresses, both tensile and compressive, could be increased substantially without sacrificing safety. Considering that both recommendations would be adopted, it is reasonable to assume that for most 40 foot long DOT 407 and 412 tanks, wall thicknesses, exclusive of corrosion allowances, could be reduced by 15 percent. For a 60″ diameter by 40′ tank, could be reduced from 10 gage (.135″) to 12 gage (.105″), and 22% reduction. This would reduce the weight of such a tank by about 1.5% and this would be independent of the reduction in the previous paragraph. The yearly savings per tank would be about $1500 and would apply to a wider spectrum of tanks than considered previously, possibly treble the number or 9000, for a total annual amount of $13.5 million.

To summarize, the first two recommendations would reduce the operating cost of a significant number (at least 3000) of tanks by $1000/year and their first costs by about $500 at a conservative allowance of $1/lb. The second two recommendations would lower the operating cost of at least 9000 tanks by about $1500 per tank per year and, at $1/lb for material, would lower the first cost per tank by $770.

Supporting calculations are attached to this report. It is hoped that this information, while somewhat limited and erring on the conservative side, would be useful in evaluating the Road Test Report.

M. R. Ward, P.E. 5/24/03
CTMA Technical Manager

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May 24, 2003

U.S. Department of Transportation
Research and Special Programs Administration
400 Seventh Street NW
Washington, DC 20590-0001

Attention: Stan Staniszewski, Office of Hazardous Materials Technology

Subject: DOT Order TRS56-02-P-70045, Dynamic Analysis of DOT 407/412 Cargo Tank Motor Vehicles.

Dear Stan:

Enclosed is the final report on the subject purchase order and has been amended in accordance with your comments on the draft we sent you earlier. We have also added a short report giving an economic analysis of the changes we are recommending in the Cargo Tank Regulations illustrating their fiscal impact on the country. The report includes copies of the Wyle Laboratories data and computer analysis on which our final report is based. This is a cooperative project between industry and government where costs and effort were shared. We believe the results of this project are tangible and important and were well worth the modest expenditure involved. We hope to participate in future such activities.

Respectfully,

M.R. Ward, P.E.
CTMA Technical Manager

Cc: Wyle Labs, Thompson Tank, Weld-It, Paramount Tank, West-Mark, Beall Tank

TITLE ECONOMIC CALCULATION SHEET No. EC1 OF ____
JOB No. ____ BY MRW CKD ____ DATE 5/29/03 REV ____ DATE ____

FOR A 60" ϕ x 40' TANK WITH 25 psig INTERNAL PRESSURE MADE OF 304 STAINLESS $\sigma_{ALLOW} = 17.5$ ksi $E = 28{,}000$ ksi

ASSUME LOADED TANK WEIGHT = 60K
@ 1.7G $M_{MAX} = 4590$ in K — @ 1.45G $M_{MAX} = 3915$ in K

IF TENSION CONTROLS DESIGN W/WELD EFF'Y = .70
SECTION MODULUS $= \frac{\pi D^2}{4} t \cdot .7 = 1979.2t = Z_T$

TRY t = .138" FOR $M_{MAX} = 4590$ in K

$$\sigma_{TENSION} = \frac{M_{MAX}}{Z_T} + \frac{PD}{4t} = \frac{4590}{1979.2 \cdot .138} + \frac{.0025 \cdot 60}{.7 \cdot 4 \cdot .138} = 17.45 \text{ ksi} < 17.50 \text{ ksi } \underline{o.}$$

TRY t = .118" FOR $M_{MAX} = 3915$ in K

$$\sigma_{TENSION} = \frac{3915}{1979.2 \cdot .118} + \frac{.0025 \cdot 60}{.7 \cdot 4 \cdot .118} = 17.51 \text{ ksi} \approx 17.50 \text{ ksi } \underline{O.K.}$$

FOR .020" SHELL THICKNESS DIFFERENCE, TANK WEIGHT DIFFERENCE IS .020 · 40.8 · 5π · 40 = 513 LB. FOR 60K TANK, PAYLOAD IS ABOUT 54K 513 LB IS ABOUT 1% OF PAYLOAD. THE LIGHTER TANK WILL CARRY 1% MORE PRODUCT

ASSUME TANK COSTS \$60,000 & TRACTOR COSTS \$70,000 DRIVER EMPLOYMENT COST IS \$50,000 PER YEAR, FUEL COST IS \$33,000 PER YEAR & MAINTENANCE IS \$11,000 PER YEAR
TOTAL COST FOR 20 YEARS IS \$130,000 + 20(94,000) = \$2,010,000

COST SAVING WITH 1% MORE CAPACITY IS \$20,100 OVER TRUCK LIFE OR \$1005 PER YEAR. IF THERE IS A POPULATION OF 3000 UNITS, ANNUAL SAVINGS ARE <u>\$3,015,000</u> <u>@ \$1/lb FIRST COST DROPS \$513</u>

<u>THIS IS FOR CHANGING G FACTOR FROM 1.7G TO 1.45G</u>

Naval Architecture
Civil Engineering
Mechanical Engineering

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(805-567-0280)

TITLE ECONOMIC CALCULATION SHEET No. EC2 OF ____
JOB No. ____ BY WRW CKD ____ DATE 5/29/03 REV ____ DATE ____

FOR COMPUTING BENDING STRESS USING 100% WELD EFFICIENCY & 70% FOR PRESSURE STRESS, ASSUME SAME TANK

FOR 1.7G CASE & $t = .136''$ $\sigma_{TENSION} = 12.45$ ksi @ 70% WELD EFF'Y

IF WELD EFF'Y FOR BENDING IS 100% & FOR PRESSURE IS 70%

IF $t = .136''$ $\sigma_{TENSION} = \frac{4590 \cdot .7}{1979.2 \cdot .136} + \frac{.0025 \cdot 60}{.7 \cdot 4 \cdot .136} = 12.33$ ksi

IF $t = .106''$ $\sigma_{TENSION} = 12.33 \cdot \frac{.136}{.106} = 15.82$ ksi CONSERVATIVE!

FOR .030" THICKNESS DIFFERENCE, TANK WEIGHT DIFFERENCE IS $513 \cdot \frac{.030}{.020} = 770$ lb OR APPROX 1.5% OF PAYLOAD

2.56

USING THE SAME COST DATA BUT ASSUMING 9000 TANKS OBTAINING SUCH SAVINGS

YEARLY SAVINGS/TANK $= \frac{1.5 \cdot 1005}{1.0} =$ $1508/YR OR $30,150 IN 20 YRS

FOR 9000 TANKS YEARLY SAVINGS $= 1508 \cdot 9000 =$ $13,500,000

@ $1/lb FIRST COST DROPS $770

Naval Architecture
Civil Engineering
Mechanical Engineering

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PROGRAM CYLTK CALCS TENSILE AND COMPRESSIVE LONG'L SHELL STRESSES

$M := 4590 \qquad D := 64 \qquad t := .25 \qquad P := 25$

$Z := 3.14159 \cdot D \cdot D \cdot t \cdot .25 \qquad s_c := \frac{M}{Z} \qquad s_b := \frac{M}{(.7 \cdot Z)} \cdot 1 + P \cdot \frac{D}{(4000 \cdot t)}$

$Z = 804.247 \qquad s_c = 5.707 \qquad s_b = 8.153$

$M_1 := 3915 \qquad s_{c1} := \frac{M_1}{Z} \qquad s_{b1} := \frac{M_1}{(.7 \cdot Z)} \cdot 1 + P \cdot \frac{D}{(4000 \cdot t)}$

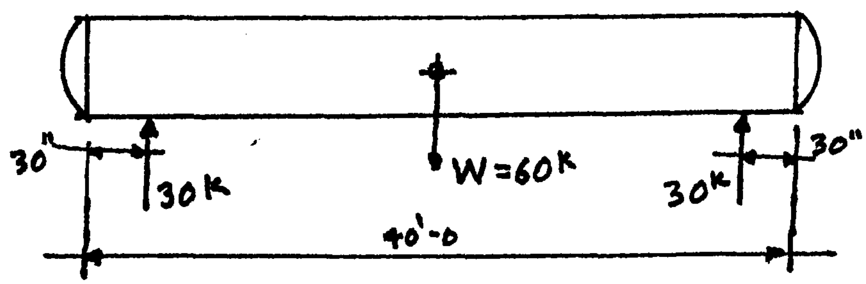
$s_{c1} = 4.868 \qquad s_{b1} = 6.954$

Calculate Bending Moments for a 40ft tank with supports 30 in from Tangent line
Tank weight is 60 kips. Moments are in kip inches.

For 1G $\quad M_{max} := 30 \cdot 12 \cdot (17.5 - 10) \qquad M_{max} = 2.7 \cdot 10^3$

For 1.45G $\quad M_{max} := 1.45 \cdot 2700 \qquad M_{max} = 3.915 \cdot 10^3$

For 1.7G $\quad M_{max} := 1.7 \cdot 2700 \qquad M_{max} = 4.59 \cdot 10^3$



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REPORT OF STUDY ON THE DYNAMIC ANALYSIS OF DOT 407/412 TYPE CARGO TANK MOTOR VEHICLES

Reference: U. S. Dep't of Transportation Order No. TRS56-02-P-70045 of 9/25/02

TABLE OF CONTENTS

REFERENCE DOCUMENTS

Wyle Laboratories Test Report No. 47977 of 2/24/03 (2 copies furnished)

Engineer in Charge: M. R. Ward P. E.
RTL Inc. RR#1 Box 91, Hollister Ranch
Gaviota, CA 93117 (805-567-0280)

Approved: M.R. Ward Date 5/24/03

REPORT ON CARGO TANK ROAD TEST

This report describes the road tests of a 6700 gallon 12 gauge (.105") type 304 stainless steel semi cargo tank performed from Jan. 20 to Jan. 23, 2003 and its results. This test was performed under a joint program funded by the Cargo Tank Manufacturers' Association and the U. S. Department of Transportation (P. O. TRS56-02-P-70045). The purpose of the test was to determine the vertical accelerations of a typical type DOT407/412 tank when subjected to dynamic loads caused by various kinds of roads and road obstructions likely to be encountered in service. The important feature of the testing was to do a harmonic analysis of the test data to find out how each harmonic contributed to the structural loads experienced by the tank. The present DOT regulations contain G factors (gravity factors) that specification tanks must meet. It is known that the structural effects of all the harmonics are not additive in creating stresses in the tank. There have been similar analyses on manufactured housing and tank trailers and such investigations should be done on liquid carrying trailers to get a more complete picture of the dynamic load situation.

DESCRIPTION OF TEST: - The test was performed on a newly built fifth wheel double rear axle semi tank 64" diameter by 480" long transported by a standard truck. The tank lacked its normal insulated jacket so that it could be instrumented easily. Four accelerometers were placed on the tank unit, one each at the forward end, the middle and the rear end of the tank and one on the front rear axle. The accelerometers were connected to a recording system which could be turned on and off to gather data at significant times during the test. The tank was fully loaded with water up to within 1% of its maximum legal carrying capacity of 80, 000 lbs (including tractor unit) and run twice over a predetermined route on public roads near Norco, California and data was recorded for the following road conditions:

a) Relatively smooth asphalt road
b) Gravel road (unpaved)
c) Rough paved road with numerous potholes which were driven over and not avoided.
d) Paved road with a steel plate barrier equivalent to a railroad crossing.
e) Double dip paved intersection.

The same test was done for a half full tank and an empty tank so that a representative amount of data was obtained. The half load test was particularly severe in that the tank was a single compartment with no internal baffles to damp out sloshing.

The data from the test runs was analyzed by computer (a Fourier analysis) to develop the accelerations associated with the various harmonics. The most extreme of the runs were analyzed for each load condition. Six runs at differing road conditions for each load condition were analyzed and the G factors for each harmonic up to the ninth were determined for each accelerometer. Also the overall root mean square (RMS) value for each run was determined so that G factors for other tests could be compared. A report from Wyle Labs is attached to this report, which includes all the data runs analyzed, and some information on the instrumentation and computer analysis they did.

The certified gross vehicle weights (GVW) for the three test conditions were as follows: full load 79740 lb, half load 53240 lbs, and empty 24220 lbs. These weights include the tractor and running gear. The main area of interest is the loads on the tank itself. The light weight of the tank is assumed to be 5000 lbs uniformly distributed. This gives an assumed uniformly distributed load for the full tank of 60.52 kips, for the half full tank 34.02 kips, and for the empty tank 5 kips. Certified GVW's do not include the weight of the driver and other occupant. There were two people in the truck during the test. This added about 500 lbs to the GVW's.

Analysis of Test Results – The results of the test indicated that there were two obvious natural frequencies for the dynamic response of the tank trailer assembly during every test run. It appears that the tank assembly on two sets of elastic supports is a two degree of freedom system with the tank acting as a rigid body. A check of expected natural frequencies indicate that, for the full load condition, the most important one, the lowest natural frequencies for the two degree of freedom system are around 1-2.5HZ and for the tank itself vibrating as a uniform beam 30-60HZ. For the full load condition, the two lowest natural frequencies were 1.48HZ and 2.14HZ for the most severe shock. For other lesser shocks, the values varied slightly but were within a fairly narrow range. The tank experienced Root Mean Square (RMS) accelerations of up to 1.3G excluding the static weight of the tank, which is higher than the DOT Regulation 49CFR178.345-3 maximum of .7G vertical acting alone and .35G acting both vertically and horizontally (when resolved amounting to .395G). According to the physics of sinusoidal harmonic motion, the deflection associated with each mode of vibration varies in proportion to the G value for each mode determined by Fourier (Harmonic) analysis divided by the square of the mode number. This means that the effect of a total G force on tank stresses must take into account the contribution of each mode of vibration. It is incorrect to take an RMS acceleration reading from a dynamic test of a tank and apply it directly to the mass of the tank to obtain forces, moments and stresses without doing a harmonic analysis so that the contribution of each vibration mode can be computed separately and added in a logical manner to obtain realistic stresses in the tank. In doing this, the phase of each mode must be considered as well. Odd modes have maximum deformations 90 degrees out of phase from even modes and this must be considered as an interaction rather than simply adding them all together.

Analyses of extreme data obtained from the several test runs have been performed. One such analysis is presented below. It represents the highest G value (1.262G RMS) and occurred in the middle of the tank when going over a steel plate laid over a ditch on a paved road at approximately 40mph.

Mode No.	Freq (HZ)	Odd G	G/n^2	Even G	G/n^2
1	1.48	.4117	.4117		
2	2.96			.0618	.0135
3	4.44	.0245	.0027		
4	5.93			.0128	.0008
5	7.41	.0121	.0004		
6	8.89			.3482	.0097
7	10.37	.1897	.0039		
8	11.85			.1467	.0023
9	13.33	.1478	.0018		
Totals		.7858	.4203	.5705	.0283

Total Values

G_{RMS}=1.262

Odd G + Even G = 1.3563

Odd G/n^2+ Even G/n^2 = .4288

Mode No.	Freq (HZ)	Odd G	G/n^2	Even G	G/n^2
1	2.14	.0477	.0477		
2	4.27			.0193	.0048
3	6.41	.0022	.0002		
4	8.95			.1040	.0065
5	10.69	.0978	.0039		
6	12.82			.5902	.0164
7	14.56	.1121	.0023		
8	17.10			.0550	.0009
9	19.24	.0367	.0005		
10	21.37			.0242	.0002
Totals		.2965	.0546	.7857	.0288

Total Values

G_{RMS}=1.262

Odd G + Even G = 1.0862

Odd G/n^2 + Even G/n^2= .0834

There were six runs each of data for three tank loading conditions, full load (60,520 lb), half load (34,020 lb), and empty (5,000 lb). There were three accelerometers on top of the tank and one on a rear axle. On the tank, they were mounted at the forward end, middle and aft ends and, except for two runs over a smooth road, there were two obvious natural frequencies analyzed. For a reasonably rigid body such as the tank, mounted on springs at its ends, it is a two degree of freedom system from an analytical standpoint. This means that one should expect two major natural frequencies for vertical motion which was the case. One of the frequencies is for translation up and down and the other is rotation of the tank about a transverse horizontal axis. As a result of all this, there were 102 G value spectra recorded and analyzed. The axle accelerometer was analyzed but was recorded as a basis of comparison to other possible dynamic tests where accelerometers might have been used. The measured overall root mean square (RMS) G values for the tank and axle were as follows:

Load Condition	Tank Min Val	Tank Max Val	Axle Min Val	Axle Max Val
Full Load (60.52K)	.192G	1.290G	.824G	4.152G
Half Load (34.02K)	.196G	2.688G	.586G	5.622G
Empty (5.00K)	.391G	1.245G	1.516G	3.472G

The half full tank, having no interior swash plates or baffles and being 40 feet long, experienced appreciable sloshing which may have increased the G values from those of the other two load conditions which had no sloshing.

A harmonic analysis was made of each run at each frequency for a total of 102 analyses. The G factor for each mode was divided by the square of the mode number to obtain the contribution of each mode to the total G factor to be used in structural design. These contributions were added together in two separate groups, odd modes in one group and even modes in the other. For the vast majority of runs, ten modes were included. The logic for this is that, for the eleventh mode, the measured G factor for that mode must be divided by 121, the square of the mode number, to obtain its contribution to the total effect. Since its effect is less than one percent of its measured value, it and higher modes can be ignored in this type of analysis. The summations of odd and even modes were combined by squaring each of them, adding them and taking the square root to obtain an overall G factor to be used in design. The ratio of this G factor to the total RMS G factor was also computed for each case. The distribution of these G factors was as follows for all cases analyzed:

Overall G factor	0G to .1G	.1G to .2G	.2G to .3G	.3G to .4G	Over .4G	Total
No. of Values In range	67	19	9	5	2	102

The two highest G values were .4449G (on the half full tank) and .4215G (on the full tank). The .4449G value when multiplied by the ratio of full load weight to half load weight gives a value of .2501G for the equivalent full load G force applied to the tank. Clearly the full load G factor is a greater load on the tank.A conservative design approach would be to take the .4215G and add .02G to it for shock effects, that is, effects of modes past the tenth mode. This would result in a maximum G factor for design of .45G. The present regulation maximum is .7G based presumably on a total RMS G value of .7G. For the particular case in question, the tank RMS value was 1.262G or 80 % more than .7G. Factoring the .45G down by the ratio .7/1.262 gives approximately .25G. It would appear that the worst case design factor of .45G is ultraconservative and would be a low cycle fatigue situation at best occurring perhaps twice a day for a tank life of 20years operating every day. This amounts to 14,610 times in the tank life, a low number for a fatigue case. With this number of cycles, fatigue should not be a factor for designs based on normal allowable stresses. In fact it might be reasonable to allow the 20 % stress increase for the worst case structural design for DOT 400 series tanks.

The theoretical natural frequency of the loaded tank on end supports was computed to be about 40HZ. This is about 25 times more than the lowest natural frequency of the loaded tank on its suspension. The suspension acts as a dynamic vibration or shock absorber because its stiffness is so much lower than the tank itself. That this is true is indicated by an analysis of the higher frequency spectrum of the same load case analyzed above where the total RMS G value was 1.262. The results are as follows:

Mode	Freq(HZ)	Odd G	Odd G/n^2	Even G	Even G/n^2	Totals
1	38.50	.0506	.0506			
2	72.99			.0075	.0019	
3	109.49	.0037	.0004			G_{rms}=1.262
4	149.98			.0114	.0007	Odd G + Even G=.1006
5	182.48	.0108	.0004			Odd G/n^2+Even G/n^2=.0544
6	218.97			.0101	.0003	$((OddG/n^2)^2+(Even\ G/n^2)^2)^{.5}$=.0516
7	255.47	.0029	.0001			(Above is 4.09% of G_{rms})
8	291.96			.0016	.0000	
Totals		.0680	.0515	.0326	.0029	

These results confirm that the suspension acts as a dynamic vibration absorber for the higher frequency shocks liable to excite flexural vibration of the tank on its supports. The test tank was long and thin. Flexural natural frequencies would be higher for most other tanks which are shorter and thicker.

Summary of Test Results: - The test results indicate the following:

a) G factors used for establishing dynamic loads on highway tanks should not be based directly on overall G factors obtained from accelerometers mounted on tanks.
b) It is necessary to do a harmonic analysis on accelerometer data used to establish practical allowable stress values for dynamic load conditions on highway cargo tanks.
c) The magnitude of present dynamic load G factors may be too conservative as presently applied to tank design and should be evaluated based on harmonic analysis of the data used to determine them.
d) As required by DOT Regulations, ASME specified values for allowable longitudinal compressive stresses in highway tanks appear to be too conservative and may not take into account the fact that tanks under maximum longitudinal compressive stress are full of product and have positive internal pressure which would reduce their tendency to buckle over tanks with external pressure. This is substantiated by the large number of MC 306 and 307 tanks as well as food grade tanks which continue to perform satisfactorily in service even though some, as the tank used in this test, do not even meet the longitudinal compressive stress requirements under the static load condition with no dynamic loads at all.
e) Evidence indicates that the most severe dynamic stress conditions occur rarely enough so that fatigue may not be a factor in design for them.
f) In severe dynamic load cases, the test tank was overstressed in longitudinal bending, primarily because of the 70 percent joint efficiency requirement of the Regulations. The major component of longitudinal tensile bending stress in a long and thin tank comes from the longitudinal bending moment, not from the membrane stress. It may be desirable to ignore the joint efficiency factor where longitudinal bending stress is over 2/3 of the maximum total longitudinal tensile stress.

Recommendations: - There are two kinds of recommendations emanating from this project. One is suggested changes to existing regulations and the other is additional projects that might be useful. The recommendations consider the dynamic load test results as well as the experience with the very thin wall tank used in the test and the stresses it experienced in the conduct of the testing.
The recommended changes to regulations are:

a) Change the G factors for dynamic loads from .35G to .25G in vertical direction for dynamic load cases with combined acceleration, deceleration, and .2G lateral loads (The .2G lateral load is probably due to centrifugal force and not vibration and should remain as it is).(49CFR178.345-3(c)(1)(iii)(B) &(C))
b) Change the G factor for maximum vertical load to .45G instead of .7G and do not consider this load subject to fatigue because it occurs so seldom. (49CFR178.345-3(c)(2)(iii)(B) & (C) and 49CFR178.345-3(c)(2)(iv)(A) &(B))
c) For longitudinal bending in tension, allow the weld efficiency factor of .7 to be 1.00 for the bending portion of the total stress and keep the .7 factor for the membrane stress as required by the ASME Code. (New item 49CFR178.345-3(b)(3)). Review with ASME.
d) Ask ASME to revisit its latest work on compressive stress in bending of cylindrical shells to account for the case where there is internal pressure in the vessel combined with longitudinal bending. In discussions with those who did the work for ASME resulting in UG-23 and Code Case 2286, it appears that the cases considered were for external pressure combined with longitudinal bending, an entirely different condition than obtains in a loaded cargo tank where bending stresses are maximum.

Recommendations for further work are:

a) Run road tests similar to this one on a wider variety of highway, rail and intermodal tanks, such as LPG tanks, cryogenic tanks with jackets, non-cylindrical DOT 406 tanks, and heavier DOT 407/412 tanks with different suspension systems and configurations (truck mounted tanks, pull trailers and semi tanks) to get a broader spectrum of data to analyze.
b) Develop standard procedures for analyzing accelerometer readings from dynamic tests to establish reasonable parameters for developing design load factors using the additional data from a). This

would consider impact or shock loads in addition to vibratory loads. They would also consider as well the effect of different suspension systems on mitigating such loads on the tanks.

c) Run some tests and do analyses to establish lateral force limits for highway cargo tanks. Lateral loads on these tanks are usually limited by overturn considerations, not by vibratory or shock forces.
d) Study thin wall long tanks to establish safe and realistic design parameters and stresses for them. There is ample evidence from the service history of large numbers of MC 300 series tanks and non-hazardous food grade tanks to indicate that safe tanks can be built with less critical and expensive high alloy material.

Some of these recommendations overlap in certain areas but can be dealt with usually as separate projects.

Acknowledgements: - This project was a cooperative effort of industry and government. The following companies, institutions and individuals contributed to its success: Gary Spoelstra and West-Mark supplied the tank, the tractor unit that hauled it and the driver, Bob Ramos. Wyle Laboratories, Norco, CA provided the accelerometers, the recording equipment and the harmonic analysis of the data. Ron McCarthy, Dan Cook and Tom Balfry of Wyle were very helpful. Tom Rogers, P. E. of Container Technology, Lubbock, TX took time out from his regular business and contributed to the success of the test and in the analysis of the results. Thompson Tank Inc. (Dave Thompson Sr. and Dave Thompson Jr.) provided video and photographic equipment. Kurt Van Diest and his staff supplied and disposed of the test water. Weld-It (Ray Schaffer), Paramount Tank (Howard Grey) and Beall Tank provided funding and accounting services. U. S. Department of Transportation provided funding for the project and Stan Staniszewski handled the project a on the government side and got it underway on the government side. RTL Inc represented by M. R. Ward, P. E. supervised the test and wrote the report. The Cargo Tank Manufacturers' Association, whose membership includes Beall, Paramount Tank, Thompson Tank and West-Mark, sponsored the industry portion of the project.

Appendix A – Test Tank Particulars

The tank used for testing was a new almost completed 304 Stainless Tank, 64" diameter by 40' long with no interior baffles, with torispherical heads dished to .10 percent of the tank diameter and supported 30" aft of the front head seam and 30.5" forward of the rear head seam. It had external stiffeners at spacings not exceeding 60 inches. The tank was built for hauling 6700 gallons of milk at 8.34 lbs/gal and was not equipped with its normal insulating jacket which consists of styrene and glass fiber insulation with a thin stainless steel cover. The tank had no access ladders when tested. The Gross Vehicle Weight (GVW) of the tank trailer, including insulation, two dual axle running gear, insulation and appurtenances is specified at 68,000 lbs. The empty tank as configured was assumed to weigh 5,000 lbs. The tank was tested under three conditions, namely fully loaded with 55,520 lbs of water (approximately 6657 gallons), approximately half load with 29,040 lbs of water, and empty with no water. The tank was equipped with a pressure relief valve set to open at 1 psig and, when operating full, was assumed to have 1 psig pressure at the top of the liquid. During the test there was some indication that a tiny amount of water had escaped from the tank which would indicate that it did have 1 psig internal pressure plus the static head of water during the test runs, at least for the full tank.

This tank meets the minimum thickness requirements for DOT 406, 407 and 412 tanks, namely .100" for stainless steel with a gallons per inch of up to 14 (actual is 13.81). It does not meet the longitudinal strength design requirements for either MC 306, 307 and 312 tanks nor for DOT 406, 407 0r 412 tanks. MC 300 series tanks must be designed for the static load condition under pressure and DOT 400 series tanks must be designed for that condition plus vertical dynamic load conditions of up to 1.7G. Table A-1 shows the calculated moments and stresses in the fully loaded tank under various load conditions specified in the Regulations as well as assuming the worst overall vertical G force experienced for all the test runs.. The longitudinal shell membrane stress was calculated from the static pressure at the middle of tha tank, including 1 psig top pressure, applied uniformly around its circumference. This slightly underestimates the membrane stress at the bottom of the tank and overestimates it at the top of the tank but this stress is insignificant compared to the longitudinal bending stress and is apparent by looking at the table. The G force adds to the apparent weight of the water and that is taken into account in computing the internal pressure. The tank shell, except for longitudinal compressive stress, is suitable for 25 psig internal pressure. The heads would also be suitable except they are dished less than allowed by the ASME Code.

Table A – 1

			Jt Effy			1.0JE	.7JE
G force	Int Press	Bd Mom	S_{pres}	S_{bdg}	S_{top}	S_{bot}	S_{bot}
1.0	2.155	2761	.328	8.774	-7.846	8.502	12.146
1.35	2.560	3727	.390	11.034	-10.644	11.424	16.320
1.395	2.612	3851	.398	11.401	-11.003	11.800	16.856
1.70	2.964	4693	.452	13.894	-13.442	14.346	20.484
2.262	3.613	6245	.551	18.488	-17.937	19.039	27.199
1.25	2.444	3451	.372	10.217	-9.845	10.589	15.127
1.45	2.675	4003	.408	11.851	-11.443	12.259	17.513
	psig	inK	ksi	ksi	ksi	ksi	ksi

According to DOT regulations, DOT 407 and 412 tanks must be designed in accordance with ASME Code stresses or 25 percent of ultimate tensile strength. Prior to the adoption of a 3.5 design margin, ASME used a 4.0 design margin, the same as 25 percent of ultimate tensile strength. The ASME rules for allowable compressive stress are more complicated, involving factors other than tensile strength. Using ASME rules for this tank, the allowable tensile stresses are 17.50 ksi for a design margin of 4.0 and 20.0 ksi for a design margin of 3.5. For allowable compressive stress, two values are permitted, one from UG-23 of Section VIII Division 1, and one using Code Case 2286. The corresponding allowable compressive stress values are 5.742 ksi for UG-23 and 7.50 ksi for Code Case 2286. The ASME Code requires a 70 percent weld joint efficiency for tanks built without 100 percent X-ray of weld seams which is a requirement for both MC and DOT tanks. The actual stress in the weld is based on 100 percent joint efficiency.

The table indicates that this tank does not meet the allowable compressive stress requirements for any of the load conditions shown, exceeding it by 2.34 times for the 1.7G condition required for DOT 400 series tanks and by 1.37 times for the static load condition, based on UG-23. The corresponding factors for Code Case 2285 are 1.79 times for 1.7G and 1.05 times for static load. Considering tensile stress, the tank exceeds the allowable of 25 percent of ultimate by a factor of 1.17 assuming 70 percent joint efficiency and 1.02 for a design margin of 3.5, all for the 1.7G load condition. For 100 percent joint efficiency and 1.7G load, the calculated tensile stress is within allowable limits for the 1.7G load condition. This tank is representative of thousands of non-hazardous materials tanks that are standard for the food industry and have been in service for many years. These tanks have experienced dynamic loads at least as severe as those encountered by hazardous material tanks since they travel over the same roads. The road tests which this tank survived had overall G factors significantly higher than those specified in the regulations. This would indicate at least two things, namely that dynamic load requirements seem to be conservative and also that allowable compressive stress requirements are even more conservative and may need to be revised. There is some evidence that ASME allowable longitudinal compressive stresses are based on a vessel being subject to external pressure along with longitudinal bending moments. For fully loaded tanks subject to maximum longitudinal bending moments, they are full of liquid and are subject to internal pressure, not external pressure so the basis for establishing them may be too conservative and not representative of actual service conditions.

It would appear to be worthwhile to explore the possibility of using higher allowable longitudinal compressive stresses for highway cargo tanks in view of the preponderant evidence that present allowables seem to be very conservative. It might also be appropriate to consider that the 70 percent joint efficiency does not apply to longitudinal bending stresses since, for structural loads of other Codes, such as the AISC Specification for the Structural Design, Fabrication and Erection of Structural Steel for Buildings, joint efficiency of full butt welded joints is assumed as 100 percent.

Appendix B – Tank Flexural Vibration

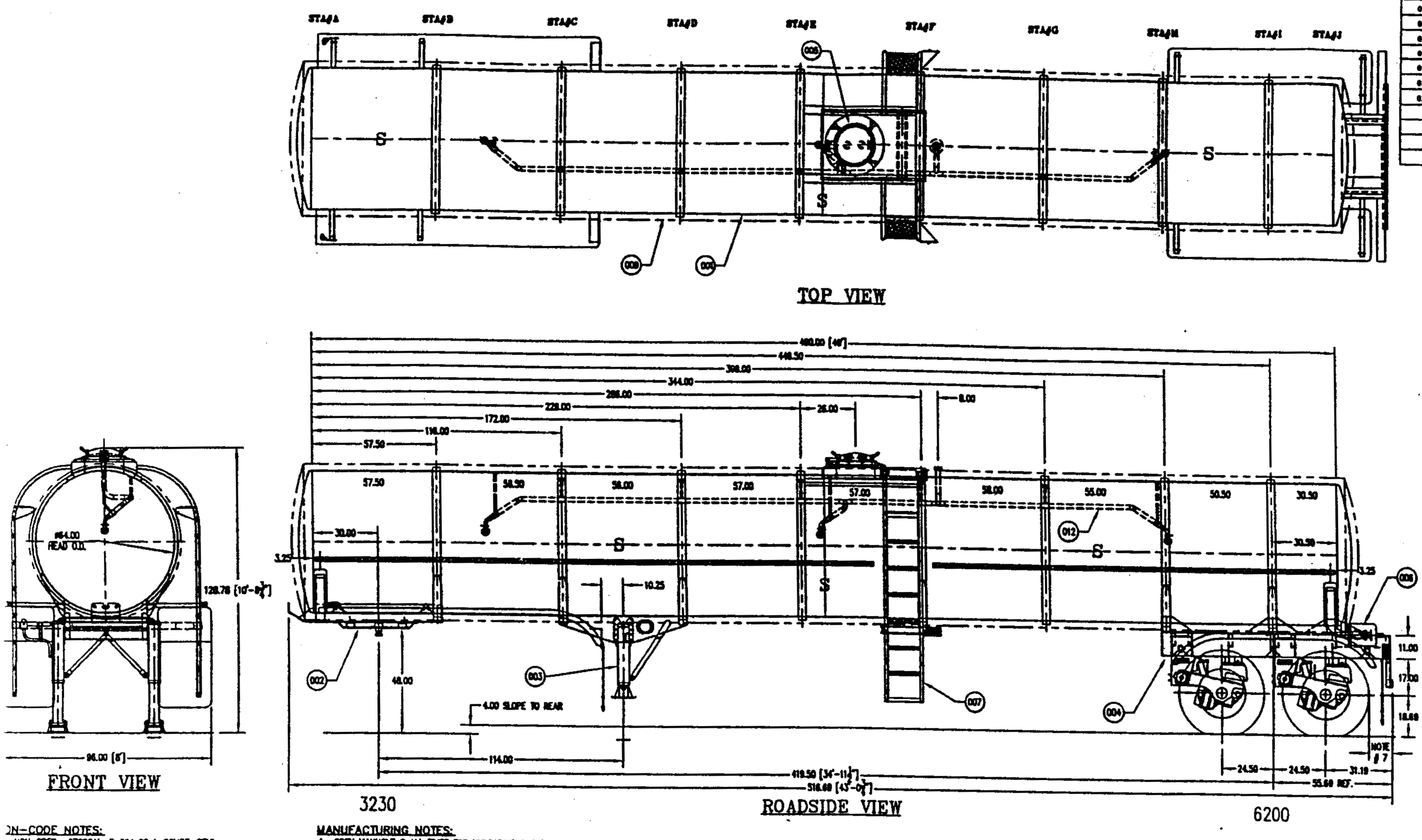
A long semi trailer cargo tank has many modes of vertical vibration. Firstly it vibrates as a rigid body mounted on springs at its ends as essentially a two degree of freedom system. The primary modes have frequencies on the order of 1 to 3HZ for each degree of freedom. In addition, the tank itself can vibrate flexurally with infinite degrees of freedom with the primary mode having a frequency of 20 to 100HZ, an order of magnitude higher than the rigid body modes. The tank used in the test was 40 feet long made of stainless steel. It was filled with water as the simulated payload. If a filled tank is subject to a shock exciting vibration, the shock wave travels at the speed of sound both through the tank shell and through the liquid. The speed of sound in steel is about 16,000 ft/sec and in water is about 4800 ft/sec. A shock wave, if undamped, would take .005 seconds to travel the length of the tank and return in the shell and.0167 seconds in the water. This is equivalent to frequencies of 200HZ and 60HZ respectively. The tank supports are about 35 feet apart. For a tank traveling at 60 mph, the time interval between shocks emanating from the same bump in the road would be about .4 seconds apart, equivalent to a frequency of about 2.5HZ. If traveling at 30 mph, the frequency would be about 1.25HZ, approximately in the primary mode range of the tank as a rigid body on springs. It is then possible that for a certain resonant speed, the two shocks would supplement each other and cause a more severe rigid body response than if they were not resonant. When driving on a smooth road such as a freeway, the travel speed would be in the 60 mph range, while on a rougher road, the speed would be more likely to be in the resonant range. When the travel speed is less the shock is less which would tend to reduce the magnitude of the shock. The kinetic energy created by a shock is probably roughly proportional to the square of the speed so that half speed would result in one quarter the shock. Two shocks in resonant sequence would at worst double the energy of one shock so travel speed may have more effect than bump severity.

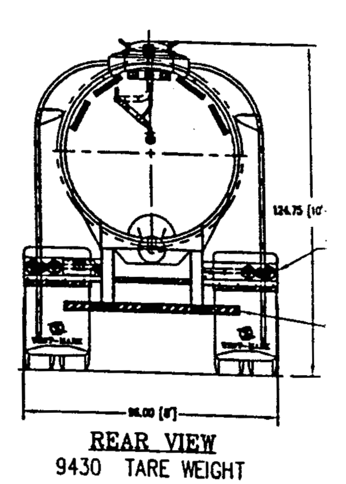
The road test measured accelerations at three points on the tank. It did not measure deformations or deflections. Stress in the tank is proportional to the deformation or deflection, not to the acceleration so it is necessary to convert acceleration to deflection for each mode of oscillation. The relationship is derived as follows:

For harmonic motion, deflection is $X = A\sin(pt)$ where X is deflection, A is maximum amplitude, t is time and p is twice pi times the natural frequency in Hertz (cycles per second). If the deflection formula is differentiated twice, the result is the acceleration namely $X'' = -Xp^2$

The deflection and hence the stress for each mode of vibration is proportional to the acceleration divided by the square of the mode number. A shock response can be divided into modes starting from the primary mode by doing a Fourier analysis of the accelerometer trace for a particular point on the tank. The first mode is the primary frequency, the second mode is twice the primary frequency and so on. The higher the mode the more complete and accurate is the analysis of accelerations. However, since the effect on stress is inversely proportional to the inverse square of the modal frequency, the effect of the tenth mode is only one hundredth of that for the first mode for the same magnitude of acceleration, a negligible effect. Normally the first few modes have the largest accelerations and the higher modes have the lowest so it is reasonable to ignore the highest modes. There is a further factor in harmonic analysis. The maximum amplitudes for the odd modes are close together and the amplitudes of the even modes are located away from those of the odd modes. This means that odd and even modes should be considered independently in harmonic analysis. The procedure followed in this report is to sum the odd mode values of acceleration, divide by the mode number squared, sum those for the even modes and combine them by taking the square root of the sum of their squares. This is a conservative approach. In almost all cases the odd modes are predominant as might be expected.

ITEM	QTY.	CAT	MATERIAL
001	1	SA	L ASSEMBLY 6700 12GA S/S 84 STD
002	1	CA	FLER ADJ S/S FILLER
003	1	LGXX.	NDING GEAR JOST 19" S/S FRAME
004	1	RS341803	STD REAR SF T/A AIR-RIDE NITRONIC S-S
005	1	MH181A81	MILK MANHOLE DECK PLATE W/OLSEN VENTS
006	1	OTXX1M03	3" 60TTF MILK VALVE ASSY. W/ S/S COVER
007	1	LDXX3M07	LADDER (2) MILK-ALUM SK-1252
008	1	LWXXXA12	LED LIGHT & WIRING 2-LIGHT SEMI W/2EA
009	1	J871S04	S/S JACKET W/ 3" STY INSUL STD MILKER
010	1	BKXS2S2M	BRAKE KIT T/A WABCO 2S2M
011	1	FKXX11A81	FENDER BRACKETS CTR F&R T/A SEMI S.S
012	1	CPXX3A01	CP. 3-BALL TOP MT ACME THRD
020	1	PMMILKER	MISC ITEMS AND CHARGES
022	1	RBXX1A00	BUMPER NON-CODE S/S 3" X 2"





N-CODE NOTES:

NON-SPEC. 6700GAL. T-304 SS 1-COMPT. SEMI.
PVC PROTECT SHELL AND HEADS I.D.
INTERIOR WELDS FINISHED TO W-S.
EXTERIOR WELDS FINISHED TO W-0.
OFFSET ALL LONGITUDINAL SEAMS 2.0", 1/4" WEEP HOLES
REQ'D ON PADS, RINGS, BOLSTERS @ LOWEST POINT.
HYDROTEST FINISHED UNIT @ 3 P.S.I., M.A.W.P. = 1 P.S.I.
MAX TEMPERATURE 165° F.
HEADS BUILT TO 64.00" O.D., WM DISHED & FLANGED
BUTT-WELD, FULL PENETRATION.
UNIT WAS DESIGNED TO HAUL MILK @ 8.34#/GAL. MAX GPI=13.92
GVWR=68,000 LBS.
INSTALL DOT/WM PLATE ON SIDE FILLER (ROAD SIDE).
RELIEF VENTING REQ'D AT ALL TIMES: (OLSEN)
INSTALL ALL COMPONENTS TO 3-A SANITARY STANDARDS.
WELD WIRE TO BE USED: ER-316L SI.

MANUFACTURING NOTES:

1. OPEN MANHOLE @ ALL TIMES FOR UNLOADING & CLEANING.
2. 60TTF OUTLET VALVE TO FACE ROADSIDE.
3. INSULATE SHELL WITH 3" OF STYRENE. INSULATE HEADS WITH 4" OF FIBERGLASS.
4. ELECTRICAL LINES TO RUN 13" OFF BOTTOM CENTERLINE OF TANK TO DRIVERSIDE.
5. INSTALL WABCO 2S2M ABS BRAKE SYSTEM.
6. LANDING LEG CRANK HANDLE INSTALLED CURBSIDE OF UNIT.
7. TIRE TO END OF FRAME 12" MAX = EXEMPT UNDERRIDEMENT BUMPER
8. ALL BOLSTERS ARE 4.00" HIGH @ BOTTOM C/L OF TANK.

1 EA 34846
1 EA 34847
1 EA 34848
1 EA 34849

SYM.	DATE	BY	REVISION
REVISIONS			

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
HUNDREDTHS ± .03
ANGLES ± 1/2°
THOUSANDTHS ± .005
DIM BETWEEN HOLES ± .015

W-M STANDARD
6700GA. T-304 S/S T.A.
NON-CODE MILK TANK

WEST-MARK
2704 RAILROAD AVE. CERES, CA 95307

DWG BY: TEL | SCALE 1/2"=1' 1:24
SHEET 1 OF 1
DATE 10-28-02 | JOB NO. X | DWG NO. 3464

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