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THOMPSON TANK, INC.

ASME - D.O.T. CERTIFICATION

D.O.T. INSPECTIONS - TESTING

DESIGN ENGINEERING - CONSTRUCTION

THOMPSON VACUUM-PRESSURE UNITS

June 29, 2004

U.S. Department of Transportation
Research and Special Programs Administration
Office of Hazardous Materials Standards
400 Seventh Street, S.W.
Washington, D.C. 20590

RSFA-2004-17074-3

*06/29/04
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Attention: Mr. Edward Mazzullo

Reference: Request for Rulemaking on Dynamic Load Factors for Cargo Tanks. P-1433

Gentlemen:

We strongly support the above identified petition P-1433 submitted by CTMA and RTL Engineering Inc., Mr. M.R. Ward P.E., copies enclosed.

The time has come to start providing some foundation for the load requirements specified in the D.O.T. Regulations that are realistic and determined by good engineering practices.

Best regards,
THOMPSON TANK, INC.

David L. Thompson
President

Enclosures
DLT/es

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To: Cargo Tank Manufacturers and Users
From: Monte Ward

5/3/04

Subject: Request for Rulemaking to DOT on Dynamic Load Factors for Cargo Tanks

The Cargo Tank Manufacturers' Association submitted a request for rulemaking to DOT to lower the vertical dynamic load factors for 400 series and 331 cargo tanks from .35 to .25 for combined dynamic load conditions and from .7 to .45 for maximum vertical load conditions. The applicable regulations are 178.345-3 and 178.337-3, both of which read the same. This request is based on the results of many years of operating experience with MC306, 307 and 312 tanks as well as DOT331 and 400 series tanks. It is also based on the results of a cooperative project between CTMA and DOT in which a 407/412 type tank was run over rough roads and measurements were taken of the dynamic response which were subjected to a harmonic analysis to determine the G factors associated with each mode of vibration or dynamic response. Normally dynamic responses are reported as overall G factors and these G factors are applied to the static weight of the tank to determine longitudinal stresses in the tank. Since overall G factors incorporate all modes of dynamic response and the primary mode is the most critical and higher modes are significantly less critical or out of phase with the primary mode, the overall G factor approach is ultraconservative and leads to overestimates of longitudinal stress. For many long, thin wall tanks of smaller diameter, the longitudinal stress is controlling in the design so that tanks are significantly thicker sometimes than they really need to be. This is also borne out by the successful operating history of MC307/312 tanks, which were designed without consideration of the increase in design load mandated by the 400 series G factors. It is also borne out by the vast number of long food grade tanks of small diameter and thin wall which, for many years, have given excellent service even though overstressed longitudinally according to DOT regulations cited.

There is no doubt that dynamic loadings must be considered in the design of hazmat tanks especially. The proposed rulemaking, I believe, does this in a reasonable and conservative manner. In order to have this rule change considered, support from significant numbers of manufacturers and operators must be documented and sent to DOT. This is a request for interested parties to consider the merits of this proposal and support it in writing if they agree that it is reasonable and useful and that the public will benefit by receiving lower cost service from more efficient tanks designed to the proposed rules.

Some particular points that can be made are: 1) Semi tanks in the ranges of 40 feet long, 12 and 10 gage thick and 48" to 66" diameter can often be overstressed by present rules but will qualify under the revised rules; 2) Many MC 307 and 312 tanks were designed without considering the stress increases mandated by the present dynamic load factors, and which would not qualify as 400 series tanks, yet they have not had longitudinal bending failure in service; 3) Thousands of food grade tanks subject to the same service dynamic loads as hazmat tanks and of significantly less robust construction are successful in service; 4) A thinner lighter tank can carry more payload and such tanks usually carry full loads. This means that fewer tanks will be needed to satisfy demand resulting in lower cost and greater safety; 5) There appears to be no downside to this proposal.

This information is offered to see if it is desired by the industry and to obtain feedback on its utility and importance in doing a better job in serving the public. Comments are welcome as well as responses to DOT. Comments to DOT should refer to Request P-1433 and go to:

U. S Department of Transportation
Research and Special Programs Administration, Office of Hazardous Materials Standards
400 Seventh St. SW, Washington, DC 20590 Attention : Edward Mazzullo

Thank you for your consideration.

Respectfully,

Monte Ward

CARGO TANK MANUFACTURERS' ASSOCIATION

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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

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REFERENCE DOCUMENTS

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

Engineer in Charge: M. R. Ward P. E.
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Approved: M. R. Ward -

Date 5/29/03

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	<u>Total Values</u>
5	7.41	.0121	.0004			G _{RMS} =1.262
6	8.89			.3482	.0097	Odd G + Even G = 1.3563
7	10.37	.1897	.0039			
8	11.85			.1467	.0023	Odd G/n ² + Even G/n ² = .4288
9	13.33	.1478	.0018			
Totals		.7858	.4203	.5705	.0283	

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			<u>Total Values</u>
4	8.95			.1040	.0065	G _{RMS} =1.262
5	10.69	.0978	.0039			
6	12.82			.5902	.0164	Odd G + Even G = 1.0862
7	14.56	.1121	.0023			
8	17.10			.0550	.0009	Odd G/n ² + Even G/n ² = .0834
9	19.24	.0367	.0005			
10	21.37			.0242	.0002	
Totals		.2965	.0546	.7857	.0288	

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	67	19	9	5	2	102
In range						

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	
1	38.50	.0506	.0506			
2	72.99			.0075	.0019	<u>Totals</u>
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 Or 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 the 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

G force	Int Press	Bd Mom	Jt Effy		S _{top}	1.0JE	.7JE
			S _{press}	S _{bndg}		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.



U.S. Department
of Transportation
Research and
Special Programs
Administration

400 Seventh St., S.W.
Washington, D.C. 20590

DEC - 5 2003

M.R. Ward, P.E.
Technical Manager
Cargo Tank Manufacturers Association
P.O. Box 790
Lakewood, CA 90714-0790

Dear Mr. Ward:

This acknowledges the receipt of your November 1, 2003 letter requesting an amendment of 49 CFR 178.345-3 to revise structural integrity for DOT 400 Series Tanks and 49 CFR 178.337-3 to revise structural integrity for DOT 331 cargo tanks.

Your request has been assigned number P-1433. This number is for tracking purposes only. Your request has not been assessed for merit. If accepted under 49 CFR 106.95 as a petition for rulemaking, it will be considered for a future rulemaking action.

If you write or call us regarding this petition, please make refer to this number.

Sincerely,

Gail L. Twitty
Transportation Assistant
Office of Hazardous Materials Standards

CARGO TANK MANUFACTURERS' ASSOCIATION

P. O. BOX 790
LAKEWOOD, CALIFORNIA 90714-0790
(800-421-7545)

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

Nov. 1, 2003

Attention: Phil Olson, Office of Hazardous Materials Technology

Subject: Petition for Rulemaking, DOT331, 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 Sept. 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 of you need more information.

Respectfully,

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

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

Description of recommended changes:

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

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

Description of recommended changes:

- a) In 178.337-3~~C~~(1)(iii)~~C~~ and 178.337-3~~C~~(1)(iv)~~A~~, change the factor .35 to .25 for vertical accelerative force for normal operating loadings.
- b) In 178.345-3~~C~~(2)(iii)~~C~~ and 178.345-3~~C~~(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~~C~~(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~~C~~(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 weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the cargo tank wall.

49CFR178.337-3~~C~~(2)(iii)~~C~~ - Change to read - ~~C~~ 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~~C~~(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.