U.S. Department
of Transportation

**Pipeline and Hazardous
Materials Safety
Administration**

1200 New Jersey Avenue, SE
Washington, D.C. 20590

APR 2 9 2008

Mr. Gary Spoelstra
Chief Engineer
West-Mark
P.O. Box 100
Ceres, California 95307

Ref. No.: 07-0031

Dear Mr. Spoelstra:

This is in response to your January 10, 2007 letter requesting clarification of the Hazardous Materials Regulations (HMR; 49 CFR Parts 171-180) applicable to cargo tanks. Your questions are based on an interpretation issued in 1993, in which we specified that DOT 400 series tanks could be designed without considering weld efficiencies less than 100% and still meet the requirements in the HMR. You state that Section XII of the ASME Code, which we will be incorporating into the HMR, will invalidate the 1993 interpretation by mandating the use of full design stress at 70% joint efficiency on a large population of tanks. The result of this would generate heavier tanks, a situation not in keeping with the present regulations if not properly addressed. Specifically, you ask for us to allow a 20% increase in allowable compressive and tensile stress for longitudinal bending in cargo tanks when considering the extreme case of .7G as specified in § 178.337-3(c)(2)(iii)(C) for MC 331 cargo tanks and paragraphs (c)(2)(iii)(C) and (c)(2)(iv)(B) of § 178.345-3 for DOT 400 series cargo tanks.

Based on the information provided, we have determined that it is acceptable to use the long standing ASME Section VIII criteria in UG-23(d) of a 20% increase in allowable stress for certain conditions. For DOT 400 series cargo tanks, the allowable compressive and tensile stress may be increased 20% when analyzing longitudinal bending in cargo tanks when considering the extreme load case of .7G as specified in § 178.345-3 (c)(2)(iii)(C) and (c)(2)(iv)(B). We will be addressing this issue in a future rulemaking.

Please note that although ASME UG-23(d) may be an appropriate alternative criterion to use in the analysis of certain loading conditions for other cargo tank specifications, we are limiting your request to that which is directly related to the co-operative research effort between DOT and industry for 400 series cargo tanks and the supporting documentation provided in your letter.

I hope this information is helpful. If you have further questions, please do not hesitate to contact this office.

Sincerely,

Hattie L. Mitchell
Chief, Regulatory Review and Reinvention
Office of Hazardous Materials Standards

Satterthwaite
§178.337-3(c)(2)
§178.345-3(c)(2)
Cargo Tanks
07-0031

WEST-MARK

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

Jan, 10-07

Attention: Stan Staniszewski, Office of Hazardous Materials Technology

Subject: Request for Interpretation of Cargo Tank Rules

Dear Stan,

This letter is a request to modify the dynamic loads for which DOT 331 and 400 series Cargo Tanks must be designed. Specifically the request is to allow the use of a 20% stress increase when analyzing DOT 331 and 400 series tanks considering a .7G vertical load "extreme" dynamic force increase for these tanks. The history behind this request is somewhat complicated. and will be explained in the following paragraphs.

Originally MC 300 series cargo tanks were required to be designed for static load conditions. When the rules were revised to accommodate DOT 400 series tanks where dynamic loads of up to .7G (the most extreme and so labeled) were superimposed on the static loads, the regulations stated that design would be in accordance with the ASME Code Section VIII which required weld joint efficiencies of 70% if welds were not tested non-destructively (x-rayed or ultrasonically inspected). The critical stresses were midspan longitudinal cmpression and tensile stresses in bending particularly in compression due to the thin walls This mandated that such tanks would have heavier walls than the MC 300 tanks for a large number of tanks. Accordingly, in 1993, DOT issued an interpretation that DOT 400 series tanks could be designed without considering weld efficiencies less than 100% and still meet the regulations. As long as the population of these tanks was under only DOT regulation, the interpretation was valid. However, there will be a special Section XII of the ASME Code covering the structural design of all cargo tanks which DOT will incorporate into its regulations by reference. This will mandate the use of full design stress at 70% joint efficiency on a large population of tanks and cause them to be heavier, a situation not in keeping with being transparent to present regulations if this request is not honored.

In 2002, a long light gage cargo tank of DOT 407/412 type equipped with accelerometers was over-the-road tested and the data analyzed harmonically to see what equivalent static load increases replicated the summation of the dynamic forces on the tank. A report on this test is attached. This was done as a cooperative project between a group of cargo tank manufacturers and DOT. It was found that the worst case equivalent dynamic condition conservatively stated was .42G, not .7G and that tanks designed by the new criterion with 100% joint efficiency were as safe or safer than MC 300 tanks as well as the enormous population of food grade tanks of lighter construction, many of which remain in service for 30 years or more. It was also found that only one of 102 measured events in the test achieved this transitory load increase so it is not a common dynamic condition.

The present interpretation allowing 100% joint efficiency is incompatible with ASME standards, To achieve the necessary relief, it is proposed that DOT allow a stress increase of 20% (the same increase allowed for wind and seismic forces in Section VIII) when using the extreme dynamic load factor of .7G. This can be incorporated into Section XII and assure transparency for this case. This also deals positively with the lesser but still significant problem of tensile overstress when using 70% joint efficiency.

We request that DOT issue an interpretation allowing a 20% increase in allowable compressive and tensile stress for longitudinal bending in cargo tanks when considering the extreme case of .7G. The applicable parts of the regulations are 49CFR178.337-3(c) (2)(C) for MC 331 tanks and 49CFR178.345-3(c)(2)(iv)(B) and (iii)(C) for DOT 400 series tanks.

Since MC331 tanks have the same extreme design load conditions as 400 series tanks, such an interpretation might be applicable to them. We don't build MC331 tanks and defer to those building such tanks to recommend the same interpretation for them.

We hope this request is adequate for its intended purpose and await your action.

Monty Ward P.E.
RTL Inc.

Gary Spoelstra
Chief Engineer
West-Mark

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			Total Values
4	5.93			.0128	.0008	G_{RMS}=1.262
5	7.41	.0121	.0004			
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			Total Values
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 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
1	38.50	.0506	.0506		
2	72.99			.0075	.0019
3	109.49	.0037	.0004		
4	149.98			.0114	.0007
5	182.48	.0108	.0004		
6	218.97			.0101	.0003
7	255.47	.0029	.0001		
8	291.96			.0016	.0000
Totals		.0680	.0515	.0326	.0029

Totals

G_{rms}=1.262

Odd G + Even G=.1006

Odd G/n^2+Even G/n^2=.0544

$((\text{Odd}G/n^2)^2+(\text{Even }G/n^2)^2)^{.5}$=.0516

(Above is 4.09% of G_{rms})

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 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.

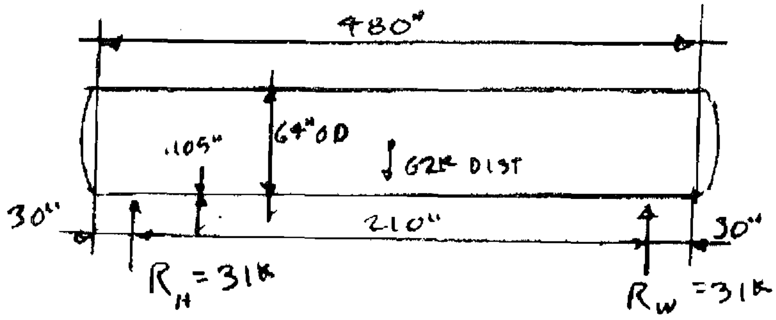
The road test measured accelerations at three points on the tank. It did not measure deformations or deflections. Stress in the tank is proportional to the deformation or deflection, not to the acceleration so it is necessary to convert acceleration to deflection for each mode of oscillation. The relationship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

CYL TANK STRESS ANALYSIS
LONG'L COMPRESSIVE BENDING

CALCULATION SHEET **RTL Inc.**

TITLE 64"φ × .105" WALL × 480" STLS TANK SHEET No. ____ OF ____
JOB No. ____ BY MW CKD ____ DATE 1|10|07 REV ____ DATE ____



MAT'L 304 STLS UTS = 84 ksi
YP = 36 ksi E = 28,000 ksi
STATIC LOAD IS 62k UNIFORMLY DISTRIBUTED, EXTREME VERTICAL LOAD IS 1.7G (105.4k)
$D/t = \frac{64}{.105} = 609.52$

DETERMINE MAX COMPR STRESS IN SHELL UNDER 1.7G EXTREME LOAD

$M_{MAX} = 31 \cdot 1.7(210 - 120) = 4743$ in k $Z_{SHL} = \pi r^2 t = \pi \cdot 32^2 \cdot .105 = 321.70\ in^3$

$\sigma_C = \frac{4743}{321.30} = 14.74$ ksi   IF ALLOWABLE STRESS IS INCREASED 20% FOR EXTREME LOAD, ASME ALLOWS 20% INCREASE FOR WIND & SEISMIC STRESSES (UG-23)

$\sigma_{C_{ALL}} = \frac{14.74 \cdot 1}{1.20} = 12.29$ ksi

ASME WILL ALLOW USE OF CODE CASE 2276-1 EQUATIONS 3-3b & 3-4 OF SECTION 3.2.1 WITH $F_Y = 36$ ksi & $E_t = 28,000$ ksi
FS = 1.9

EQ 3-3b $\sigma_{C_{ALL}} = \frac{466\ F_Y}{(331 + D/t)(FS)} = \frac{466 \cdot 36}{(331 + 609.52)\ 1.9} = 12.79$ ksi > 12.25 ksi O.K.

EQ 3-4 $\bar{c} = 1.0$ $D/t = 609.52$ $E_t = 28,000$ ksi (TANGENT MODULUS)

$C_x = \frac{409\ \bar{c}}{389 + D/t} = \frac{409 \cdot 1}{389 + 609.52} = .4096$

$\sigma_{C_{ALL}} = \frac{C_x E_t t}{D_o (FS)} = \frac{.4096 \cdot 28,000 \cdot .105}{64 \cdot 1.9} = 13.49$ ksi > 12.25 ksi O.K.

CONCLUSION TANK MEETS CODE REQ'TS USING 20% STRESS INCREASE FOR EXTREME 1.7G LOAD CONDITION

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