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GUIDELINES FOR STRUCTURAL EVALUATION OF CARGO TANKS

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CHAPTER 1. INTRODUCTION

This document has been prepared for use as a guideline for engineers performing reviews of structural designs of cargo tanks. These reviews are usually performed as part of the Compliance Reviews performed by Safety Investigators.

The Secretary of Transportation, under 49 CFR Part 1-48, has granted the authority to perform Compliance Reviews to the Federal Highway Administration (FHWA) and more specifically, the Office of Motor Carriers (OMC). It is OMC's responsibility to regulate interstate commerce on our nation's highways as it pertains to truck and bus transportation. Since cargo tanks fall under this responsibility, it is necessary for OMC to ensure the operation and construction of these tanks are in compliance with the Code of Federal Regulations (CFR). The Office of Motor Carrier's authority within the cargo tank realm has been extended beyond strictly transportation, to encompass not only the design and construction, but the continuing qualification and maintenance of cargo tanks as well. The portions of the federal regulations pertaining to these areas may be found in 49 CFR Parts 178 & 180. One of the tools the Office of Motor Carriers utilizes to evaluate regulatory compliance is the Compliance Review, performed by Safety Investigators (SI's). The evaluation of cargo tank manufacturing facilities will, under many circumstances, be performed by SI's with the assistance of an engineer. Under these conditions, the engineer will be responsible for performing the structural evaluation portion (49 CFR Part 178) of the Compliance Review. The SI will be responsible for evaluating the facility's compliance with Part 180, which was established for the continuing qualification and maintenance once the tanks have been constructed and placed in service.

These guidelines emphasize structural evaluation of DOT 400 series cargo tanks. Most of the cargo tanks on the roads today are MC 300 series cargo tanks, however, as of September 1, 1995, the 300 series tanks are no longer authorized for construction.

THE DOT 406, 407, and 412 specification cargo tanks will replace the MC 306, 307, and 312 tanks. The manufacture of DOT 400 series tanks was authorized beginning on December 31, 1990, and some manufacturers have already elected to produce the 400's. It is important to know that those companies choosing this option are required to design and construct in accordance with the specification in effect at the time of manufacture.

Specifications for MC 306 tanks (49 CFR 178.,340 and 178.341) became effective on December 1, 1967. The MC 306 is characteristically used for gasoline, fuel oil, alcohol, and other liquid flammables. Typically, the 306 is constructed of aluminum, is designed for atmospheric pressure, and has an elliptical cross section. Specification DOT 406 cargo tanks (49 CFR 178.320, 178.345, and 178.346) have replaced the 306's, effective September 1, 1995.

Specifications for MC 307 tanks (49 CFR 178.340 and 178.342) became effective on December 1, 1967. The MC 307 is characteristically used for solvents, plasticizers, casinghead gas, etc. Typically, the 307 is constructed of stainless steel, is designed for a pressure of at least 25 psig, and has a circular cross section. Specification DOT 407 cargo tanks (49 CFR 178.320, 178.345, and 178.347) have replaced the 307's, effective September 1, 1995.

Specifications for MC 312 tanks (40 CFR 178.340 and 178.343) became effective on December 1, 1967. The MC 312 is characteristically used for corrosive materials. Typically, the 312 is constructed of stainless or carbon steel, is designed for a pressure of 35 psig, and has a circular cross section. Specification DOT 412 cargo tanks (49 CFR 178.320, 178.345, and 178.348) have replaced the 312's, effective September 1, 1995.

Some tanks are manufactured to meet multiple specifications, such as a DOT 407/412. This is generally done in order for the user to have the ability to haul a wider range of materials.

Other specification cargo tanks exist that will not be addressed in this document. Specification MC 331 (49 CFR 178.337) and MC 338 (49 CFR 178.338) cargo tanks are high pressure vessels which demand the use of the ASME Code for their design and construction. These tanks are typically used for atmospheric gases or cryogenics. The 331's are designed for pressures anywhere from 100 to 500 psig, whereas the 338's are designed for pressures between 25.3 and 500 psig.

The structural evaluation portion of the Compliance Review must consider the requirements for:

1. Structural integrity of the tank
2. Bottom damage protection
3. Overturn damage protection
4. Rear-end damage protection

Portions of Part 178 in Title 49 of the Code of Federal Regulations must be followed for each of the four items above, and various analysis methods must be employed, some of which are included in this document. Generally, the loads imposed on cargo tanks and their components are dynamic in nature. It is the intent of the regulations to simplify and idealize these loads into equivalent steady-state conditions, and to utilize these idealizations in static analyses for the purposes of computing stresses (allowable loads).

Details of analysis procedures for many items that must be included in a structural evaluation are contained in this document; however, this manual does not address every specific detail that may be encountered. A few additional important references are the PRESSURE VESSEL HANDBOOK, and the ASME BOILER AND PRESSURE VESSEL CODE, Section VIII, Division 1, and Section II, Part D. These publications, along with other applicable reference materials, are listed in References.

It is important to note that in most cases, DOT 400 series specification cargo tanks are to be designed and constructed in accordance with the ASME Code, with some exceptions. Ensure that you have access to the Code in effect at the time of construction, and that you are versed in the exceptions indicated in each specification.

The layout of this manual was arranged in this manner purposefully. In most cases, it indicated the natural progression of steps to be completed when performing a structural evaluation of a cargo tank during a Compliance Review. It should be noted that many facilities manufacture to more than one specification, and in many cases, more than one design within each specification. It will be the responsibility of the engineer to evaluate the manufacturer's design methods and techniques on a case by case basis. Common sense would dictate that a company building 406's, all of the same design, would necessitate one analysis for that design. If a company is more versatile and builds custom 406's, 407's, and 412's of varying designs, the logical approach would be to determine the design types producing the worse case loading scenarios, and evaluate those designs.

Geometric shapes of some cargo tanks are shown in figure 1.

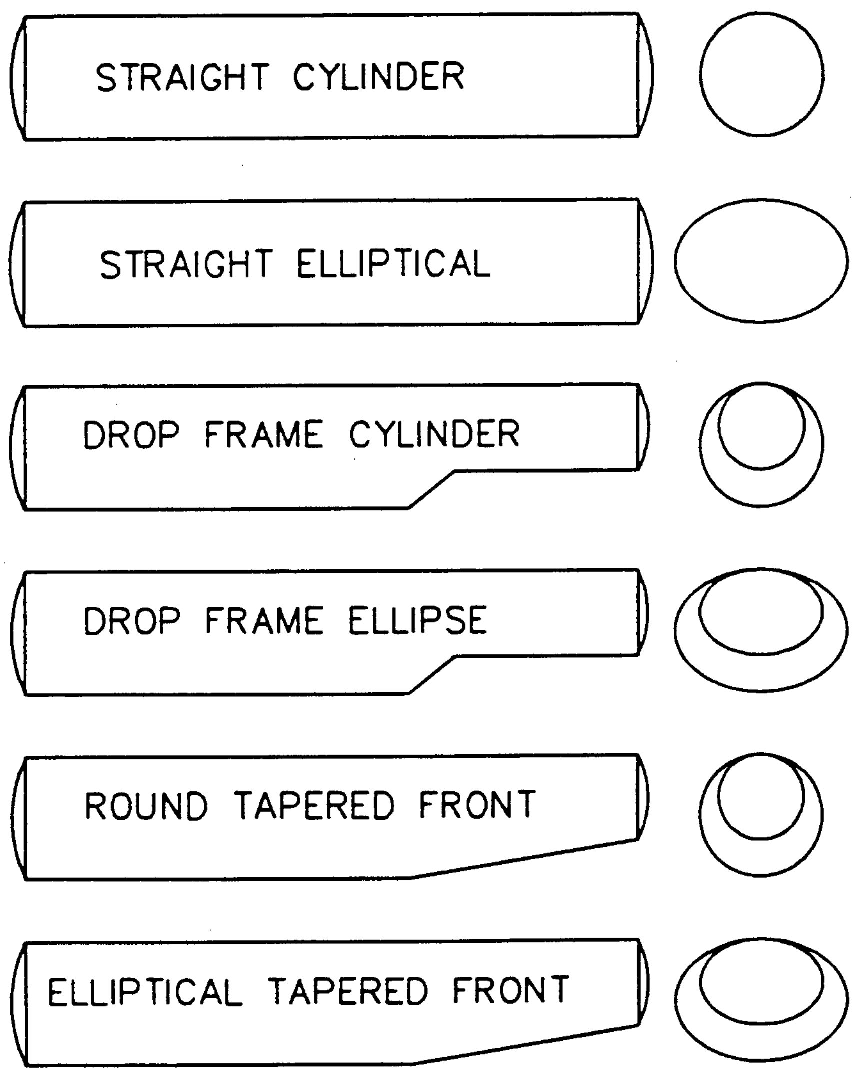

Figure 1. Common shapes of cargo tanks.

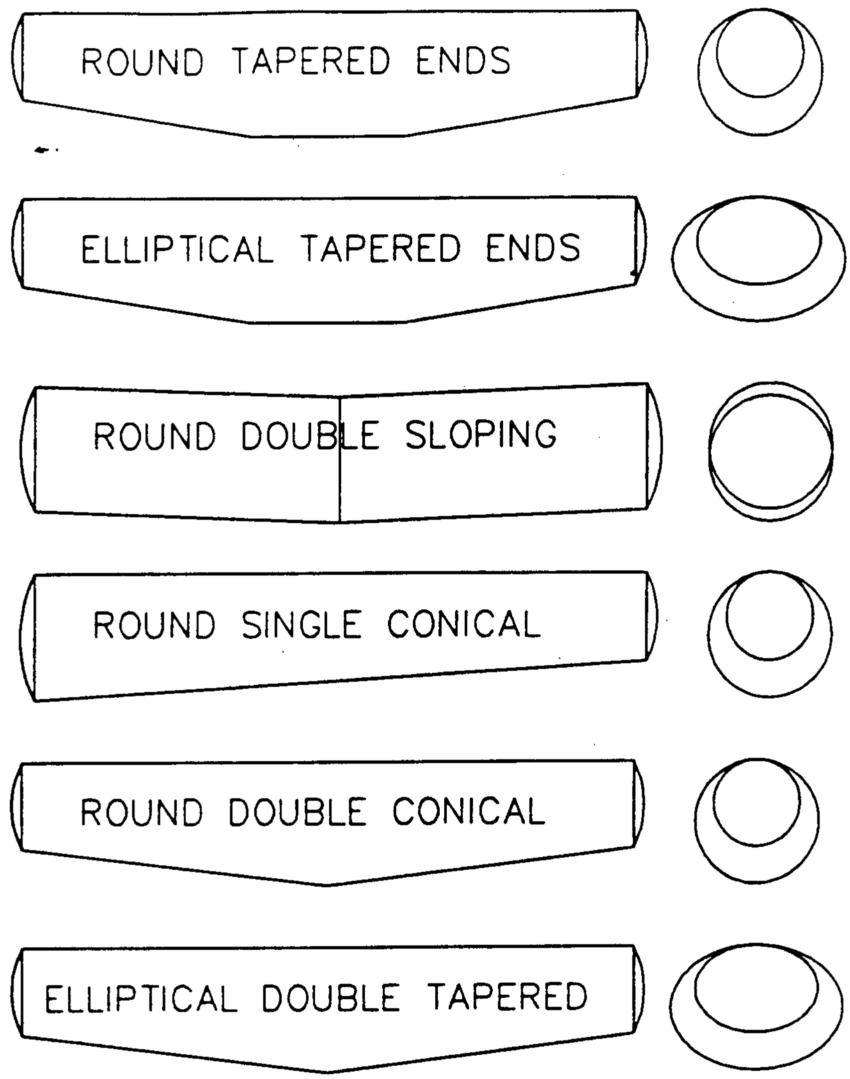

Figure 1. (Continued) Common shapes of cargo tanks.

Geometric shapes of various types of heads and baffles are shown in figure 2 through 6. Torispherical and ellipsoidal heads are frequently used on low-pressure tanks and hemispherical heads are frequently used on tanks designed for higher pressures.

Heads are formed from flat sheets of material. The forming process causes a reduction in thickness of the material which must be taken in account in design and construction of cargo tanks.

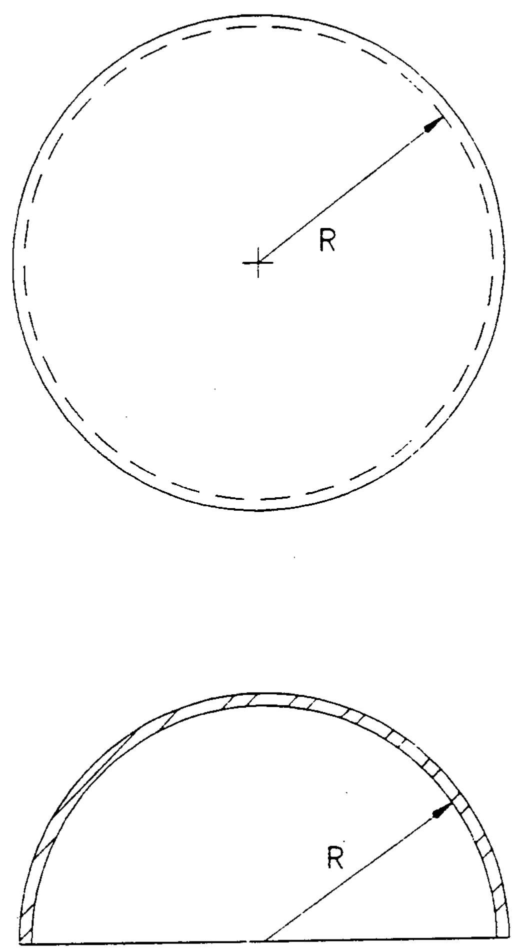

Figure 2. Typical hemispherical head used on cargo tanks designed for higher pressure.

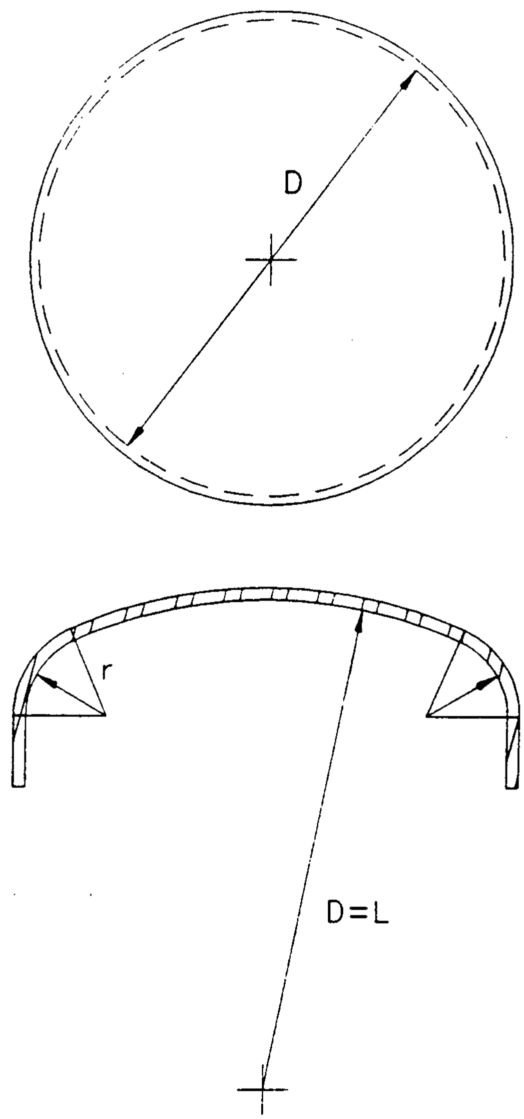

Figure 3. Typical ASME flanged and dished (torispherical) head used on cargo tanks.

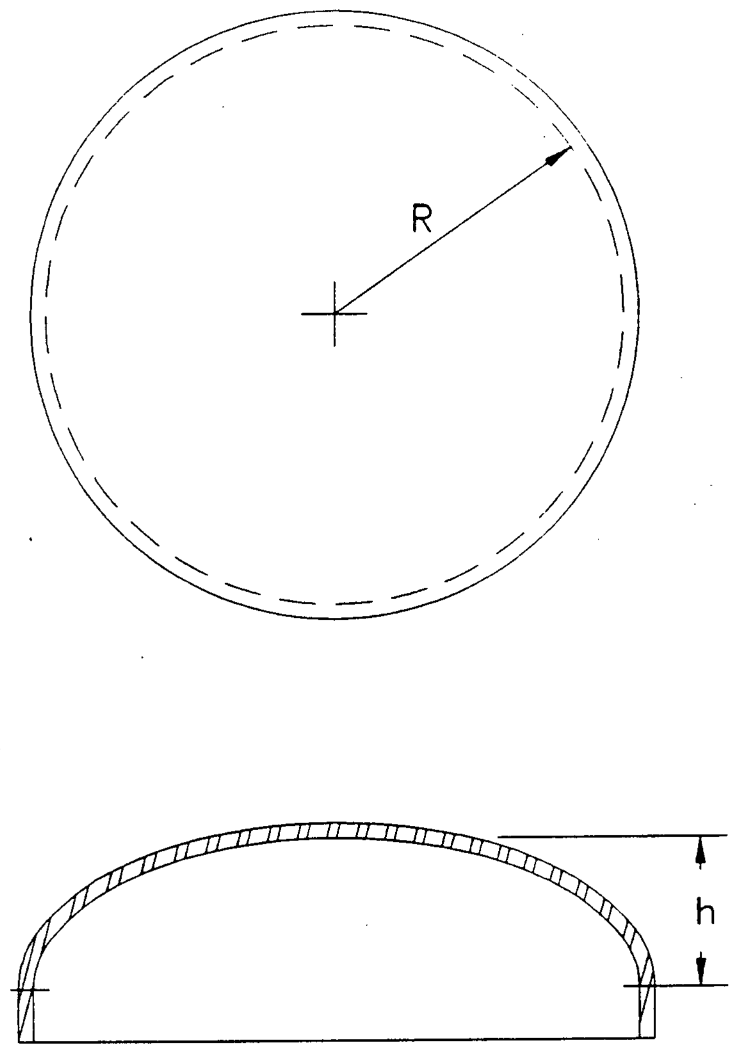

Figure 4. Typical ellipsoidal head used on cylindrical cargo tanks. Ellipsoidal heads may also be used on tanks having elliptical cross section.

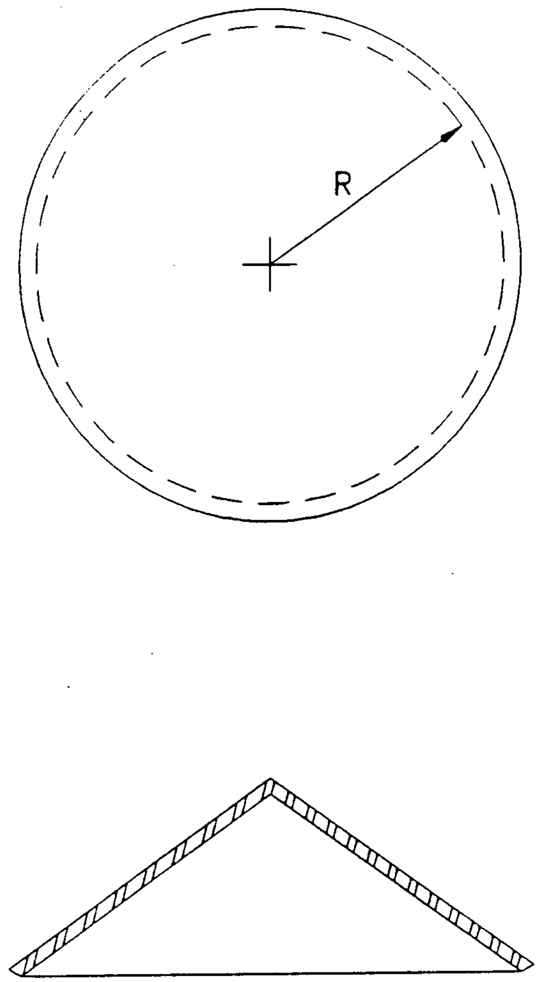

Figure 5. Conical head used on some cargo tanks.

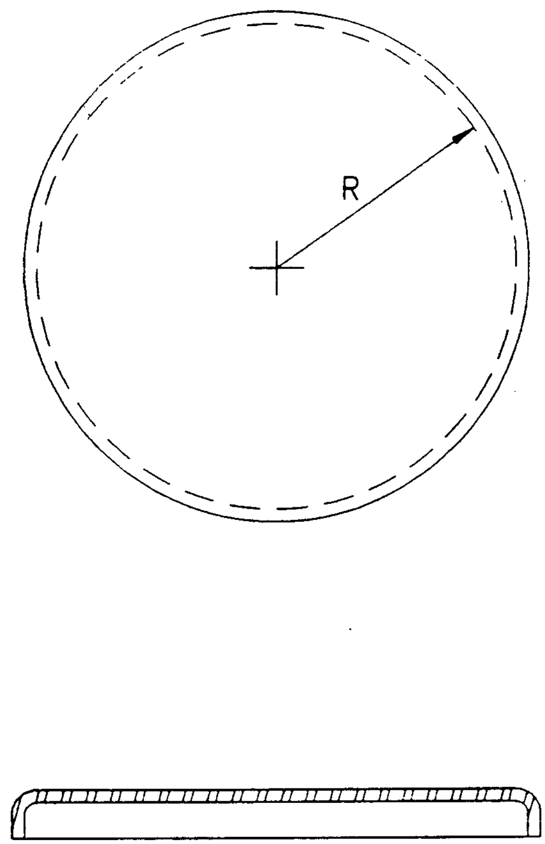

Figure 6. Typical flanged flat head.

§178.345 General design and construction requirements applicable to Specification DOT 406 (§178.346), DOT 407 (§178.347), and DOT 412 (§178.348) cargo tank motor vehicles.

§178.345-1 General requirements.

(a) Specification DOT 406, DOT 407 and DOT 412 cargo tank motor vehicles must conform to the requirements of this section in addition to the requirements of the applicable specification contained in §§178.346, 178.347 or 178.348.

(b) All specification requirements are minimum requirements.

(c) *Definitions.* The following terms apply to §§178.345, 178.346, 178.347 and 178.348.

Appurtenance means any cargo tank accessory attachment that has no lading retention or containment function and provides no structural support to the cargo tank.

Baffle means a non-liquid-tight transverse partition device that deflects, checks or regulates fluid motion in a tank.

Bulkhead means a liquid-tight transverse closure at the ends of or between cargo tanks.

Charging line means a hose, tube, pipe, or similar device used to pressurize a tank with material other than the lading.

Companion flange means one of two mating flanges where the flange faces are in contact or separated only by a thin leak sealing gasket and are secured to one another by bolts or clamps.

Connecting structure means the structure joining two cargo tanks.

Constructed and certified in conformance with the ASME Code means the cargo tank is constructed and stamped in accordance with the ASME Code, and is inspected and certified by an Authorized Inspector.

Constructed in accordance with the ASME Code means the cargo tank is constructed in accordance with the ASME Code with the authorized exceptions (see §§178.346, 178.347, and 178.348) and is inspected and certified by a Registered Inspector.

External self-closing stop-valve means a self-closing stop-valve designed so that the self-stored energy source is located outside the tank and the welded flange.

Extreme dynamic loading means the maximum single-acting loading a cargo tank may experience during its expected life, excluding accident loadings.

Flange means the structural ring for guiding or attachment of a pipe or fitting with another flange (companion flange), pipe, fitting or other attachment.

Inspection pressure means the pressure used to determine leak tightness of the tank when testing with pneumatic pressure.

Internal self-closing stop-valve means a self-closing stop-valve designed so that the self-stored energy source is located inside the tank or tank sump, or within the welded flange, and the valve seat is located within the tank or within one inch of the external face of the welded flange or sump of the tank.

Lading means the hazardous material contained in a cargo tank.

Loading/unloading connection means the fitting in the loading/unloading line farthest from the loading/unloading outlet to which the loading/unloading hose or device is attached.

Loading/unloading outlet means the tank outlet used for normal loading/unloading operations.

Loading/unloading stop-valve means the stop valve farthest from the tank loading/unloading outlet to which the loading/unloading connection is attached.

Maximum allowable working pressure or *MAWP* See §178.345-1(k).

Multi-specification cargo tank motor vehicle means a cargo tank motor vehicle equipped with two or more cargo tanks fabricated to more than one cargo tank specification.

Normal Operating Loading means the loading a cargo tank may be expected to experience routinely in operation.

Nozzle means the subassembly consisting of a pipe or tubular section with or without a welded or forged flange on one end.

Outlet means any opening in the shell or head of a tank, (including the means for attaching a closure), except that the following are not outlets: A threaded opening securely closed during transportation with a threaded plug or a threaded cap, a flanged opening securely closed during transportation with a bolted or welded blank flange, a manhole, or gauging devices, thermometer wells, and safety relief devices.

Outlet stop-valve means the stop-valve at the tank loading/unloading outlet.

Pipe coupling means a fitting with internal threads on both ends.

Rear bumper means the structure designed to prevent a vehicle or object from under-riding the rear of a motor vehicle. See §393.86 of this title.

Rear-end tank protection device means the structure designed to protect a cargo tank and any lading retention piping or devices in case of a rear end collision.

Sacrificial Device means an element, such as a shear section designed to fail under load in order to prevent damage to any lading retention part or device. The device must break under strain at no more than 70 percent of the strength of the weakest piping element between the tank and the sacrificial device. Operation of the sacrificial device must leave the remaining piping and its attachment to the tank intact and capable of retaining lading.

Self-closing stop valve means a stop-valve held in the closed position by means of self-stored energy, which opens only by application of an external force and which closes when the external force is removed.

Shear section means a sacrificial device fabricated in such a manner as to abruptly reduce the wall thickness of the adjacent piping or valve material by at least 30 percent.

Shell means the circumferential portion of a tank defined by the basic design radius or radii excluding the closing heads.

Stop-valve means a valve that stops the flow of lading.

Sump means a protrusion from the bottom of a tank shell designed to facilitate complete loading and unloading of lading.

Tank means a container, consisting of a shell and heads, that forms a pressure tight vessel having openings designed to accept pressure tight fittings or closures, but excludes any appurtenances, reinforcements, fittings, or closures.

Test pressure means the pressure to which a tank is subjected to determine pressure integrity.

Toughness of material means the capability of a material to absorb the energy represented by the area under the stress strain curve (indicating the energy absorbed per unit volume of the material) up to the point of rupture.

Vacuum tank means a tank that is loaded by reducing the pressure in the tank to below atmospheric pressure.

Variable specification cargo tank means a cargo tank that is constructed in accordance with one specification, but which may be altered to meet another specification by changing relief device, closures, lading discharge devices, and other lading retention devices.

Void means the space between tank heads or bulkheads and a connecting structure.

Welded flange means a flange attached to the tank by a weld joining the tank shell to the cylindrical outer surface of the flange, or by a fillet weld joining the tank shell to a flange shaped to fit the shell contour.

(d) A manufacturer of a cargo tank must hold a current ASME certificate of authorization and must be registered with the Department in accordance with part 107, subpart F of this chapter.

(e) All construction must be certified by an Authorized Inspector or by a Registered Inspector as applicable to the cargo tank.

(f) Each cargo tank must be designed and constructed in conformance with the requirements of the applicable cargo tank specification. Each DOT 412 cargo tank with a maximum allowable working pressure greater than 15 psig, and each DOT 407 cargo tank with a maximum allowable working pressure greater than 35 psig must be "constructed and certified in conformance with the ASME Code" except as limited or modified by the applicable cargo tank specification. Other cargo tanks must be "constructed in accordance with the ASME Code", except as limited or modified by the applicable cargo tank specification.

(g) Requirements relating to parts and accessories on motor vehicles, which are contained in part 393 of the Federal Motor Carrier Safety Regulations of this title, are incorporated into these specifications.

(h) Any additional requirements prescribed in part 173 of this subchapter that pertain to the transportation of a specific lading are incorporated into these specifications.

(i) *Cargo tank motor vehicle composed of multiple cargo tanks.*

(1) A cargo tank motor vehicle composed of more than one cargo tank may be constructed with the cargo tanks made to the same specification or to different specifications. Each cargo tank must conform in all respects with the specification for which it is certified.

(2) The strength of the connecting structure joining multiple cargo tanks in a cargo tank motor vehicle must meet the structural design requirements in §178.345-3. Any void within the connecting structure must be vented to the atmosphere and have a drain located on the bottom centerline. Each drain must be accessible and must be kept open at all times. The drain in any void within the connecting structure of a carbon steel, self-supporting cargo tank may be either a single drain of at least 1.0 inch diameter, or two or more drains of at least 0.5 inch diameter, 6.0 inches apart, one of which is located on the bottom centerline.

(j) *Variable specification cargo tank.* A cargo tank that may be physically altered to conform to another cargo tank specification must have the required physical alterations to convert from one specification to another clearly indicated on the variable specification plate.

(k) *Maximum Allowable Working Pressure (MAWP).* The MAWP for each cargo tank must be greater than or equal to the largest of the following (The MAWP derived is the pressure to be used as prescribed in the ASME Code in the design of the tank):

(1) The pressure prescribed for the lading in part 173;

(2) Vapor pressure of the most volatile lading, at 115°F (expressed in psig), plus the maximum static pressure exerted by the lading at the maximum lading density, plus any pressure exerted by a gas padding (including air in the ullage space or dome), if used; or

(3) The maximum pressure in the tank during loading or unloading.

§178.345-2 Material and material thickness.

(a) All material for shell, heads, bulkheads, and baffles must conform to section II, parts A and B, of the ASME Code except as follows:

(1) The following steels are also authorized for cargo tanks "constructed in accordance with the ASME Code".

ASTM A 569
ASTM A 570
ASTM A 572
ASTM A 607
ASTM A 656
ASTM A 715

(2) Aluminum alloys suitable for fusion welding and conforming with the 0, H32 or H34 tempers of one of the following ASTM specifications may be used for cargo tanks "constructed in accordance with the ASME Code":

ASTM B-209 Alloy 5052
ASTM B-209 Alloy 5086
ASTM B-209 Alloy 5154
ASTM B-209 Alloy 5254
ASTM B-209 Alloy 5454
ASTM B-209 Alloy 5652

All heads, bulkheads and baffles must be of 0 temper (annealed) or stronger tempers. All shell materials shall be of H32 or H34 tempers except that the lower ultimate strength tempers may be used if the minimum shell thicknesses in the tables are increased in inverse proportion to the lesser ultimate strength.

(b) *Minimum thickness.* The minimum thickness for the shell and heads must be such that the maximum stress levels specified in §178.345-3 of this subpart are not exceeded. In no case may the shell or head thickness be less than that specified in the applicable specification.

(c) *Corrosion or abrasion protection.* When required by 49 CFR part 173 for a particular lading, a cargo tank or a part thereof, subject to thinning by corrosion or mechanical abrasion due to the lading, must be protected by providing the tank or part of the tank with a suitable increase in thickness of material, a lining or some other suitable method of protection.

(1) *Corrosion allowance.* Material added for corrosion allowance need not be of uniform thickness if different rates of attack can reasonably be expected for various areas of the tank.

(2) *Lining.* Lining material must consist of a nonporous, homogeneous material not less elastic than the parent metal and substantially immune to attack by the lading. The lining material must be bonded or attached by other appropriate means to the tank wall and must be imperforate when applied. Any joint or seam in the lining must be made by fusing the materials together, or by other satisfactory means.

§178.345-3 Structural integrity.

(a) *General requirements and acceptance criteria.*

(1) The maximum calculated design stress at any point in the tank wall may not exceed the maximum allowable stress value prescribed in section VIII of the ASME Code, or 25 percent of the tensile strength of the material used at design conditions.

(2) The relevant physical properties of the materials used in each cargo tank may be established either by a certified test report from the material manufacturer or by testing in conformance with a recognized national standard. In either case, the ultimate tensile strength of the material used in the design may not exceed 120 percent of the minimum ultimate tensile strength specified in either the ASME Code or the ASTM standard to which the material is manufactured.

(3) The maximum design stress at any point in the cargo tank must be calculated separately for the loading conditions described in paragraphs (b) and (c) of this section. Alternate test or analytical methods, or a combination thereof, may be used in place of the procedures described in paragraphs (b) and (c) of this section, if the methods are accurate and verifiable.

(4) Corrosion allowance material may not be included to satisfy any of

the design calculation requirements of this section.

(b) ASME Code design and construction. The static design and construction of each cargo tank must be in accordance with Section VIII, Division 1 of the ASME Code. The tank design must include calculation of stresses generated by the MAWP, the weight of lading, the weight of structures supported by the tank wall and the effect of temperature gradients resulting from lading and ambient temperatures extremes. When dissimilar materials are used, their thermal coefficients must be used in the calculation of thermal stresses.

(1) Stress concentrations in tension, bending and torsion which occur at pads, cradles, or other supports must be considered in accordance with appendix G of section VIII, Division 1 of the ASME Code.

(2) Longitudinal compressive buckling stress for ASME certified vessels must be calculated using paragraph UG-23(b), Section VIII, Division 1 of the ASME Code. For cargo tanks not required to be certified in accordance with the ASME Code, compressive buckling stress may be calculated using alternative analysis methods which are accurate and certifiable. When alternative methods are used calculations must include both the static loads described in this paragraph and the dynamic loads described in paragraph (c) of this section.

(c) Shell design. Shell stresses resulting from static or dynamic loadings, or combinations thereof, are not uniform throughout the cargo tank motor vehicle. The vertical, longitudinal and lateral normal operating loadings can occur simultaneously and must be combined. The vertical, longitudinal and lateral extreme dynamic loadings occur separately and need not be combined.

(1) Normal operating loadings. The following procedure addresses stress in the tank shell resulting from normal operating loadings. The effective stress (the maximum principal stress at any point) must be determined by the following formula:

$$S=0.5(S_y+S_x) \pm (0.25(S_y-S_x)^2+S_s^2)^{0.5}$$

Where:

(i) S=effective stress at any given point under the combination of static and normal operating loadings that can occur at the same time, in psi.

(ii) S_y=circumferential stress generated by the MAWP and external pressure, when applicable, plus static head, in psi.

(iii) S_x= the following net longitudinal stress generated by the following static and normal operating loading conditions, in psi:

(A) The longitudinal stresses resulting from the MAWP and external pressure, when applicable, plus static head, in combination with the bending stress generated by the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the cargo tank wall;

(B) The tensile or compressive stress resulting from normal operating longitudinal acceleration or deceleration. In each case, the forces applied must be 0.35 times the vertical reaction at the suspension assembly, applied at the road surface, and as transmitted to the cargo tank wall through the suspension assembly of a trailer during deceleration; or the horizontal pivot of the truck tractor or converter dolly fifth wheel, or the drawbar hinge on the fixed dolly during acceleration; or anchoring and support members of a truck during acceleration and deceleration, 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. The following loadings must be included:

(1) The axial load generated by a decelerative force,

(2) The bending moment generated by a decelerative force,

(3) The axial load generated by an accelerative force, and

(4) The bending moment generated by an accelerative force; and

(C) The tensile or compressive stress generated by the bending moment resulting from normal operating vertical accelerative force equal to 0.35 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 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 cargo tank wall.

(iv) S_s=The following shear stresses generated by the following static and normal operating loading conditions, in psi:

(A) The static shear stress resulting from the vertical reaction at the suspension assembly of the trailer, and the horizontal pivot of the upper coupler (fifth wheel) or turntable; or 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 cargo tank wall;

(B) The vertical shear stress generated by a normal operating accelerative force equal to 0.35 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 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 cargo tank wall;

(C) The lateral shear stress generated by a normal operating lateral accelerative force equal to 0.2 times the vertical reaction at each suspension assembly of a trailer, applied at the road surface, and as transmitted to the cargo tank wall through the suspension assembly of a trailer, and the horizontal pivot of the upper coupler (fifth wheel) or turntable; or 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 cargo tank wall; and

(D) The torsional shear stress generated by the same lateral forces as described in paragraph (c)(1)(iv)(C) of this section.

(2) Extreme dynamic loadings. The following procedure addresses stress in the tank shell resulting from extreme dynamic loadings. The effective stress (the maximum principal stress at any point) must be determined by the following formula:

$$S=0.5(S_y+S_x)\pm[0.25(S_y-S_x)^2+S_s^2]^{0.5}$$

Where:

(i) S=effective stress at any given point under a combination of static and extreme dynamic loadings that can occur at the same time, in psi.

(ii) S_y=circumferential stress generated by MAWP and external pressure, when applicable, plus static head, in psi.

(iii) S_x=the following net longitudinal stress generated by the following static and extreme dynamic loading conditions, in psi.

(A) The longitudinal stresses resulting from the MAWP and external pressure, when applicable, plus static head, in combination with the bending stress generated by the static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the tank wall;

(B) The tensile or compressive stress resulting from extreme longitudinal acceleration or deceleration. In each case the forces applied must be 0.7 times the vertical reaction at the suspension assembly, applied at the road surface, and as transmitted to the cargo tank wall through the suspension assembly of a trailer during deceleration; or the horizontal pivot of the truck tractor or converter dolly fifth wheel, or the drawbar hinge on the fixed dolly during acceleration; or anchoring and support members of a truck during acceleration and deceleration, 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. The following loadings must be included:

(1) The axial load generated by a decelerative force,
(2) The bending moment generated by a decelerative force,
(3) The axial load generated by an accelerative force, and
(4) The bending moment generated by an accelerative force; and

(C) The tensile or compressive stress generated by the bending moment resulting from an extreme vertical accelerative force equal to 0.7 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 cargo tank wall.

(iv) S_s=The following shear stresses generated by static and extreme dynamic loading conditions, in psi:

(A) The static shear stress resulting from the vertical reaction at the suspension assembly of a trailer, and the horizontal pivot of the upper coupler (fifth wheel) or turntable; or 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 cargo tank wall;

(B) The vertical shear stress generated by an extreme vertical accelerative force equal to 0.7 times the vertical reaction at the suspension assembly of a trailer, and the horizontal pivot of the upper coupler (fifth wheel) or turntable; or 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 cargo tank wall;

(C) The lateral shear stress generated by an extreme lateral accelerative force equal to 0.4 times the vertical reaction at the suspension assembly of a trailer, applied at the road surface, and as transmitted to the cargo tank wall through the suspension assembly of a trailer, and the horizontal pivot of the upper coupler (fifth wheel) or turntable; or 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 cargo tank wall; and

(D) The torsional shear stress generated by the same lateral forces as described in paragraph (c)(2)(iv)(C) of this section.

(d) In no case may the minimum thickness of the cargo tank shell and heads be less than that prescribed in §178.346-2, 178.347-2 or §178.348-2 as applicable.

(e) For a cargo tank mounted on a frame or built with integral structural supports, the calculation of effective stresses for the loading conditions in paragraph (c) of this section may include the structural contribution of the frame or the integral structural supports.

(f) The design, construction, and installation of an appurtenance to the cargo tank must conform to the following requirements.

(1) Structural members, the suspension subframe, accident protection and external rings must be used as sites for attachment of appurtenances and other accessories to the cargo tank, when practicable.

(2) A lightweight attachment to the cargo tank wall, such as a conduit clip, brakeline clip, skirting structure, lamp mounting bracket or placard holder, must be of a construction having lesser strength than the cargo tank wall materials and may not be more than 72 percent of the thickness of the material to which it is attached.

(3) Except as prescribed in paragraphs (f)(1) and (f)(2) of this section, the welding of any appurtenance to the cargo tank wall must be made by attachment of a mounting pad, so that there will be no adverse effect upon the lading retention integrity of the cargo tank if any force less than that prescribed in §178.345-8(b)(1) of the subchapter is applied from any direction. The thickness of the mounting pad may not be less than that of the shell or head to which it is attached, and not more than 1.5 times the shell or head thickness. However, a pad with a minimum thickness of 0.187 inch may be used when the shell or head thickness is over 0.187 inch. If weep holes or tell-tale holes are used, the pad must be drilled or punched at its lowest point before it is welded. Each pad must--

(i) Extend at least 2 inches in each direction from any point of attachment of an appurtenance;

(ii) Have rounded corners, or otherwise be shaped in a manner to minimize stress concentrations on the shell or head; and

(iii) Be attached by a continuous weld around the pad except for a small gap at the lowest point for draining.

§178.345-4 Joints

(a) All joints between tank shell, heads, baffles, baffle attaching rings, and bulkheads must be welded in conformance with the ASME Code welding procedures.

(b) Where practical all welds must be easily accessible for inspection.

§178.345-5 Manhole assemblies.

(a) Each cargo tank with capacity greater than 400 gallons must be accessible through a manhole at least 15 inches in diameter.

(b) Each manhole, fill opening and washout assembly must be structurally capable of withstanding, without leakage or permanent deformation that would affect its structural integrity, a static internal fluid pressure of at least 36 psig, or cargo tank test pressure, whichever is greater. The manhole assembly manufacturer shall verify compliance with this requirement by hydrostatically testing at least one percent (or one manhole closure, whichever is greater) of all manhole closures of each type produced each 3 months, as follows:

(1) The manhole, fill opening, or washout assembly must be tested with the venting devices blocked. Any leakage or deformation that would affect the product retention capability of the assembly shall constitute a failure.

(2) If the manhole, fill opening, or washout assembly tested fails, then five more covers from the same lot must be tested. If one of these five covers fails, then all covers in the lot from which the tested covers were selected are to be 100% tested or rejected for service.

(c) Each manhole, filler and washout cover must be fitted with a safety device that prevents the cover from opening fully when internal pressure is present.

(d) Each manhole and fill cover must be secured with fastenings that will prevent opening of the covers as a result of vibration under normal transportation conditions or shock impact due to a rollover accident on the roadway or shoulder where the fill cover is not struck by a substantial obstacle.

(e) Each manhole cover must be permanently marked by stamping or other means with:

(1) Manufacturer's name;

(2) Test pressure ____psig;

(3) A statement certifying that the manhole cover meets the requirements in §178.345-5.

(f) All fittings and devices mounted on a manhole cover, coming in contact with the lading, must withstand the same static internal fluid pressure and contain the same permanent compliance markings as that required for the manhole cover. The fitting or device manufacturer shall verify compliance using the same test procedure and frequency of testing as specified in §178.345-5(b).

§178.345-6 Supports and anchoring.

(a) A cargo tank with a frame not integral to the tank must have the tank secured by restraining devices to eliminate any motion between the tank and frame that may abrade the tank shell due to the stopping, starting, or turning of the vehicle. The design calculations of the support elements must include the stresses indicated in §178.345-3(b) and as generated by the loads described in §178.345-3(c). Such restraining devices must be readily accessible for inspection and maintenance, except that insulation and jacketing are permitted to cover the restraining devices.

(b) A cargo tank designed and constructed so that it constitutes, in whole or in part, the structural member used in lieu of a frame must be supported in such a manner that the resulting stress levels in the tank do not exceed those specified in §178.345-3(a). The design calculations of the support elements must include the stresses indicated in §178.345-3(b) and as generated by the loads described in §178.345(c).

§178.345-7 Circumferential reinforcements.

(a) A tank with a shell thickness of less than 3/8 inch must be circumferentially reinforced with bulkheads, baffles, ring stiffeners, or any combination thereof, in addition to the tank heads.

(1) Circumferential reinforcement must be located so that the thickness and tensile strength of the shell material in combination with the frame and reinforcement produces structural integrity at least equal to that prescribed in §178.345-3 and in such a manner that the maximum unreinforced portion of the shell does not exceed 60 inches. For cargo tanks designed to be loaded by vacuum, spacing of circumferential reinforcement may exceed 60 inches provided the maximum unreinforced portion of the shell conforms with the requirements of Section VIII, Division 1 of the ASME Code.

(2) Where circumferential joints are made between conical shell sections, or between conical and cylindrical shell sections, and the angle between adjacent sections is less than 160 degrees, circumferential reinforcement

must be located within one inch of the shell joint, unless otherwise reinforced with structural members capable of maintaining shell stress levels authorized in §178.345-3. When the joint is formed by the large ends of adjacent conical shell sections or by the large end of a conical shell and a cylindrical shell section, this angle is measured inside the shell; when the joint is formed by the small end of a conical shell section and a cylindrical shell section, it is measured outside the shell.

(b) Except for doubler plates and knuckle pads, no reinforcement may cover any circumferential joint.

(c) When a baffle or baffle attachment ring is used as a circumferential reinforcement member, it must produce structural integrity at least equal to that prescribed in §178.345-3 and must be circumferentially welded to the tank shell. The welded portion may not be less than 50 percent of the total circumference of the tank and the length of any unwelded space on the joint may not exceed 40 times the shell thickness unless reinforced external to the tank.

(d) When a ring stiffener is used as a circumferential reinforcement member, whether internal or external, reinforcement must be continuous around the circumference of the cargo tank shell and must be in accordance with the following:

(1) The section modulus about the neutral axis of the ring section parallel to the shell must be at least equal to that derived from the applicable formula:

I/C=0.00027WL, for MS, HSLA and SS; or
I/C=0.000467WL, for aluminum allows;
Where:
I/C= Section modulus in inches^3
W = Tank width, or diameter in inches
L = Spacing of ring stiffener in inches; i.e., the maximum longitudinal distance from the midpoint of the unsupported shell on one side of the ring stiffener to the midpoint of the unsupported shell on the opposite side of the ring stiffener.

(2) If a ring stiffener is welded to the tank shell, a portion of the shell may be considered as part of the ring section for purposes of computing the ring section modulus. This portion of the shell may be used provided at least 50 percent of the total circumference of the tank is welded and the length of any unwelded space on the joint does not exceed 40 times the shell thickness. The

maximum portion of the shell to be used in these calculations is as follows:

Number of circumferential ring stiffener-to-shell welds	W	Shell Section
1	---	20t
2	Less than 20t	20t+W
2	20t or more	40t

where:
t=Shell thickness, inches;
W=Length of unwelded joint between parallel circumferential ring stiffener-to-shell welds.

(3) When used to meet the vacuum requirements of this section, ring stiffeners must be as prescribed in the ASME Code.

(4) If configuration of internal or external ring stiffener encloses an air space, this air space must be arranged for venting and be equipped with drainage facilities which must be kept operative at all times.

(5) Hat shaped or open channel ring stiffeners which prevent visual inspection of the tank shell are prohibited on cargo tank motor vehicles constructed of carbon steel.

§178.345-8 Accident damage protection.

(a) *General.* Each cargo tank motor vehicle must be designed and constructed in accordance with the requirements of this section and the applicable individual specification to minimize the potential for the loss of lading due to an accident.

(1) Any dome, sump, or washout cover plate projecting from the cargo tank wall that retains lading in any tank orientation, must be as strong and tough as the cargo tank wall and have a thickness at least equal to that specified by the appropriate cargo tank specification. Any such projection located in the lower 1/3 of the tank circumference (or cross section perimeter for non-circular tanks) that extends more than half its diameter at the point of attachment to the

tank or more than 4 inches from the cargo tank wall, or located in the upper 2/3 of the tank circumference (or cross section perimeter for non-circular tanks) that extends more than 1/4 its diameter or more than 2 inches from the point of attachment to the tank must have accident damage protection that are:

(i) As specified in this section;

(ii) 125 percent as strong as the otherwise required accident damage protection device; or

(iii) Attached to the cargo tank in accordance with the requirements of paragraph (a)(3) of this section.

(2) Outlets, valves, closures, piping, or any devices that if damaged in an accident could result in a loss of lading from the cargo tank must be protected by accident damage protection devices as specified in this section.

(3) Accident damage protection devices attached to the wall of a cargo tank must be able to withstand or deflect away from the cargo tank the loads specified in this section. They must be designed, constructed, and installed so as to maximize the distribution of loads to the tank wall and minimize the possibility of adversely affecting the lading retention integrity of the cargo tank. Accident induced stresses resulting from the appropriate accident damage protection device requirements in combination with the stresses from the tank operating at the MAWP may not result in a tank wall stress greater than the ultimate strength of the material of construction using a safety factor of 1.3. Deformation of the protection device is acceptable provided the devices being protected are not damaged when loads specified in this section are applied.

(4) Any piping that extends beyond an accident damage protection device must be equipped with a stop-valve and a sacrificial device such as a shear section. The sacrificial device must be located in the piping system outboard of the stop-valve and within the accident damage protection device to prevent any accidental loss of lading. The device must break at no more than 70 percent of the load that would be required to cause the failure of the protected lading retention device, part or tank wall. The failure of the sacrificial device must leave the protected lading retention device and its attachment to the tank wall intact and capable of retaining product.

(5) *Minimum road clearance.* The minimum allowable road clearance of any cargo tank motor vehicle component or protection device located between any two adjacent axles on a vehicle or vehicle combination must be at least one-half inch for each foot separating such axles, and in no case less than 12 inches.

(b) *Bottom damage protection.* Each outlet, projection or piping located in the lower 1/3 of the tank circumference (or cross section perimeter for non-circular tanks) that could be damaged in an accident thereby resulting in the loss of lading must be protected by a bottom damage protection device, except as provided by paragraph (a)(1) of this section and §173.33(e) of this subchapter. Outlets, projections and piping may be grouped or clustered together and protected by a single protection device.

(1) Any bottom damage protection device must be able to withstand a force of 155,000 pounds (based on the ultimate strength of the material) from the front, side, or rear, uniformly distributed over each surface of the device, over an area not to exceed 6 square feet, and a width not to exceed 6 feet. Suspension components and structural mounting members may be used to provide all, or part, of the protection. The device must extend no less than 6 inches beyond any component that may contain lading in transit.

(2) A lading discharge opening equipped with an internal self-closing stop-valve need not conform to paragraph (b)(1) of this section provided it is protected so as to reasonably assure against the accidental loss of lading. This protection must be provided by a sacrificial device located outboard of each internal self-closing stop-valve and within 4 inches of the major radius of the tank shell or within 4 inches of a sump, but in no case more than 8 inches from the major radius of the tank shell. The device must break at no more than 70 percent of the load that would be required to cause the failure of the protected lading retention device, part, or tank wall. The failure of the sacrificial device must leave the protected lading retention device or part and its attachment to the tank wall intact and capable of retaining product.

(c) *Rollover Damage Protection.* Each closure for openings, including but not limited to manhole, filling or inspection openings, and each valve, fitting, pressure relief device, vapor recovery stop valve or other lading retaining fitting located in the upper 2/3 of a cargo tank circumference (or cross section perimeter for non-circular tanks) must be protected by being located within or between adjacent rollover damage protection devices, or by being 125 percent of the strength that would be provided by the otherwise required damage protection device.

(1) A rollover damage protection device on a cargo tank motor vehicle must be designed and installed to withstand loads equal to twice the weight of the loaded cargo tank motor vehicle applied as follows: normal to the tank shell (perpendicular to the tank surface) and tangential (perpendicular to the normal load) from any direction. The stresses shall not exceed the ultimate strength of the material of construction. These design loads may be considered to be uniformly distributed and independently applied. If more than one rollover

protection device is used, each device must be capable of carrying its proportionate share of the required loads and in each case at least one-fourth the total tangential load. The design must be proven capable of carrying the required loads by calculations, tests, or a combination of tests and calculations.
(2) A rollover damage protection device that would otherwise allow the accumulation of liquid on the top of the tank, must be provided with a drain that directs the liquid to a safe point of discharge away from any structural component of the cargo tank motor vehicle.

(d) *Rear-end protection.* Each cargo tank motor vehicle must be provided with a rear-end protection device to protect the tank and piping in the event of a rear-end collision and reduce the likelihood of damage which could result in the loss of lading. The rear-end tank protection device must conform to the following requirements. (Nothing in this paragraph shall be construed to relieve a manufacturer of responsibility for complying with the requirements of §393.86 of this title):

(1) The rear-end tank protection device must be designed so that it can deflect at least 6 inches horizontally forward with no contact between any part of the cargo tank motor vehicle which contains lading during transit and with any part of the rear-end protection device, or with a vertical plane passing through the outboard surface of the protection device.

(2) The dimensions of the rear-end tank protection device shall conform to the following:

(i) The bottom surface of the rear-end protection device must be at least 4 inches below the lower surface of any part at the rear of the cargo tank motor vehicle which contains lading during transit and not more than 60 inches from the ground when the vehicle is empty.

(ii) The maximum width of a notch, indentation, or separation between sections of a rear-end tank protection device may not exceed 24 inches. A notched, indented, or separated rear-end protection device may be used only when the piping at the rear of the tank is equipped with a sacrificial device outboard of a shut-off valve.

(iii) The widest part of the motor vehicle at the rear may not extend more than 18 inches beyond the outermost ends of the device or (if separated) devices on either side of the vehicle.

(3) The structure of the rear-end protection device and its attachment to the vehicle must be designed to satisfy the conditions specified in paragraph (d)(1) of this section when subjected to an impact of the cargo tank motor vehicle at rated payload, at a deceleration of 2 "g". Such impact must be

considered as being uniformly applied in the horizontal plane at an angle of 10 degrees or less to the longitudinal axis of the vehicle.

(e) *Longitudinal deceleration protection.* In order to account for stresses due to longitudinal impact in an accident, the tank shell and heads must be able to withstand the load resulting from the design pressure in combination with the dynamic pressure resulting from a longitudinal deceleration of 2 "g". For the loading condition, the allowable stress value used may not exceed the ultimate strength of the material of construction using a safety factor of 1.3. Performance testing, analytical methods, or a combination thereof, may be used to prove this capability provided the methods are accurate and verifiable. For cargo tanks with internal baffles, the decelerative force may be reduced by 0.25 "g" for each baffle assembly, but in no case may the total reduction in decelerative force exceed 1.0 "g".

§178.346 Specification DOT 406; cargo tank motor vehicle.

§178.346-1 General requirements.

(a) Each Specification DOT 406 cargo tank motor vehicle must meet the general design and construction requirements in §178.345, in addition to the specific requirements contained in this section.

(b) *Maximum Allowable Working Pressure:* The MAWP of each cargo tank must be no lower than 2.65 psig and no higher than 4 psig.

(c) Vacuum loaded cargo tanks must not be constructed to this specification.

(d) Each cargo tank must be "constructed in accordance with the ASME Code" except as modified herein:

(1) The record-keeping requirements contained in the ASME Code Section VIII, Division I do not apply. Parts UG 90 thru 94 of Section VIII, Division I do not apply. Inspection and certifi-cation must be made by an inspector registered in accordance with subpart F of part 107.

(2) Loadings must be as prescribed in §178.346-3.

(3) The knuckle radius of flanged heads must be at least three times the material thickness, and in no case less than 0.5 inch. Stuffed (inserted) heads may be attached to the shell by a fillet weld. The knuckle radius and dish radius versus diameter limitations of UG-32 do not apply. Shell sections of cargo tanks designed with a non-circular cross section need not be given a preliminary curvature, as prescribed in UG-79(b).

(4) Marking, certification, data reports, and nameplates must be prescribed in §§178.345-14, 178.346-14, 178.345-15, and 178.346-15.

(5) Manhole closure assemblies must conform to §§178.345-5 and 178.346-5.

(6) Pressure relief devices must be as prescribed in §§178.345-10 and 178.346-10.

(7) The hydrostatic or pneumatic test must be as prescribed in §§178.345-13 and 178-346-13.

(8) The following paragraphs in parts UG and UW of the ASME Code, Section VIII, Division I do not apply: UG-11, UG-12, UG-22(g), UG-32(e), UG-34, UG-35, UG-44, UG-76, UG-77, UG-80, UG-81, UG-96, UG-97, UW-13(b)(2), UW-13.1(f) and the dimensional requirements found in Figure UW-13.1.

(9) Single full fillet lap joints without plug welds may be used for arc

or gas welded longitudinal seams without radiographic examination under the following conditions:

(i) For a truck-mounted cargo tank, no more than two such joints may be used on the top half of the tank and no more than two joints may be used on the bottom half. They may not be located farther from the top and bottom centerline than 16 percent of the shell's circumference.

(ii) For self-supporting cargo tank, no more than two such joints may be used on the top of the tank. They may not be located farther from the top centerline than 12.5 percent of the shell's circumference.

(iii) Compliance test. Two test specimens of the material to be used in the manufacture of a cargo tank must be tested to failure in tension. The test specimens must be of the same thicknesses and joint configuration as the cargo tank, and joined by the same welding procedures. The test specimen may represent all the tanks that are made of the same materials and welding procedures, have the same joint configuration, and are made in the same facility within 6 months after the tests are completed. Before welding, the fit-up of the joints must represent production conditions the would result in the least joint strength. Evidence of joint fit-up and test results must be retained at the manufacturers' facility.

(iv) Weld joint efficiency. The lower value of stress at failure attained in the two tensile test specimens shall be used to compute the efficiency of the joint. Determine the failure ratio by dividing the stress at failure by the mechanical properties of the adjacent metal; this value, when multiplied by 0.75, is the design weld joint efficiency.

(10) The requirements of paragraph UW-9(d), of Section VIII, Division 1, ASME Code do not apply.

§178.346-2 Material and thickness of material.

The type and thickness of material for DOT 406 cargo tank motor vehicles must conform to §178.345-2 of this part, but may in no case be less than that indicated in Tables I and II below.

Table I. Minimum Thickness of Heads (or Bulkheads and Baffles When Used as Tank Reinforcement) Using Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) or Aluminum (AL)--Expressed in Decimals of an Inch After Forming.

Material	Volume capacity in gallons per inch of length								
	14 or less			Over 14 to 23			Over 23		
	MS	HSLA SS	AL	MS	HSLA SS	AL	MS	HSLA SS	AL
Thickness	.10	.100	.16	.115	.115	.173	.129	.129	.187

Table II. Minimum Thickness of Shell Using Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) or Aluminum (AL)--Expressed in Decimals of an Inch After Forming[1]

Cargo tank motor vehicle rated capacity (gallons)	MS	SS/HSLA	AL
More than 0 to at least 4,500	0.100	0.100	0.151
More than 4,500 to at least 8,000	0.115	0.100	0.160
More than 8,000 to at least 14,000	0.129	0.129	0.173
More than 14,000	0.143	0.143	0.187

[1]Maximum distance between bulkheads, baffles, or ring stiffeners shall not exceed 60 inches.

§178.346-3 Structural Integrity.

The structural integrity of each cargo tank motor vehicle must conform to §178.345-3.

§178.346-4 Joints.

All joints in the fabrication of each cargo tank must conform to §178.345-4.

§178.346-5 Manhole assemblies.

Each manhole assembly must conform to §178.345-5.

§178.346-6 Supports and anchoring.

Supports and anchoring on each cargo tank motor vehicle must conform to §178.345-6.

§178.346-7 Circumferential reinforcement.

The circumferential reinforcement on each cargo tank must conform to §178.345-7.

§178.346-8 Accident damage protection.

Each cargo tank motor vehicle must be protected from accident damage in accordance with §178.345-8.

§178.347 Specification DOT 407; cargo tank motor vehicle.

§178.347-1 General requirements.

(a) Each specification DOT 407 cargo tank motor vehicle must conform to the general design and construction requirements in §178.345 in addition to the specific requirements contained in this section.

(b) Each tank must be of a circular cross-section and have an MAWP of at least 25 psig.

(c) Any cargo tank built to this specification with a MAWP greater than 35 psig and each tank designed to be loaded by vacuum must be "constructed and certified in accordance with the ASME Code". The external design pressure for a cargo tank loaded by vacuum must be at least 15 psi.

(d) Each cargo tank built to this specification with MAWP of 35 psig or less must be "constructed in accordance with the ASME Code" except as modified herein:

(1) The record-keeping requirements contained in the ASME Code, Section VIII, Division I, do not apply. The inspection requirements of parts UG-90 thru 94 do not apply. Inspection and certification must be made by an inspector registered in accordance with subpart F of part 107.

(2) Loadings must be as prescribed in §178.345-3.

(3) The knuckle radius of flanged heads must be at least three times the material thickness, and in no case less than 0.5 inch. Stuffed (inserted) heads may be attached to the shell by a fillet weld. The knuckle radius and dish radius versus diameter limitations of U-32 do not apply for cargo tank motor vehicles with a MAWP of 35 psig or less.

(4) Marking, certification, data reports and nameplates must be as prescribed in §§178.345-14, 178.347-14, 178.345-15, and 178.347-15.

(5) Manhole closure assemblies must conform to §§178.345-5 and 178.347-5.

(6) Pressure relief devices must be as prescribed in §§178.345-10 and 178.347-10.

(7) The hydrostatic or pneumatic test must be as prescribed in §§178.345-13 and 178.347-13.

(8) The following paragraphs in parts UG and UW of the ASME Code, Section VIII, Division I do not apply: UG-11, UG-12, UG-22(g), UG-32(e), UG-34, UG-35, UG-44, UG-76, UG-77, UG-80, UG-81, UG-96, UG-97, UW-13(b)(2), UW-13.1(f), and the dimensional requirements found in Figure UW-

13.1.

§178.347-2 Material and thickness of material.

(a) The type and thickness of material for DOT 407 specification cargo tanks must conform to §178.345-2 and this section. In no case may the thickness be less than that indicated in Tables I and II below.

Table I. Minimum Thickness of Heads (or Bulkheads and Baffles When Used as Tank Reinforcement) Using Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) or Aluminum (AL)--Expressed in decimals of an Inch After Forming

Volume capacity in gallons per inch.	10 or Less	Over 10 to 14	Over 14 to 18	Over 18 to 22	Over 22 to 26	Over 26 to 30	Over 30
Thickness (MS)	0.100	0.100	0.115	0.129	0.129	0.143	0.156
Thickness (HSLA)	0.100	0.100	0.115	0.129	0.129	0.143	0.156
Thickness (SS)	0.100	0.100	0.115	0.129	0.129	0.143	0.156
Thickness (AL)	0.160	0.160	0.173	0.187	0.194	0.216	0.237

Table II. Minimum Thickness of Shell Using Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) or Aluminum (AL)--Expressed in Decimals of an Inch After Forming

Volume capacity in gallons per inch.	10 or Less	Over 10 to 14	Over 14 to 18	Over 18 to 22	Over 22 to 26	Over 26 to 30	Over 30
Thickness (MS)	0.100	0.100	0.115	0.129	0.129	0.143	0.156
Thickness (HSLA)	0.100	0.100	0.115	0.129	0.129	0.143	0.156
Thickness (SS)	0.100	0.100	0.115	0.129	0.129	0.143	0.156
Thickness (AL)	0.151	0.151	0.160	0.173	0.194	0.216	0.237

§178.347-3 Structural integrity.

The structural integrity of each cargo tank motor vehicle must conform to

§178.345-3.

§178.347-4 Joints.

All joints in the fabrication of each cargo tank must conform to §178.345-4.

§178.347-5 Manhole assemblies.

Each manhole assembly must conform to §178.345-5, except that each manhole assembly must be capable of withstanding internal fluid pressures of 40 psig or test pressure of the tank, whichever is greater.

§178.347-6 Supports and anchoring.

Supports and anchoring on each cargo tank motor vehicle must be in conformance with §178.345-6.

§178.347-7 Circumferential reinforcement.

The circumferential reinforcement on each cargo tank must conform to §178.345-7.

§178.347-8 Accident damage protection.

Each cargo tank motor vehicle must be protected from accident damage in accordance with §178.345-8.

178.348 Specification DOT 412; cargo tank motor vehicle.

§178.348-1 General requirements.

(a) Each specification DOT 412 cargo tank motor vehicle must conform to the general design and construction requirements in §178.345 in addition to the specific requirements of this section.

(b) The MAWP of each cargo tank must be at least 5 psig.

(c) The MAWP for each cargo tank designed to be loaded by vacuum must be at least 25 psig internal and 15 psig external.

(d) Each cargo tank having a MAWP greater than 15 psig must be of circular cross-section.

(e) Each cargo tank having a--

(1) MAWP greater than 15 psig must be "constructed and certified in conformance with the ASME Code"; or

(2) MAWP of 15 psig or less must be "constructed in accordance with the ASME Code", except as modified herein:

(i) The recordkeeping requirements contained in the ASME Code, Section VIII, Division I, do not apply. Parts UG-90 thru 94 of Section VIII, Division I do not apply. Inspection and certification must be made by an inspector registered in accordance with subpart F of part 107.

(ii) Loadings must be as prescribed in §178.348-3.

(iii) The knuckle radius of flanged heads must be at least three times the material thickness, and in no case less than 0.5 inch. Stuffed (inserted) heads may be attached to the shell by a fillet weld. The knuckle radius and dish radius versus diameter limitations of UG-32 do not apply for cargo tank motor vehicles with a MAWP of 15 psig or less. Shell sections of cargo tanks designed with a non-circular cross section need not be given a preliminary curvature, as prescribed in UG-79(b).

(iv) Marking, certification, data reports, and nameplates must be as prescribed in §§178.345-14, 178.348-14, 178.345-15, and 178.348-15.

(v) Manhole closure assemblies must conform to §§178.345-5 and 178.348-5.

(vi) Pressure relief devices must be as prescribed in §§178.345-10 and 178.348-10.

(vii) The hydrostatic or pneumatic test must be as prescribed in §§178.345-13.

(viii) The following paragraphs in parts UG and UW of the ASME Code, Section VIII, Division I do not apply: UG-11, UG-12, UG-22(g), UG-32(e), UG-34, UG-35, UG-44, UG-76, UG-77, UG-80, UG-81, UG-96, UG-97, UW-13(b)(2), UW-13.1(f), and the dimensional requirements found in Figure UW-13.1.

§178.348-2 Material and thickness of material.

(a) The type and thickness of material for DOT 412 cargo tanks must conform to §178.345-2 of this part, but in no case may the thickness be less than that indicated in Tables I and II.

Table I.--Minimum Thickness of Heads (or Bulkheads and Baffles When Used as Tank Reinforcement) Using Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) or Aluminum (AL)--Expressed in Decimals of an Inch After Forming.

Volume capacity (gallons per inch)	10 or Less				Over 10 to 14				Over 14 to 18			18 and over		
Lading density at 60°F in pounds per gallon	10 lbs and less	Over 10 to 13 lbs	Over 13 to 16 lbs	Over 16 lbs	10 lbs and less	Over 10 to 13 lbs	Over 13 to 16 lbs	Over 16 lbs	10 lbs and less	Over 10 to 13 lbs	Over 13 to 16 lbs	10 lbs and less	Over 10 to 13 lbs	Over 13 to 16 lbs
Thickness (inch), steel	.100	.129	.157	.187	.129	.157	.187	.250	.157	.250	.250	.157	.250	.312
Thickness (inch), aluminum	.144	.187	.227	.270	.187	.227	.270	.360	.227	.360	.360	.227	.360	.450

Table II.--Minimum Thickness of Shell Using Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) or Aluminum (AL)--Expressed in Decimals of an Inch After Forming.

Volume capacity (gallons per inch)	10 or Less				Over 10 to 14				Over 14 to 18			18 and over		
Lading density at $60^{o}F$ in pounds per gallon	10 lbs and less	Over 10 to 13 lbs	Over 13 to 16 lbs	Over 16 lbs	10 lbs and less	Over 10 to 13 lbs	Over 13 to 16 lbs	Over 16 lbs	10 lbs and less	Over 10 to 13 lbs	Over 13 to 16 lbs	10 lbs and less	Over 10 to 13 lbs	Over 13 to 16 lbs
Thickness (steel) Distances between heads(and bulkheads baffles and ring stiffeners when used as tank reinforcement):														
36 in. or less-----------	.100	.129	.157	.187	.100	.129	.157	.187	.100	.129	.157	.129	.157	.187
Over 36 in. to 54 inches--	.100	.129	.157	.187	.100	.129	.157	.187	.129	.157	.187	.157	.250	.250
Over 54 in. to 60 inches--	.100	.129	.157	.187	.129	.157	.187	.250	.157	.250	.250	.187	.250	.312
Thickness (aluminum) Distances between heads (and bulkheads baffles and ring stiffeners when used as tank reinforcement):														
36 in. or less----------	.144	.187	.227	.270	.144	.187	.227	.270	.144	.187	.227	.187	.227	.270
Over 36 in. to 54 inches-	.144	.187	.227	.270	.144	.187	.227	.270	.187	.227	.270	.157	.360	.360
Over 54 in. to 60 inches-	.144	.187	.227	.270	.187	.227	.270	.360	.227	.360	.360	.270	.360	.450

§178.348-3 Structural Integrity.

The structural integrity of each cargo tank motor vehicle must conform to §178.345-3.

§178.348-4 Joints.

All joints in the fabrication of each cargo tank must conform to §178.345-4.

§178.348-5 Manhole assemblies.

Each manhole assembly must conform to §178.345-5.

§178.348-6 Supports and anchoring.

Supports and anchoring on each cargo tank motor vehicle must be in conformance with §178.345-6.

§178.348-7 Circumferential reinforcement.

The circumferential reinforcement on each cargo tank must conform to §178.345-7.

§178.348-8 Accident Damage Protection.

Each cargo tank motor vehicle must be protected from accident damage in accordance with §178.345-8.

§393.86 Rear end protection.

Every motor vehicle, except truck tractor, pole trailers, and vehicles engaged in driveaway-towaway operations, the date of manufacture of which is subsequent to December 31, 1952, which is so constructed that the body or the chassis assembly if without a body has a clearance at the rear end of more than 30 inches from the ground when empty, shall be provided with bumpers or devices serving similar purposes which shall be so constructed and located that:

(a) The clearance between the effective bottom of the bumpers or devices and the ground shall not exceed 30 inches with the vehicle empty;

(b) the maximum distance between the closest points between bumpers, or devices, if more than one is used, shall not exceed 24 inches;

(c) the maximum transverse distance from the widest part of the motor vehicle at the rear to the bumper or device shall not exceed 18 inches;

(d) the bumpers or devices shall be located not more than 24 inches forward of the extreme rear of the vehicle; and

(e) the bumpers or devices shall be substantially constructed and firmly attached.

Motor vehicles constructed and maintained so that the body, chassis, or other parts of the vehicle afford the rear end protection contemplated shall be deemed to be in compliance with this section.

CHAPTER 3. INFORMATION COLLECTION

Information to be requested from a manufacturer should include the following:

1. Production drawings of the tank including all relevant dimensions, date of manufacture and regulation(s) to which tank is designed.
2. Material specifications for the tank material.
3. Design calculations to verify that the tank design meets applicable requirements.
4. Tank capacity in gallons.
5. Maximum product density.
6. Maximum design weight of lading.
7. Gross vehicle weight rating (GVWR).
8. Weight of undercarriage for tank trailer.
9. Weight of tank and appurtenances.
10. Height from ground to center of tank.
11. Production drawings of overturn protection devices, including all relevant dimensions and methods of attachment to the tank.
12. Material specifications for all materials used to manufacture overturn protection devices and attach them to the tank.
13. Design calculations or test results to verify that the overturn protection devices satisfy the criteria quoted above, if they are available. Such documentation is required for Series 400 tanks but is optional for Series 300 tanks.
14. Production drawings of the rear-end protection device, including all relevant dimensions and methods of attachment to the cargo tank motor vehicle.
15. Material specifications for all materials used to manufacture the rear-end protection device and attach it to the vehicle.
16. Design calculations or test results to verify that the rear-end protection device satisfies the requirements if they are available. Documentation is not required for rear-end protection devices.
17. Specifications for and size of bolts connecting rear-end protection structure to trailer.
18. Production drawings of bottom damage protection devices, including all relevant dimensions and methods of attachment to the tank.
19. Material specifications for all materials used to manufacture bottom damage protection devices and attach them to the tank.

20. Design calculations or test results to verify that the bottom damage protection devices satisfy the criteria quoted above, if they are available. Such documentation is required for Series 400 tanks but is optional for Series 300 tanks.
21. Information from name plate.

This information should be examined carefully in the presence of the manufacturer's Design Certifying Engineer, so that initial questions and clarifications can be resolved immediately.

CHAPTER 4. METHODS OF ANALYSIS

Methods of analysis for computation of stresses in or allowable loads for cargo tanks are presented in this chapter. Many types of loads are imposed on cargo tanks. Some loads are static while others are dynamic and quite complex. Loadings described in the DOT regulations are intended to be used in static analysis procedures. Methods of analysis for the following types of loads are presented:

- Internal/external pressure
- Gravity of tank, appurtenances and lading
- Vertical acceleration
- Longitudinal acceleration
- Longitudinal deceleration
- Lateral acceleration
- Temperature gradients

Allowable stresses for materials commonly used in cargo tanks, including compressive and shear critical buckling stresses, are also addressed in this chapter.

DOT regulations require that static design of all DOT 406/407/412 cargo tanks be in accordance with the ASME Code. Static design involves all loads imposed on the cargo tank while it is at rest but does not include dynamic road loads. Any DOT 407 tank with a Maximum Allowable Working Pressure (MAWP) greater than 35 psig or designed to be loaded by vacuum must be <u>constructed and certified</u> in accordance with the ASME Code. Any DOT 412 tank with a MAWP greater than 15 psig must be <u>constructed and certified</u> in conformance with ASME Code.

Both ASME and non-ASME analysis/design methods are addressed in this chapter.

Stresses due to internal/external pressure:

<u>Circular cross sections:</u>

For circumferential stress in cylindrical vessels (vessels having circular cross sections), when the thickness does not exceed one-half of the inside radius, or P does not exceed 0.385Se, the following formula from ASME Code part UG-27 may be used to compute the required wall thickness:

$$t = PR/(Se-0.6P)$$

where:

- t = required thickness of wall of vessel, in.
- P = internal pressure in vessel, psi
- R = inside radius of vessel, in.
- S = maximum allowable tensile stress in wall of vessel, psi
- e = joint efficiency of welded longitudinal joint if one is present. It is the ratio of the tensile strength of a joint to the tensile strength of the adjacent vessel wall.

This formula can be arranged to compute the allowable internal pressure based on circumferential stress as follows:

$$P = Set/(R+0.6t)$$

where:

- P = allowable internal pressure in vessel, psi
- S = maximum allowable tensile stress in wall of vessel, psi
- e = joint efficiency of welded longitudinal joint
- R = inside radius of vessel, in.
- t = actual thickness of wall of vessel, in.

The formula can be arranged to solve for the actual circumferential tensile stress in the wall of the vessel as follows:

$$S = (PR/t) + 0.6P$$

where:

- S = actual tensile stress in wall of vessel, psi
- P = internal pressure in vessel, psi
- R = inside radius of vessel, in.
- t = actual thickness of wall of vessel, in.

The joint efficiency, e, is omitted from the formula above because it should be associated with strength rather than with computed stress. The value of stress computed from the above equation can be combined with stresses computed for other loads to compute the principal tensile stress in a vessel wall.

For longitudinal stress in cylindrical vessels when the thickness does not exceed one-half of the inside radius, or P does not exceed 1.25Se the following formulas may be used:

The required thickness of wall based on longitudinal stress can be computed using the following; however, thickness based on internal pressure would be controlled by the circumferential stress:

$$t = PR/(2Se+0.4P)$$

where:

t = required thickness of wall of vessel
P = internal pressure in vessel, psi
R = inside radius of vessel, in.
S = maximum allowable tensile stress in wall of vessel, psi
e = joint efficiency of welded circumferential joint if one is present. It is the ratio of the tensile strength of a joint to the tensile strength of the adjacent vessel wall.

This formula can be rearranged to compute the allowable internal pressure based on longitudinal stress as follows; however, allowable pressure would be controlled by the circumferential stress:

$$P = 2Set/(R-0.4t)$$

where:

P = allowable internal pressure in vessel, psi
S = maximum allowable tensile stress in wall of vessel, psi
e = joint efficiency of welded longitudinal joint
R = inside radius of vessel, in.
t = actual thickness of wall of vessel, in.

This formula can be arranged to compute the actual longitudinal tensile stress in the wall of the vessel as follows:

$$S = (PR/2t) - 0.2P$$

where:

t = thickness of wall of vessel, in.
P = internal pressure in vessel, psi
R = inside radius of vessel, in.

S = computed circumferential tensile stress in wall of vessel, psi.

Again, the joint efficiency, e, is omitted because it should be associated with strength rather than computed stress.

The formulas presented above are from the ASME Code but can be used for non-ASME designs. The engineer may also choose to use simplier formulas for thin wall pressure vessels for non-ASME designs.

For circumferential stress in a cylindrical vessel:

$$S = PR/t$$

where:

S = computed tensile stress in wall of vessel, psi
P = internal pressure in vessel, psi
R = inside radius of vessel, in.
t = thickness of wall of vessel, in.

For longitudinal stress in a cylindrical vessel:

$$S = PR/2t$$

where:

S = computed tensile stress in wall of vessel, psi
P = internal pressure in vessel, psi
R = inside radius of vessel, in.
t = thickness of wall of vessel, in.

Non-circular cross sections:

The formulas presented above for cylindrical vessels under internal pressure are not appropriate for vessels having non-circular cross sections. Bending stresses are not induced in the walls of cylindrical vessels, but are induced in vessels of non-circular cross section. Under internal pressure, non-circular cross sections tend to deform into circular cross sections and thereby induce bending stresses in the walls.

DOT Specification 178.346-1 (406 tanks) requires a MAWP of no lower than 2.65 psig and no higher than 4 psig. Many DOT 406 tanks have elliptical or approximately elliptical cross sections. Elliptical sections are frequently approximated with circular arcs having two or more different radii as shown in figure 7.

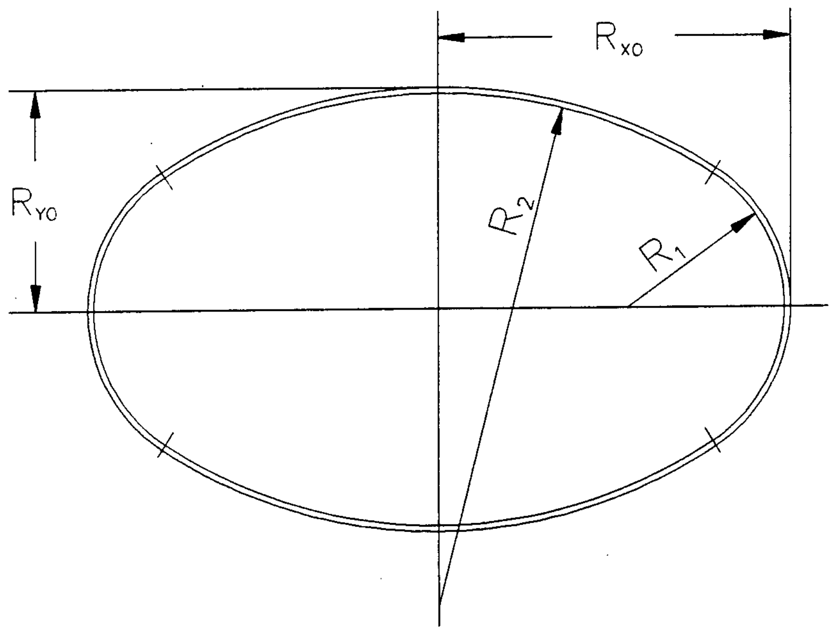

Figure 7. Approximately elliptical cross-section constructed of circular arcs.

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Heads: (pressure on concave side)

For ellipsoidal heads on circular tanks where the ratio of major to minor axis is 2:1, the following formulas may be used:

$$t = PD/(2Se-0.2P)$$

or $P = 2Set/(D+0.2t)$

or $S = (PD/2t) + 0.1P$

General formulas for other ellipsoidal heads on circular tanks are:

$$t = PDK/(2Se-0.2P)$$

or $P = 2Set/(KD+0.2t)$

or $S = (KPD/2t) + 0.1P$

where:

$K = (1/6)[2+(D/2h)^2]$

D = Inside diameter of head at its juncture with shell, inches.

h = inside depth of ellipsoidal head (from point of tangency of flange to inside surface at center), inches.

Example 3. Internal pressure - Ellipsoidal head - Pressure on concave side.

An ellipsoidal head with an inside diameter, D, of 60 inches and a thickness, t, of 0.200 inches is made of SA36 steel. The head is subjected to an internal pressure on its concave side of 25 psi. Magnitude of tensile stress in the head is to be computed.

The tensile stress in the head is:

$$S = (PD/2t) + 0.1P$$
$$S = [(25)(60)/2(.200)] + 0.1(25)$$
$$S = 3{,}753 \text{ psi}$$

For ASME torispherical heads in which the knuckle radius is 6% of the inside crown radius (i.e. L/r=16.67) and the inside crown radius equals the inside diameter of the skirt the following formulas may be used:

$$t = 0.885PL/(Se-0.1P)$$

or $\quad P = Set/(0.885L+0.1t)$

or $\quad S = (.885PL/t) + 0.1P$

General formulas for other torispherical heads are:

$$t = PLM/(2Se-0.2P)$$

or $\quad P = 2Set/(LM+0.2t)$

or $\quad S = (PLM/2t) + 0.1P$

where:

$M = (1/4)[3+(L/r)^{1/2}]$

L = Inside radius of spherical portion of torispherical head (inside crown radius), inches.

r = Knuckle inside radius, inches.

Example 4. Internal pressure - Torispherical head - Pressure on concave side.

A torispherical head with an inside diameter, D, an inside crown radius, L, of 68 inches and a thickness, t, of 0.219 inches is made of SA36 steel. The head is subjected to an internal pressure of 45 psi on its concave side. (See figure 3, page 7.) Magnitude of tensile stress in the head is to be computed.

The tensile stress in the head is:

$$S = (.885PL/t) + 0.1P$$
$$S = (0.885(45)(68)/.219) + 0.1(45)$$
$$S = 12,370 \text{ psi}$$

For hemispherical heads whose wall thickness does not exceed 0.356 R or P does not exceed 0.665Se, the following formulas may be used:

$$t = PR/(2Se-0.2P)$$

or $\quad P = 2Set/(R+0.2t)$

or $\quad S = (PR/2t) + 0.1P$

Example 5. Internal pressure - Hemispherical head - Pressure on concave side.

A hemispherical head has a radius, L, of 30 in. and a thickness, t, of 0.200 inches is made of SA36 steel. The head is subjected to an internal pressure, P, of 45 psi. (See figure 2, page 6.) The magnitude of the tensile stress is to be computed.

The tensile stress in the head is:

$$S = PR/2t + 0.1P$$
$$S = 45(30)/2(.200) + 0.1(45)$$
$$S = 3{,}380 \text{ psi}$$

For ASME Code designs, procedures given in UG-28 of the Code, which make use of charts provided therein, may be used to determine the required wall thickness of a tank shell for a given external pressure or the allowable external pressure for a given wall thickness. Procedures presented in appendix L of the ASME Code show the computation of an allowable compressive buckling stress for evaluating stresses due to external pressure combined with other loads.

Example 6. External pressure on shell (ASME design).

A DOT 407 tank is made of SA240/316L stainless steel. The outside radius of the tank, R_o, is 28.625 inches. The top portion of the shell, which is subjected to an axial compressive stress, is 0.165 inches thick and the tank is 500 inches long with stiffeners spaced at 50 inches. The tank operates at a temperature range of zero to 100°F. The critical compressive buckling strength in the top portion of the shell in the longitudinal direction is to be computed using the ASME Code.

Begin by checking to determine if the thickness satisfies the requirements in UG-28(c)(1) which is for cylinders with $D_o/t \geq 10$. Note that this procedure addresses stresses from only external pressure and does not include stresses from other structural loadings. This procedure also assumes that adequate stiffener rings are provided. Adequacy of stiffener rings is addressed later in this chapter.

Step 1 - $L/D_o = 50/57.25 = 0.873$
$D_o/t = 57.25/.165 = 347.0$

Note: L represents the length between circumferential reinforcing devices (stiffeners).

Step 2 - Go to figure G in subpart 3 of Section II, Part D, pages 674 and 675 of the 1992 Code.

Step 3 - Using the values of L/D_o or D_o/t from Step 1, the chart gives a value for Factor A of about 2.5×10^{-4}.

Step 4 - Find material chart in subpart 3 of Section II, Part D. The proper chart can be found in the "External Pressure Chart No." column of the material specification.
For SA240/316L the material specification is on pages 66 and 67 of Table 1A in Section II, Part D, and the material chart for this material, HA-4, can be found on page 681 of subpart 3.

Step 5 - Using Factor A of 2.5×10^{-4} and a design temperature of "up to 100°F" in Chart HA-4, Factor B can be determined to be about 3,500.

Step 6 - The maximum allowable external working pressure is determined from the formula in UG-28 to be:
$P_a = 4B/[3(D_o/t)] = 4(3{,}500)/[3(347)] = 13.4$ psi

Step 7 - Since Factor B was obtainable in Step 5 from the material chart, Step 7 is not applicable.

Step 8 - If P_a is greater than or equal to the external pressure on the tank the design is adequate.

Once the thickness of the tank is determined to be adequate using UG-28, UG-23(b) should be used to determine the maximum allowable compressive stress.

Step 1 - Calculate Factor A from the formula:
$A = 0.125/(R_o/t)$
For this example:
$A = 0.125/(28.625/.165)$
$A = 7.21 \times 10^{-4}$

Step 2 - Go to material chart in subpart 3 of Section II, Part D. For SA240/316L the material chart is HA-4 on page 681 of Section II, Part D.

Step 3 - Using Factor A as 7.2×10^{-4} and a design temperature of "up to 100°F", Factor B, which is the allowable compressive stress can be determined to be 8,800 psi.

Step 4 - Not necessary since Factor B was determined in Step 3.

Step 5 - Since the allowable compressive stress found from Factor B (8,800 psi) is less than the allowable tensile stress, (17,500 psi from Table 1A of Section II, Part D) the allowable longitudinal compressive stress for this tank shell is 8,800 psi.

Example 7. External pressure on heads (convex side) (ASME design). See UG-33 of the ASME Code, Section VIII, Division 1.

Given: DOT 407 tank, D_o=60", t=0.165", MAWP=35 psi. Material is SA240/316L SS. Tank is vacuum loaded and must withstand external pressure of one atmosphere (14.7 psi). Example analyses for various types of heads are illustrated below.

<u>Hemispherical heads:</u>

Step 1 - UG-33(c) refers the designer to UG-28(d)

Step 2 - Compute the Factor A

$A = 0.125/(R_o/t) = 0.125/(30/0.165) = 6.88 \times 10^{-4}$

Step 3 - Go to the material chart in subpart 3 of Section II, Part D. Determine the value of B.

Assuming the tank is designed to handle a product temperature of up to 300°F, Factor B would be approximately 6,600.

Step 4 - Calculate the maximum allowable external pressure:

$P_a = B/(R_o/t) = 6{,}600/(30/0.165) = 36.3$ psi

36.3 psi > 14.7 psi OK

<u>Ellipsoidal heads:</u>

$R_o = 0.9D_o = 0.9(60) = 54''$

Step 1 - Compute the Factor A.

$A = 0.125/(R_o/t) = 0.125/(54/0.165) = 3.82 \times 10^{-4}$

Step 2 - Go to the material chart in subpart 3 of Section II, Part D. Determine the value of B.
Assuming the tank is designed to handle a product temperature of up to 300°F, Factor B would be approximately 4,800.

Step 3 - Calculate the maximum allowable external pressure:
$P_a = B/(R_o/t) = 4{,}800/(54/0.165) = 14.67$ psi
14.7 psi = atmospheric pressure (full vacuum). This head design is borderline.

Torispherical heads:

$R_o = 72''$

Step 1 - Compute the Factor A.
$A = 0.125/(R_o/t) = 0.125(72/0.165) = 2.86X10^{-4}$

Step 2 - Go to the material chart in subpart 3 of Section II, Part D. Determine the value of B.
Assuming the tank is designed to handle a product temperature of up to 300°F, Factor B would be approximately 3,800.

Step 3 - Calculate the maximum allowable external pressure:
$P_a = B/(R_o/t) = 3{,}800(72/0.165) = 8.71$ psi.

The maximum allowable pressure calculated, 8.71 psi, is less than design pressure (14.7 psi) so this design would be inadequate.

Pressure due to static head:

Consider a 1-ft. tall column of water acting on 1 square foot of surface.

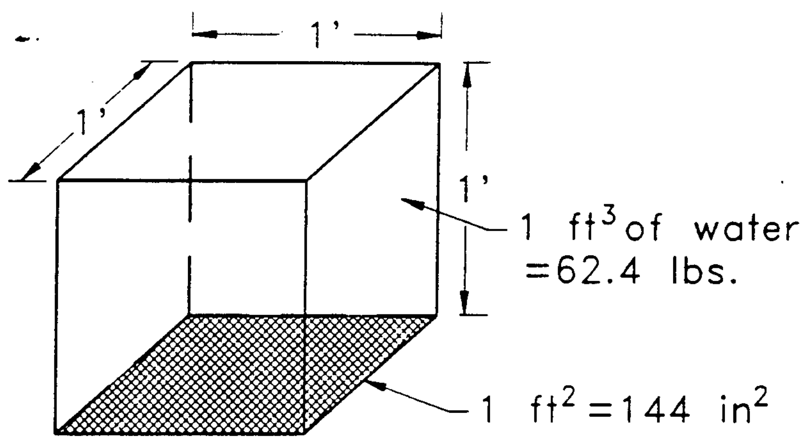


Figure 9. Illustration of parameters for static head.

The pressure on the bottom surface is:

62.4 lbs./144 in.^{2} = 0.433 psi

Water exerts 0.433 psi per foot of head or the inverse of 0.433 is 1/0.433 = 2.31. In other words, 2.31 feet of head of water is equivalent to 1 psi (2.31 ft/psi.). This is a hydrostatic pressure, i.e., it is the same value in all directions.

A similar factor for another liquid can be obtained by dividing 2.31 by the relative specific gravity of the other liquid. The relative specific gravity of water is 1.00. Relative specific gravity for another liquid can be obtained by dividing its density in lbs/gal. by 8.33. The density of water is 8.33 lbs/gal.

231 cu.in. = 1 gal
7.5 gal. = 1 cu. ft.

The steady-static pressure due to constant acceleration of a column of liquid (oriented in any direction) is equal to the static head pressure multiplied by the acceleration in g's.

Stresses due to static head pressure:

Stresses due to static head pressure can be computed in the same manner and with the same equation used for stresses from MAWP. Static head is usually known in units of inches or feet of height of the liquid. If the liquid is water with a relative specific gravity of 1.00 or 8.33 lbs/gal, the static head pressure can be computed by multiplying the feet of head by 0.433. The relative specific gravity of another liquid can be obtained by dividing the density in lbs/gal by 8.33. The static head pressure for that liquid can be obtained by multiplying the feet of head by 0.433 and by the relative specific gravity. During design and analysis, the specific gravity of the heaviest product (liquid) should be used.

Example 8. Computation of stress due to static head.

A cylindrical cargo tank with a vertical height, h, of 56.9 inches is used for transporting gasoline which has a specific gravity, γ, of 0.7. The tank has a thickness, t, of .165 inches. To determine the static head pressure at the bottom of the tank:

$$P = (h/12)(0.433)(\gamma)$$
$$P = (56.9/12)(0.433)(0.7) = 1.44 \text{ psi}$$

Circumferential stress due to this pressure can be computed by:

$$S_{y2} = (PR/t) + 0.6P$$
$$S_{y2} = 1.44(28.5)/.165 + 0.6(1.44)$$
$$S_{y2} = 250 \text{ psi at bottom of tank}$$

Stresses due to static head can also be accounted for by adding the static head pressure to the MAWP for calculations of stress (figure 9). Static head pressure is zero at the top of the tank during static and normal operating loading conditions. The following formulas can be used:

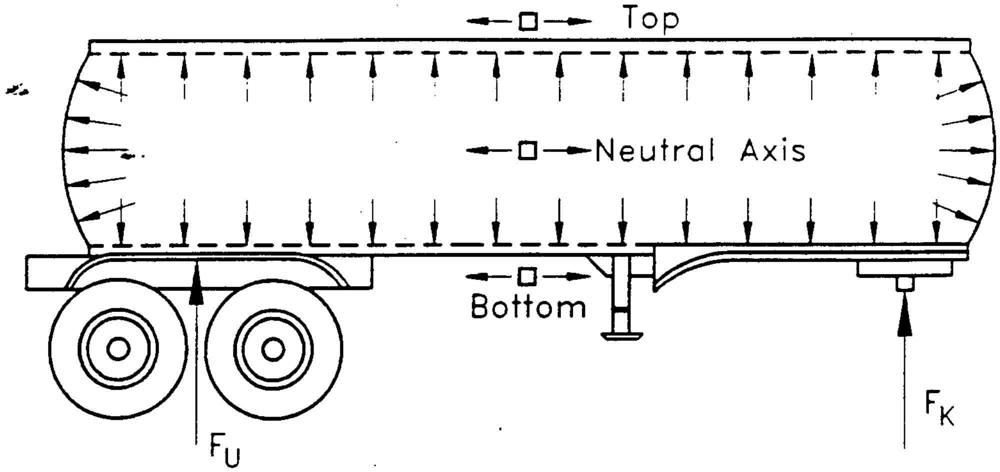


Figure 10. Illustration for computation of stresses due to combined internal pressure and static head.

At top of tank:

$$S_x = (P_m)R/2t$$
$$s_y = (P_m)R/t$$

At neutral axis (near midheight):

$$S_x = [P_m+P_h/2]R/2t$$
$$S_y = [P_m+P_h/2]R/t$$

At bottom of tank:

$$S_x = [P_m+P_h]R/2t$$
$$S_y = [P_m+P_h]R/t$$

where P_m = MAWP or zero or vacuum pressure, psi.

P_h = pressure due to static head of product, psi.

Flexural Shear in Thin-Walled Vessels:

A simplified formula for maximum flexural shear stress in a thin-wall flexural member of circular cross section can be derived from the flexural shear stress formula from basic strength of materials:

$$S_s = VA\bar{y}/Ib$$

For a solid semi-circle, the area is $\pi R^2/2$ and the centroid is located at $4R/3\pi$ from the flat side where R is the radius. Then $A\bar{y}$ is $2R^3/3$. Then $A\bar{y}$ for a thin-wall semi-circle would be $2(R_o^3-R_i^3)/3$. The moment of inertia, I, is $\pi(R_o^4-R_i^4)/4$. b in the equation for S_s is 2t which is $2(R_o-R_i)$. When these parameters are substituted into the equation for S_s, the result is:

$$S_s = [V(2)(R_o^3-R_i^3)/3] \,/\, [\pi(R_o^4-R_i^4)(2)(R_o-R_i)/4]$$

$$S_s = [V(R_o^3-R_i^3)] \,/\, [3\pi(R_o^2-R_i^2)(R_o^2+R_i^2)(R_o-R_i)/4]$$

$$S_s = [V(R_o^3-R_i^3)] \,/\, [3\pi(R_o^2-R_i^2)(R_o^3-R_iR_o^2+R_i^2R_o-R_i^3)/4]$$

$$(R_o-R_i)^3 \rightarrow \text{zero}$$

and $S_s = V/0.5A$

where:

S_s = maximum shear stress in wall of tank, psi. This maximum occurs in the wall at the neutral axis of the cross section.

V = shear force on the cross section, lbs.

A = area of the cross sections, in^2.

If R_o/t is greater than 5, the error is less than 1%.

Stresses due to static gravity weight:

For structural analysis of a trailer type cargo tank, the tank can be idealized as a simply supported beam with overhangs on each end. On single or tandem axle trailers, the reaction on the rear undercarriage can be idealized as a concentrated force, especially for computation of maximum bending moment. Three concentrated forces or a distributed force could be used if the engineer chooses to do so. If the trailer has more than two axles, the reaction should be idealized in a manner consistent with the nature of the suspension system and subframe. The manner in which the reaction is idealized would influence the magnitude of shear force adjacent to the reaction. At the kingpin, idealization of the reaction as a concentrated force is appropriate. Idealization of a typical trailer cargo tank is shown in figure 11.

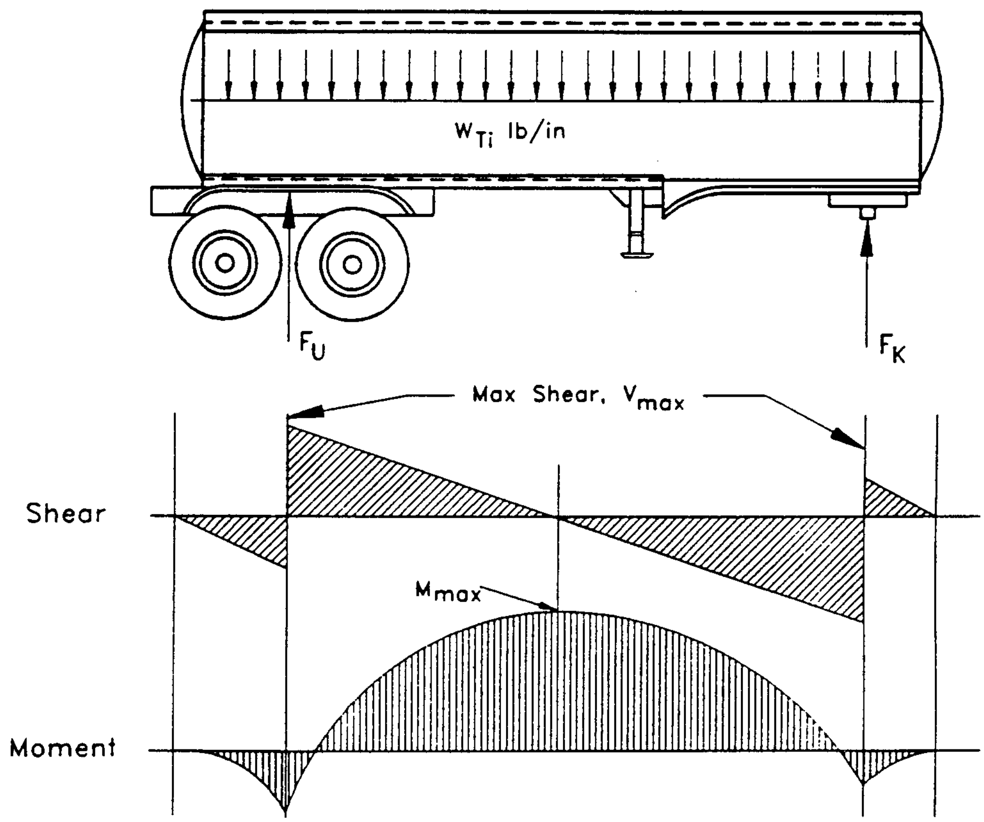


Figure 11. Diagrams for static weight of tank and lading.

Maximum shear stress, S_s, occurs in the side walls of the tank at the neutral axis at the cross section of maximum shear force and can be computed using the following equation:

$$S_{s1} = V_{max}/0.5A$$

where:

S_{s1} = maximum flexural shear stress, psi
V_{max} = maximum shear force, lbs.
A = cross sectional area of tank, in^2

Maximum longitudinal normal stress occurs at point of maximum moment and can be computed using the flexure formula:

$$S_{x3} = M_{max}/Z_e$$

where:

S_{x3} = maximum flexural stress, psi
M_{max} = maximum bending moment, in-lbs.
Z_e = elastic section modulus, in^3

The stresses computed above are due to static weight of the tank and lading. When a vehicle is in motion, it is subjected to dynamic loadings.

The loading diagram shown assumes a uniformly distributed load from the static weight of the tank and lading. Small (less than 1,000 lbs.) appurtenances, attachments and fittings on the tank can be included in the weight of the tank and assumed to be uniformly distributed. However, larger masses attached to the tank should be treated as concentrated loads and the appropriate shear and moment diagrams should be constructed.

Example 9. Computation of shear and bending stresses in elliptical tank.

The static gravity bending stresses in an DOT 406 tank trailer are to be computed. The tank has an elliptical cross section with continuous overturn rails along the top and continuous frame rails along the bottom. Details of the cross section are shown in figure 12.

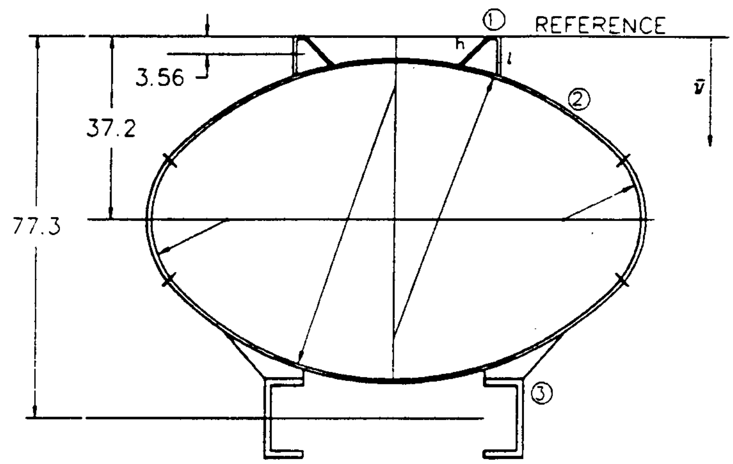


Figure 12. Cross section of elliptical DOT 406 tank.

Material = 5454 H32 (Aluminum)
S_u = 36,000 psi
S_a = .25(36,000) = 9,000 psi
Thickness = 0.173 inches
Capacity = 7,200 gallons
MAWP = 3 psi
W_s = 8,000 lbs. (shell)
W_u = 6,000 lbs. (undercarriage)
W_L = 46,000 lbs. (lading)
W_T = $W_L + W_s$ (filled tank)
W_T = 46,000 + 8,000
W_T = 54,000 lbs.
W_{Ti} = 54,000/500 = 108 lb/in.

Cross section properties of DOT 406 tank.

(1) Overturn Rail (See figure 45)
5454 H32
8"=h, 9.5"=ℓ
0.173"=t

(2) DOT 406 Elliptical Tank
5454 H32
92" X 64"
0.173"=t

(3) Frame Rails
12" by 2.96" by 7.41 lb/ft. American Standard Aluminum Channel

Areas of Segments (not including support attachments)

Overturn Rails
A_1 = [(8+9.5)](0.173)
A_1 = 3.0/rail
A_1 = 6.0 in^2 (for 2 rails)

Tank Shell
$A_2 = \pi(R_{yo}R_{xo}-R_{yi}R_{xi})$
$A_2 = \pi[(32)(46)-(31.827)(45.827)]$
A_2 = 42.3 in^2

Frame Rails
A_3 = 6.30/rail
A_3 = 12.6 in^2 (for 2 rails)

Centroid of Section

Segment	A_i	$\bar{y}_i$	$A_i\bar{y}_i$
Overturn Rails	6.0	3.56	21.4
Tank Shell	42.3	37.20	1,574
Frame Rails	12.6	77.63	978
	60.9		2,573

$\bar{x}$ = centerline of section (symmetrical)
$\bar{y} = \Sigma A_i \bar{y}_i / \Sigma A_i$ = 2,573/60.9 = 42.3 inches from reference axis

Moment of Inertia

$I_1 = bh^3/12$
$I_1 = (0.173)(8)^3/12+(0.173)(9.5)^3/12$
$I_1 = 19.7\ in^4$
$I_2 = \pi(R_{yo}{}^3R_{xo}-R_{yi}{}^3R_{xi})/4$
$I_2 = \pi[(32^3)(46)-(31.827^3)(45.827)]/4$
$I_2 = 23{,}477\ in^4$
$I_3 = 131.8\ in^4$ per rail
$I_3 = 263.6\ in^4$

$I_{TOT} = \Sigma I_o+\Sigma Ad^2$
$I_{TOT} = (I_1+Ad_1{}^2)+(I_2+Ad_2{}^2)+(I_3+Ad_3{}^2)$
$I_{TOT} = [(19.7)+(6.0)(42.3-3.56)^2]+[(23{,}477)+(42.3)(42.3-37.2)^2]$
$+[(263.6)+(12.6)(42.3-77.63)^2]$
$I_{TOT} = 9{,}024 + 24{,}577 + 15{,}991$
$I_{TOT} = 49{,}592\ in^4$

Section modulus at top surface of overturn rail:

$Z_e = 49{,}592/42.3$
$Z_e = 1{,}172\ in^3$

Section modulus at top surface of tank wall:

$Z_e = 49{,}592/(42.3-37.2+32)$
$Z_e = 49{,}592/37.1$
$Z_e = 1{,}337\ in^3$

Section modulus at bottom surface of bottom frame rails:

$Z_e = 49{,}592/(77.3+6-42.3)$
$Z_e = 49{,}592/41$
$Z_e = 1{,}210\ in^3$

Section modulus at bottom surface of tank wall:

$Z_e = 49{,}592/(37.2+32-42.3)$
$Z_e = 49{,}592/26.9$
$Z_e = 1{,}844\ in^3$

Weights, dimensions, reactions, shear diagram and bending moment diagram for the tank idealized as a simply supported beam are presented in figure 13.

The maximum bending moment occurs at 214 in. forward of the center of the rear suspension and is 2,361,000 in.-lbs. The highest longitudinal tensile stress in the tank wall occurs at the bottom surface of the tank wall and is:

$$S_{x3} = M_{max}/Z_e$$
$$S_{x3} = 2{,}361{,}000/1{,}800$$
$$S_{x3} = 1{,}312 \text{ psi (tension)}$$

The maximum compressive stress in the tank wall at the top surface is:

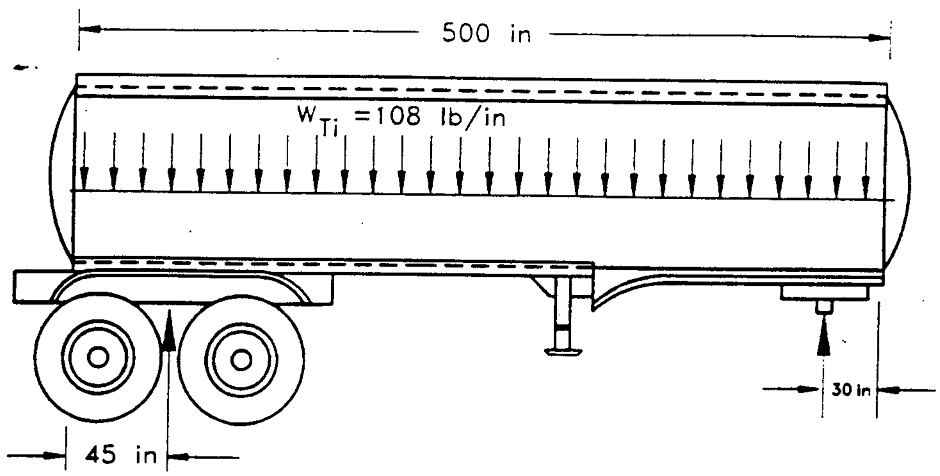
$$S_{x3} = M_{max}/Z_e$$
$$S_{x3} = 2{,}361{,}000/1{,}330$$
$$S_{x3} = -1{,}809 \text{ psi (compression)}$$

The maximum tensile stress at the bottom surface of the longitudinal frame rails is:

$$S_{x3} = M_{max}/Z_e$$
$$S_{x3} = 2{,}361{,}000/1{,}185$$
$$S_{x3} = 2{,}069 \text{ psi (tension)}$$

The maximum compressive stress at the top surface of the continuous overturn protection rails is:

$$S_{x3} = M_{max}/Z_e$$
$$S_{x3} = 2{,}361{,}000/1{,}166$$
$$S_{x3} = -2{,}025 \text{ psi (compression)}$$



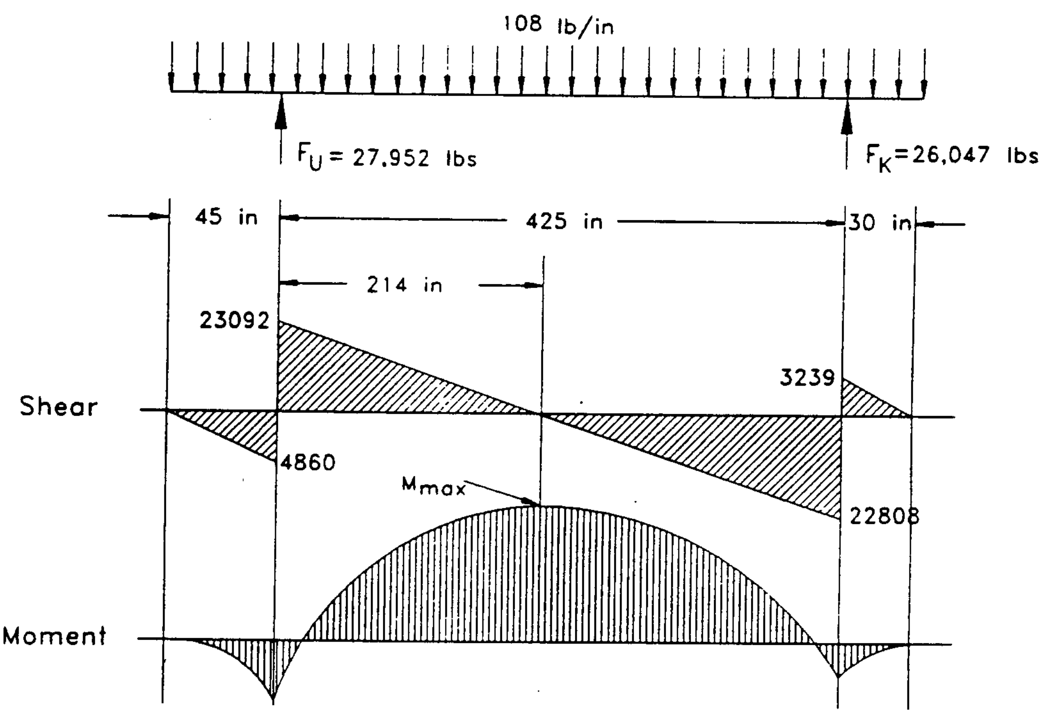


Figure 13. Shear and bending moment diagrams for example 9.

Example 10. Shear and bending stresses due to static weight.

A cylindrical tank has an inside radius, R_i, of 28.46 inches and an outside radius, R_o, of 28.625 inches. The tank is subject to a uniformly distributed weight, W_{Ti}, of 100 lbs/in. The reactions are located at the kingpin and rear tandems as shown.

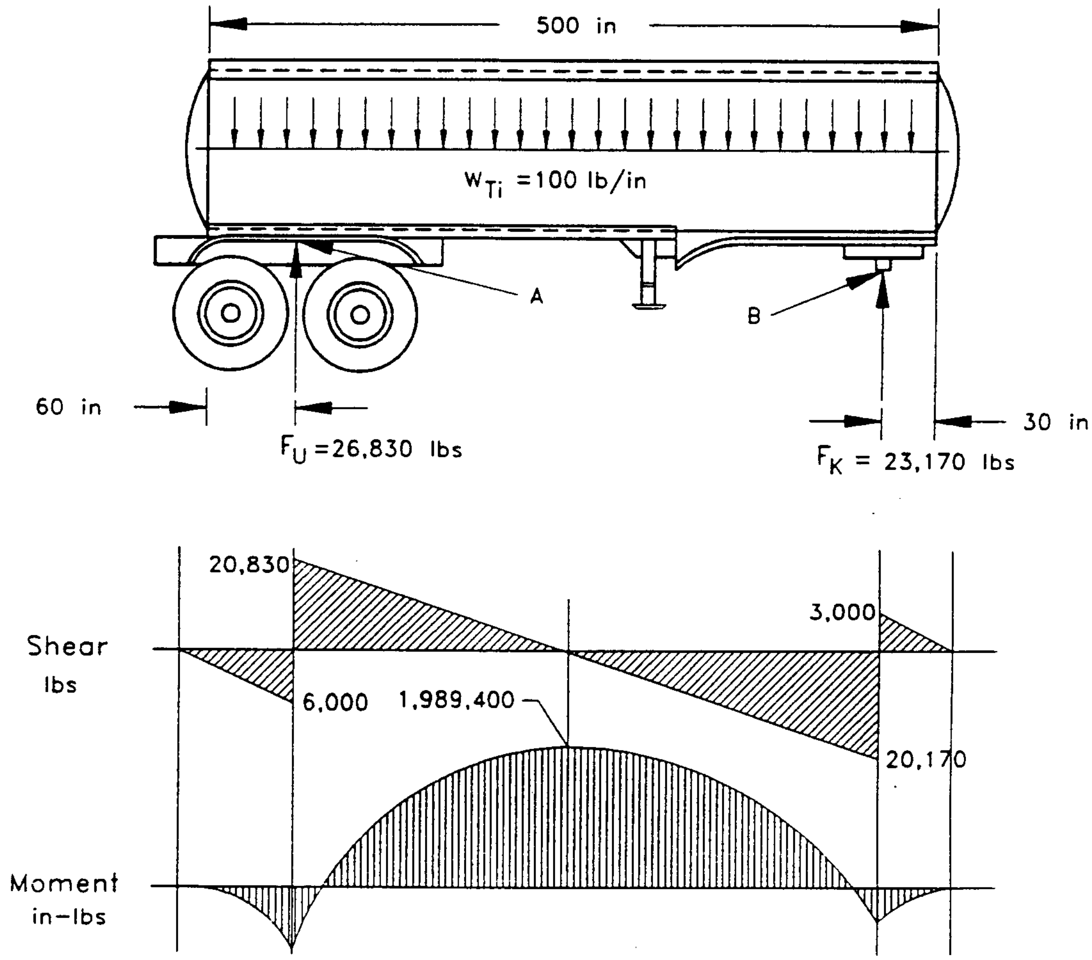


Figure 14. Shear and bending moment diagrams for example 10.

The maximum bending stress due to static weight will be at the point of maximum moment. (Unless there is a welded joint near the point of maximum moment with a weld efficiency of less than 1.0.) To find the point of maximum moment one must first compute the reactions.

$$\sum M_A = 100(500)[(500/2)-60] - F_k(410) = 0$$
$$F_k = 23{,}170 \text{ lbs.}$$
$$F_u = 50{,}000 - 23{,}170$$
$$F_u = 26{,}830 \text{ lbs.}$$

The maximum moment is at the point where the shear, $V = 0$. The distance, L_{max}, from the rear of the tank to the location of maximum moment may be computed as follows:

$$L_{max} = F_u/W_{Ti} = 26{,}830/100 = 268.3 \text{ inches from the rear of the tank or } 208.3 \text{ inches from the reaction, } F_u$$
$$M_{max} = F_u(208.3) - W_{Ti}(268.3)^2/2$$
$$M_{max} = 26{,}830(208.3) - 100(268.3)^2/2$$
$$M_{max} = 1{,}989{,}400 \text{ in-lbs.}$$

The bending stress due to static weight is then computed by:

$$S_{x3} = M_{max}/Z_e$$

where:

Z_e = elastic section modulus = I/c

$I = \pi(R_o^4-R_i^4)/4 = \pi(28.625^4-28.46^4)/4 = 12{,}053 \text{ in}^4$

$c = 28.6$ inches

$Z_e = 12{,}053/28.6 = 421.1 \text{ in}^3$

so:

$$S_{x3} = 1{,}989{,}400 \text{ in-lbs}/421.1 \text{ in}^3 = 4{,}724 \text{ psi}$$

The shear forces can be computed or taken from the shear diagram. The maximum flexural shear stress will occur at the cross section subjected to the highest shear force. It will occur in the side walls at the neutral axis which is at or near mid-depth and can be closely approximated, as stated earlier, by:

$$S_{sl} = V_{max}/(0.5A)$$
$$A = \pi(28.625^2 - 28.460^2)$$
$$A = 29.6\ in^2$$
$$S_{sl} = 20{,}830\ lbs./(0.5)(29.6)$$
$$S_{sl} = 1{,}407\ psi$$

Stresses due to weight of lading and structures (appurtenances) supported by tank wall can be considered along with bending stresses due to static weight. Lightweight appurtenances can be considered to be uniformly distributed along the length of the tank and included with the uniformly distributed self weight of the tank. Heavier appurtenances should be considered to be concentrated loads at their attachments.

Local stresses in the tank wall that result from attachments of heavier appurtenances, rollover protection devices, and other supports, should be evaluated in accordance with procedures in appendix G of the ASME Code and the Welding Research Council Bulletin 107 to the extent that they are applicable. A difficulty exists in applying procedures in (WRC) Bulletin 107 in that the procedures do not account for doubler plates (or pads) that are commonly used in mounting attachments to cargo tank walls.

Stresses due to vertical acceleration:

Shear and bending moment diagrams for vertical accelerative force can be constructed in the same manner as those for gravity weight as shown in figure 15. Weight of the tank, appurtenances and lading, as well as the reactions for gravity weight, are multiplied by a factor, F_2. The value of F_2 is the specified vertical acceleration in g's. It is 0.35 for normal operating loadings and 0.70 for extreme dynamic loadings.

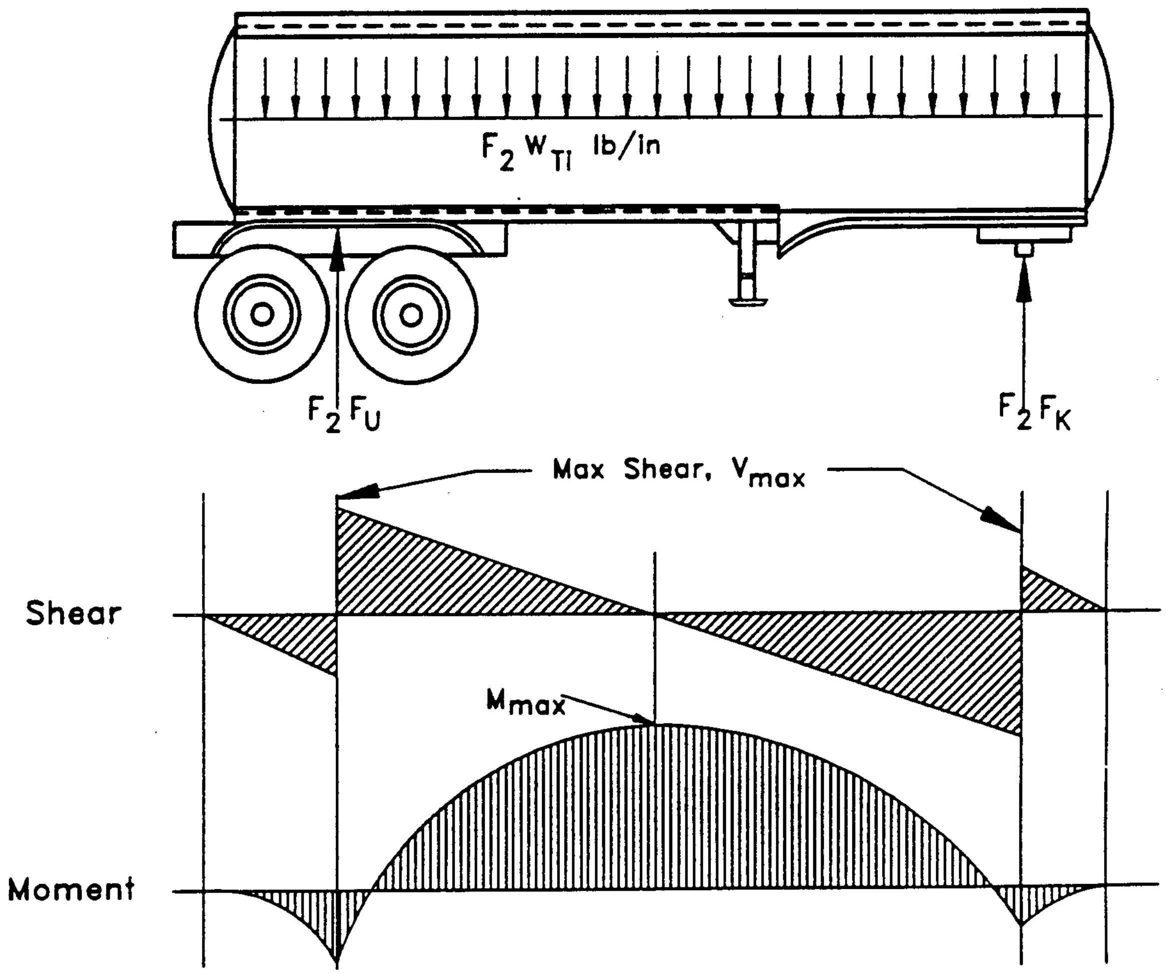


Figure 15. Idealization of cargo tank for vertical accelerative force.

Example 11. Computation of shear stress generated by normal operating vertical accelerative force.

This calculation is identical to the preceeding one in example 10 with a factor of 0.35 applied to account for normal operating loads.

$S_{s2} = V_{max}/0.5A$
$S_{s2} = 0.35(20,830)/(0.5)(29.6)$
$S_{s2} = 493$ psi

It is noted that $S_{s2} = F_2 S_{s1}$.

Example 12. Computation of bending stresses due to normal operating vertical accelerative force.

The vertical accelerative force due to normal operating loads would act as shown in figure 16.

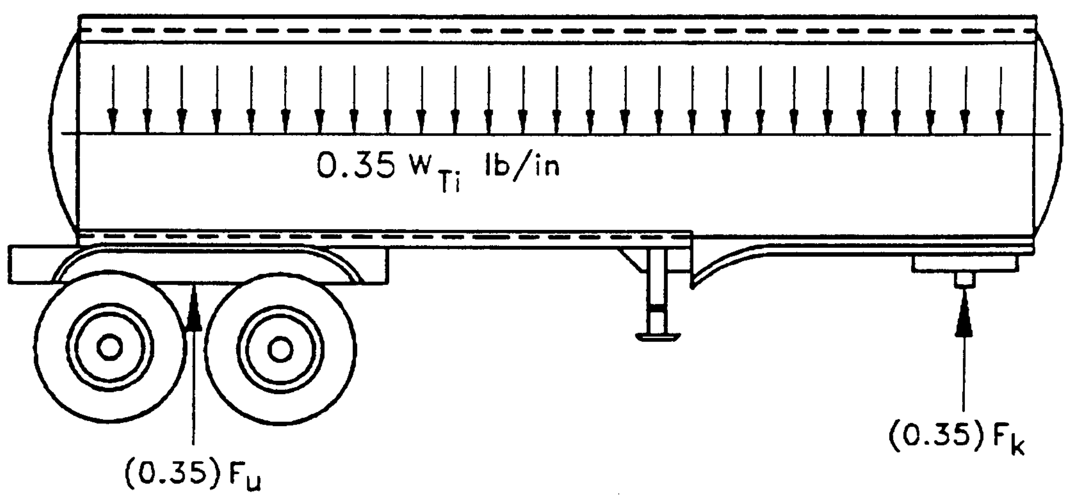


Figure 16. Idealization of cargo tank for normal operating vertical accelerative force.

$S_{x4} = M_{max}/Z_e$

where:

M_{max} = (0.35)(1,989,400) in-lbs.
Z_e = 421.1 in^3

$$S_{x4} = [0.35(1{,}989{,}400/421.1) = 1{,}653 \text{ psi}$$

It is noted that $S_{x4} = F_2 S_{x3}$.

Stresses due to longitudinal deceleration:

<u>Axial Stresses</u>

The axial load generated by a decelerative force can be considered as shown in figure 17. For normal operating longitudinal deceleration, the factor, F_1, is 0.35 and for extreme dynamic longitudinal deceleration, it is 0.70. The tensile stress can be computed using:

$$S_{x5} = F_x/A$$
$$S_{x5} = F_1(F_u+W_u)/A$$

where:

(F_u+W_u) = total weight of rear of trailer, lbs.

A = cross section area of tank, in^2.

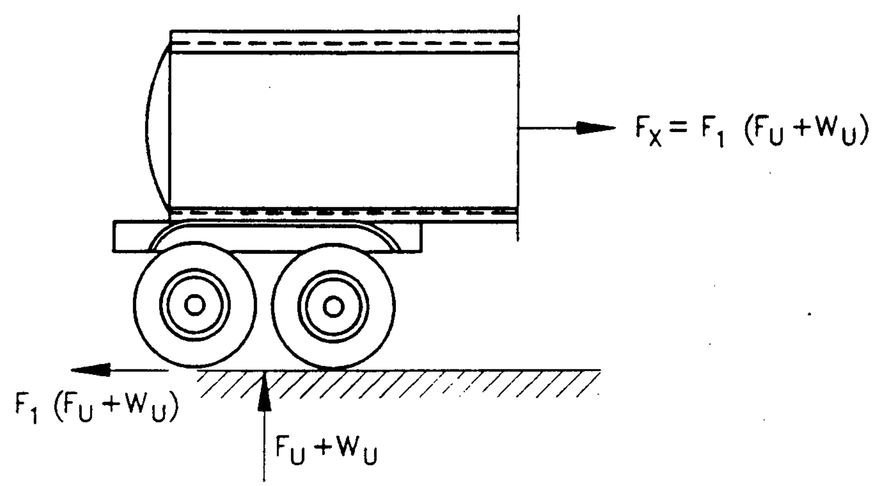


Figure 17. Idealization of cargo tank for computation of axial tensile stress due to longitudinal decelerative force from application of trailer brakes.

Example 13. Computation of axial tensile stress due to a decelerative force (trailer brakes).

Continuing from example 10, and given the following values, the axial tensile stress (only trailer brakes applied) can be computed as shown:

Weight of the undercarriage at the rear tandem (W_u) = 6,000 lbs.
Normal operating load factor (F_1) = 0.35
Reaction at rear tandem (F_u) = 26,830 lbs.
Area of tank cross section (A) = 29.6 in^2

$$S_{x5} = F_x/A = [F_1(F_u+W_u)]/A = [0.35(26,830+6,000)]/29.6$$
$$S_{x5} = 388 \text{ psi}$$

Note that it may be necessary in some cases to compute the compressive stress in the tank due to deceleration (tractor brakes applied only); however, the worst case loading scenarios will generally yield higher stresses in tension due to the above computation. If there is a circumferential weld located in close proximity to the rear tandem, it may be necessary to evaluate the stresses at a material strength lessened by a joint efficiency value (if less than unity).

Bending Stresses

The bending moment generated by a longitudinal decelerative force can be considered as shown in figure 18. For normal operating deceleration, the factor, F_1, is 0.35 and for extreme dynamic deceleration it is 0.70. The tensile stress in the bottom of the tank and the compressive stress in the top of the tank can be computed using:

$$S_{x6} = M/Z_e = 0.35(F_u+W_u)H_v/Z_e$$

where:

(F_u+W_u) = total weight of rear of trailer, lbs.
H_v = height from road to centerline of tank cross section, inches.
Z_e = elastic section modulus of tank cross section.

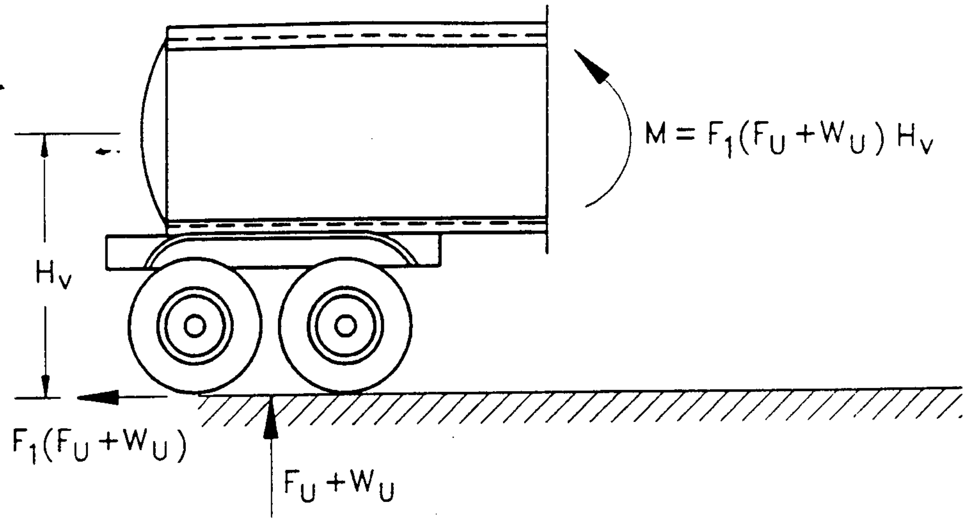


Figure 18. Idealization of cargo tank for computation of flexural tensile and compressive stresses due to longitudinal decelerative force from application of trailer brakes.

Example 14. Stresses due to bending moment generated by a decelerative force (trailer brakes).

Continuing from example 10, and given the following values, the bending moment due to trailer brake application only is computed as shown.

Distance from the application of the force (at the road surface) to the center of the tank (H_v) = 85 inches

Elastic section modulus (Z_e) = 421.1 in^3
Normal operating load factor (F_1) = 0.35

S_{x6} = M/Z_e = $[F_1(F_u+W_u)(H_v)]/Z_e$
S_{x6} = [0.35(26,830 + 6,000)(85)]/421.1
S_{x6} = 2,319 psi

Stresses due to longitudinal acceleration:

Axial Stresses

The axial load generated by an accelerative force can be considered as shown in Figure 19. For normal operating longitudinal acceleration, the factor F_1, is 0.35 and for extreme dynamic longitudinal acceleration, is 0.70. The tensile stress can be computed by using:

$$S_{x7} = 0.35\ F_k/A$$

where:

F_k = reaction force at kingpin, lbs.
A = cross sectional area of tank, in^2

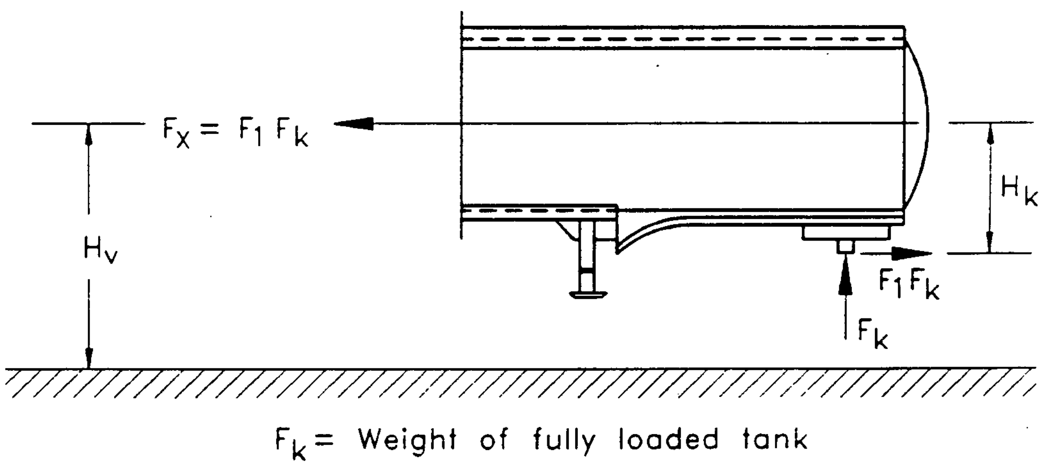


Figure 19. Idealization of cargo tank for computation of axial tensile stress due to accelerative force.

Example 15. Stresses due to axial force generated by an accelerative force.

A cylindrical tank is subject to an accelerative force of 0.35 times the reaction force at the kingpin during normal operating loads. The area, A, of the tank is 29.6 inches and the reaction force, F_k is 23,170 lbs. Axial stress due to the load is computed by:

$$S_{x7} = 0.35(F_k)/A = 0.35(23{,}170)/29.6$$
$$S_{x7} = 274 \text{ psi (tension)}$$

Bending Stresses

The bending moment generated by an accelerative force can be considered as shown in Figure 20. For normal operating acceleration, the factor, F_1, would be 0.35 and for extreme dynamic loading, it would be 0.70. The tensile stress in the bottom of the tank and the compressive stress in the top of the tank can be computed using:

$$S_{x8} = M\backslash Z_e = 0.35\ F_k H_k / Z_e$$

where:

F_k = reaction force at kingpin, lbs.
H_k = distance from transverse hinge point of fifth wheel to centroid of transverse cross section of tank, in.
Z_e = elastic section modulus of tank, in^3.

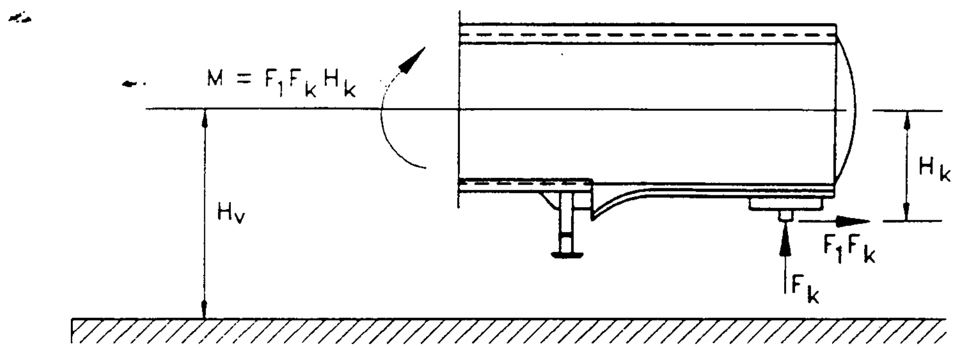


Figure 20. Idealization of cargo tank for computation of flexural tensile and compressive stresses due to accelerative force.

Example 16. Stresses due to bending moment generated by accelerative force.

Continuing from Example 15, and given the following values, the bending moment at the kingpin location due to acceleration can be computed as shown.

The height from the fifth wheel hinge point to the center of the tank (H_k) = 31.6 inches.
Elastic section modulus (Z_e) = 421.1 in^3
Normal operating load factor (F_l) = 0.35
F_k = reaction of kingpin = 23,170 lbs.

$$S_{x8} = M/Z_e = [(F_l)(F_k)(H_k)]/Z_e$$
$$S_{x8} = [0.35(23{,}170)(31.6)]/421.1$$
$$S_{x8} = 609 \text{ psi}$$

Stresses due to lateral acceleration:

Shear Stress

Flexural shear stresses and torsional shear stresses generated by a lateral accelerative force can be addressed as illustrated in figures 19 and 20. Weight of the tank, appurtenances and lading are multiplied for a factor, F_3, which is the lateral accelerative force in g's. It's magnitude is 0.2 for normal operating loadings and 0.4 for extreme dynamic loadings.

For flexural shear stresses, lateral supports are assumed at the vertical center of the tank and lateral shear forces are determined as illustrated in figure 21.

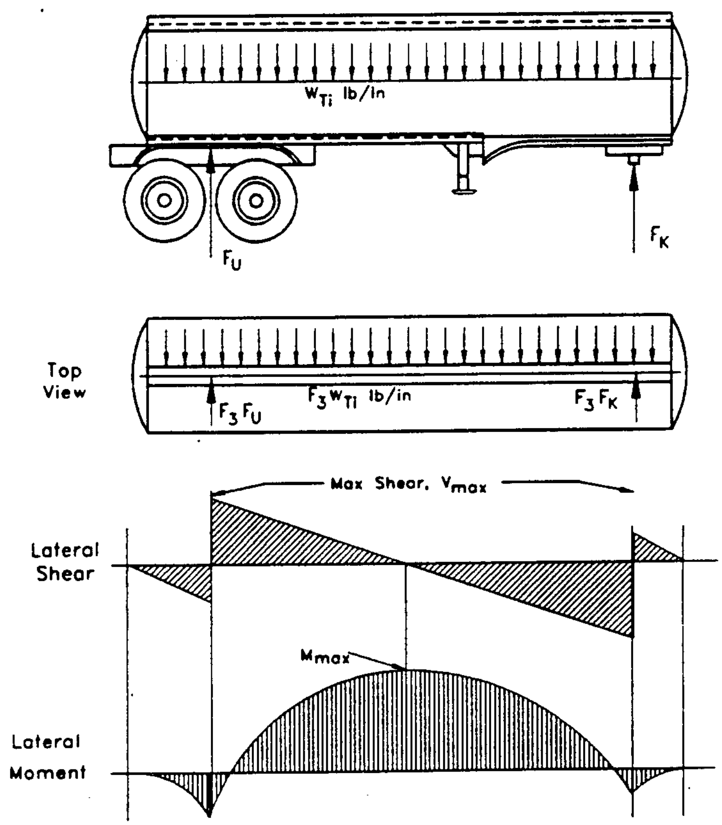


Figure 21. Shear force diagram for a lateral accelerative force.

Example 17. Computation of shear and bending stress due to a lateral accelerative force.

A cylindrical cargo tank has a reaction force at the undercarriage, F_u, of 26,830 lbs. The tank has an inside radius, R_i, of 28.46 inches and an outside radius, R_o, of 28.625 inches. The lateral accelerative force for normal operating loadings is 0.2 times each of the reaction forces. The lateral shear force diagram would be the same as the diagram from example 9 with all values multiplied by 0.2. The maximum shear stress in the wall of the tank would occur at the top and bottom of the tank on a section immediately forward of the reaction force at the undercarriage and is computed as follows:

$$S_{s3} = V_{max}/(0.5A)$$
$$A = \pi(R_o^2 - R_1^2)$$
$$A = \pi(28.625^2 - 28.46^2)$$
$$A = 29.6 \text{ in}^2$$
$$S_{s3} = 0.2(20{,}830)/(29.6/2)$$
$$S_{s3} = 281 \text{ psi}$$

The maximum bending moment in the lateral direction from figure 21 is 397,880 in-lbs. For this example, the section modulus for bending in the lateral direction is assumed to be 421.1; however, it is noted that the value might differ from that for bending in the vertical direction. The bending stresses are:

$$S_{x9} = M/Z_e$$
$$S_{x9} = 397{,}880/421.1$$
$$S_{x9} = \pm 945 \text{ psi}$$

The loading condition for torsional shear can be visualized by considering the tank (trailer) to be fixed against rotation at the front end and subjected to an overturning torque at the rear axles as shown in figure 22.

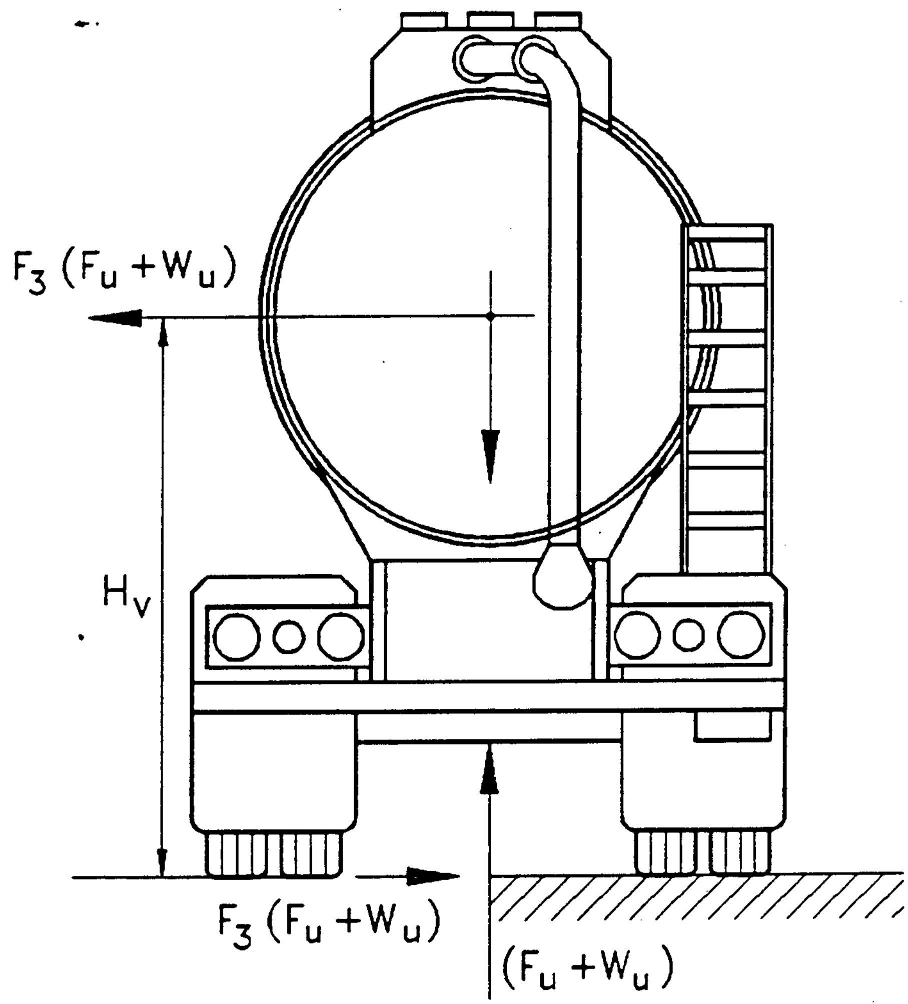


Figure 22. Illustration of the lateral accelerative force that produces torsional loading in a tank trailer.

The overturning torque for normal operating loading is:

$$T = F_3(F_u+W_u)H_v$$
$$T = 0.2\ (F_u+W_u)H_v$$

For tanks having circular cross sections of uniform wall thickness, the torsional shear stress may be computed using:

$$S_{s4} = TR_o/J$$

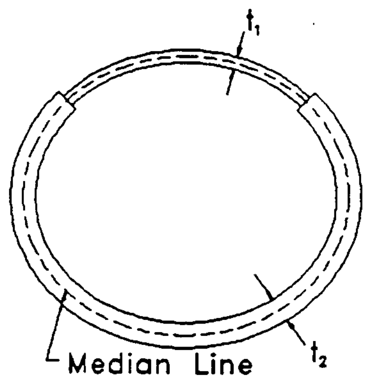
where:

S_{s4} = torsional shear stress in tank wall, psi.

R_o = outside radius of tank, in.

$J = 2\pi R_{avg}^3 t$

For thin-walled tanks having non-circular cross sections or circular cross sections of non-uniform wall thickness the torsional shear stresses may also be computed using the following procedure.



$$S_{s4} = T/2tA_m$$

where:

T = applied torque

t = wall thickness at point where stress is being computed

A_m = area enclosed by median line of tank wall

Figure 23. Definition of torsional shear stress formula for tanks having non-circular cross sections or circular cross sections of non-uniform wall thickness. Cross section may be any closed shape.

Example 18. Computation of torsional shear stress in cylindrical tanks.

A cylindrical tank has a reaction force at the undercarriage, F_u, of 26,830 lbs. and weight of the undercarriage, W_u, is 6,000 lbs. The dimensions of the tank are:

R_i = 28.46 inches
R_o = 28.625 inches
t = 0.165 inches
H_v = 85 inches

The overturning torque for normal operating loads is:

$$T = 0.2(W_u+F_u)H_v$$
$$T = 0.2(6{,}000+26{,}830)85$$
$$T = 558{,}110 \text{ in-lbs.}$$

Torsional shear stress is computed by:

$$S_{s4} = TR_o/J$$
$$R_o = 28.625$$
$$R_{avg} = (28.625+28.46)/2$$
$$R_{avg} = 28.54$$
$$J = 2\pi R^3_{avg}t$$
$$J = 2\pi(28.54)^3(.165)$$
$$J = 24{,}100 \text{ in}^4$$
$$S_{s4} = 558{,}110(28.625)/24{,}100$$
$$S_{s4} = 663 \text{ psi}$$

Example 19. Computation of torsional shear stress in tanks with non-circular cross sections or circular cross sections with non-uniform wall thickness.

The applied overturning torque is computed as outlined for cylindrical tanks. To determine the stress in a vessel at a point where the thickness is 0.165 inches, and the area enclosed by the median line of the tank wall, A_m, is 2,559 in^2.

$S_{s4} = T/2tA_m$

$T = 558{,}110$ in-lbs. (from previous example)

$A = \pi R^2_{avg}$

$A = \pi(28.54)^2$

$S_{s4} = 558{,}110/[2(.165)(2{,}559)]$

$S_{s4} = 661$ psi

The area enclosed by the median line of the tank wall, A_m, can be approximated by using the formula for the area of a ellipse.

$A = \pi R_{xo} R_{yo}$

or by using the manufacturers figure for gallons per inch and multiply by 231 in^3/gal

A_m = ___(gal/in)(231 in^3/gal) = _____ in^2

Stresses due to temperature gradients:

§178.345-3(b) requires that stresses due to the effect of temperature gradients resulting from lading and ambient temperature extremes be included in design.

Formulas for computing stresses due to a temperature gradient through the wall of a thin-walled cylinder for the case when the temperature gradient results from <u>steady heat flow</u> are given in Timoshenko's <u>Theory of Elasticity</u>[6]. If the temperature on the inside is T_i and the temperature on the outside is zero, the temperature at any distance, r, from the center is:

$$T = T_i \frac{\log(b/r)}{\log(b/a)}$$

where:

b is the outside radius of the cylinder
a is the inside radius of the cylinder

Note that ΔT, the difference in the temperature on the inside and outside, may be used for T_i if the temperature on the outside is not zero.

If temperature on the inside is higher than that on the outside, a compressive stress will exist (in the plane of the wall) on the inside surface of the wall and a tensile stress will exist (in the plane of the wall) on the outside surface. These stresses will exist in both the circumferential and longitudinal direction. They will be equal in magnitude in the two directions on the inside surface and on the outside surface. On the inside surface:

$$S_x = S_y = -\alpha E T_i/(2)(1-\nu)$$

On the outside surface:

$$S_x = S_y = \alpha E T_i/(2)(1-\nu)$$

where:

α is thermal coefficient for the wall material
E is modulus of elasticity for the wall material

T_i is difference in temperature on inside surface and outside surface, deg F.

ν is Poisson's ratio for the wall material

This analysis procedure is not applicable at locations near heads or bulkheads.

Allowable stresses:

Allowable tensile stress for non-ASME tanks.

The regulations in §178.345-3, state that the maximum calculated stress in the tank wall may not exceed allowable values given in Section VIII of the ASME Code or 25% of the tensile strength of the material. The regulations continue to state that the strength of materials may be established by reports from the manufacturer or by testing to a national standard, but the value may not exceed 120 percent of that given by the ASME Code or ASTM Standard. Section VIII refers to Section II, Part D, where extensive tables are given.

Allowable tensile stress for ASME tanks.

The ASME Code Section VIII and Section II, Part D are used to determine allowable tensile stress for materials at various temperatures. Extensive tables are included in Section II, Part D.

Allowable compressive stress for non-ASME tanks.

For non-ASME Code designs, the allowable compressive stress of a cargo tank wall may be computed using either the Alcoa formula or the Roark and Young formula: (reference case #13 and #15, Table 35, 5th ed.)

Alcoa formula with no factor of safety.

$$S_{bA} = (\pi/4)^2(E)/\{(R_s/t_s)[1+(R_s/t_s)^{1/2}/35]^2\}$$

Roark and Young formula with no factor of safety.

$$S_{bY} = 0.6Et/R_i$$

However, Roark and Young state that, for case #15, tests indicate an actual buckling strength of from 40 to 60 percent of the theoretical value, or:

$$S_{bY} = 0.3Et/R_i$$

This latter formula compares more closely with the Alcoa formula and should be used for computing compressive buckling stress for non-ASME tanks.

A factor of safety of 1.5, applied to either of the above formulas should be used for computing the allowable compressive strength of cargo tanks. The allowable compressive stress would be:

$$S_b = S_{bA}/1.5$$

or

$$S_b = S_{bY}/1.5$$

For ASME certified tank designs, the ASME Code must be followed.

Example 20. Compressive buckling strength of cargo tank shell (non-ASME).

A DOT 407 tank is made of SB209 A95454-H32 aluminum (S_u = 36,000 psi, S_{yield} = 26,000 psi). The outside diameter of the shell is 63.5 inches. The top portion of the shell, which is subjected to an axial compressive stress, is 0.165 inches thick.

The critical compressive buckling of the top portion of the shell is to be computed. The inside radius of the shell is 63.5/2 - .165 = 31.585 inches. The modulus of elasticity is $12.5X10^6$ psi.

For the Alcoa formula with no factor of safety, the critical buckling strength would be:

$$S_{bA} = (\pi/4)^2(E)/\{(R_s/t_s)[1+(R_s/t_s)^{1/2}/35]^2\}$$

$$S_{bA} = (\pi/4)^2(12.5X10^6)/\{(191.4)[1+(191.4)^{1/2}/35]^2\}$$
$$S_{bA} = 20{,}693 \text{ psi}$$

The allowable compressive buckling strength would be:

$$S_b = S_{bA}/1.5$$
$$S_b = 20{,}693/1.5$$
$$S_b = 13{,}795 \text{ psi}$$

For the Roark and Young formula with no factor of safety, the critical buckling strength would be:

$$S_{bY} = 0.3Et/R_i$$
$$S_{bY} = 0.3(12.5X10^6)(.165)/31.585$$
$$S_{bY} = 19{,}590 \text{ psi}$$

The allowable compressive buckling strength would be:

$$S_b = S_{bY}/1.5$$
$$S_b = 19{,}590/1.5$$
$$S_b = 13{,}060 \text{ psi}$$

Allowable compressive stress for ASME tanks.

Tables and charts in the ASME Code are used to determine the allowable compressive stress in a cargo tank wall. The procedure is illustrated in example 6 on page 57. The allowable compressive stress is the value of B from Step 5, page 58, $B = S_b = 3{,}500$ psi.

Stiffener Rings:

Stiffener rings, baffles and/or bulkheads are used to provide lateral support for walls of cargo tanks. This is done to generally maintain the shape of the tank and to provide lateral stability to the portion of the tank wall subject to compressive stresses. Ability of a stiffener ring to perform its intended function is dependent upon its strength and stiffness. Section 178.345-7 of DOT regulations places requirements on the spacing and elastic section modulus of stiffener rings. A portion of the tank wall may be considered to function compositely with a ring as shown in figure 24. The ASME Code has a similar requirement for moment of inertia of a ring. A portion of the tank wall may be used if the engineer chooses to do so as illustrated in figure 26.

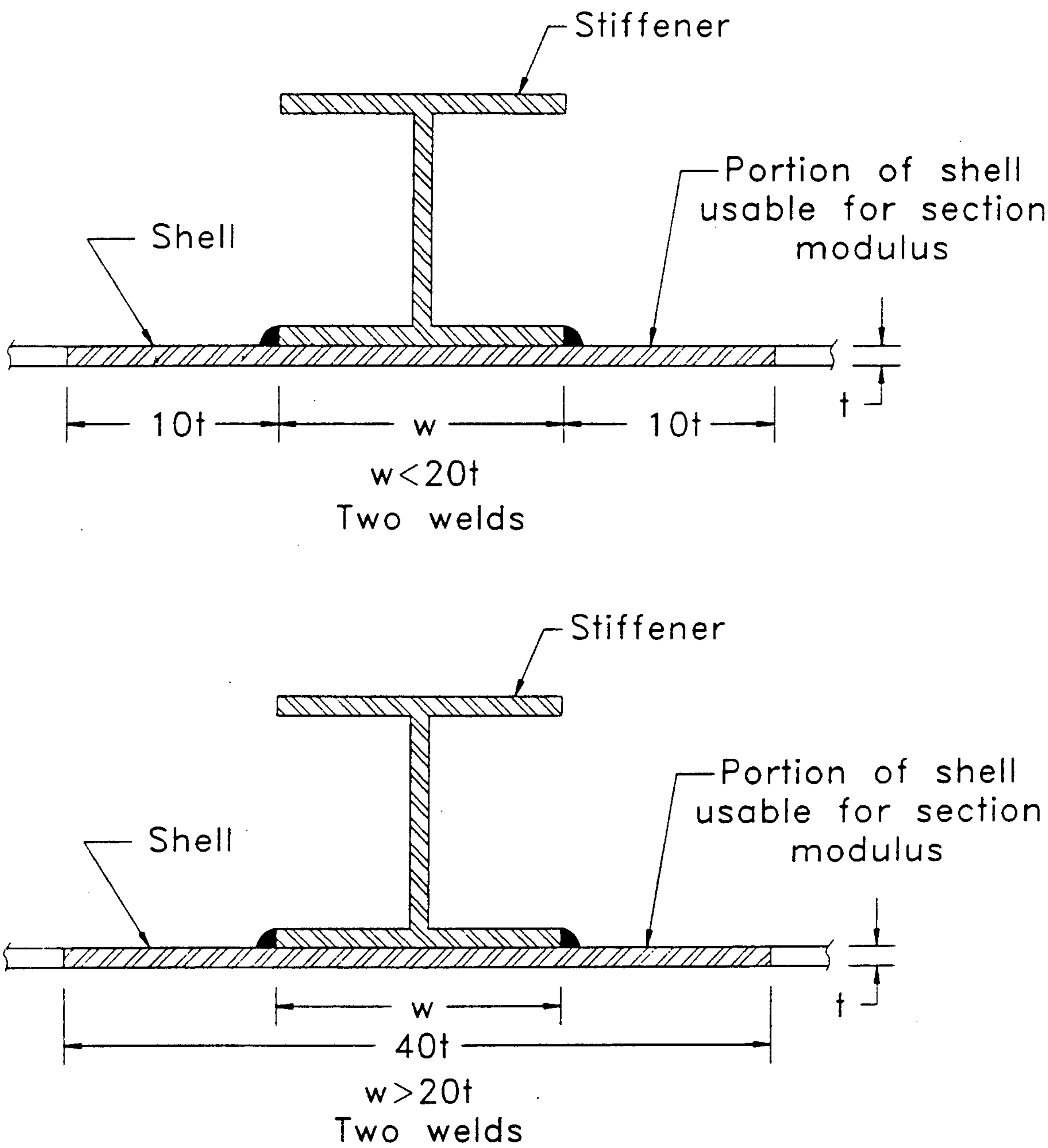


Figure 24. Definition of cross section for computation of section modulus of ring stiffener for DOT (non-ASME) tanks.

Example 21. Computation of section modulus for ring stiffener by DOT regulations (non-ASME).

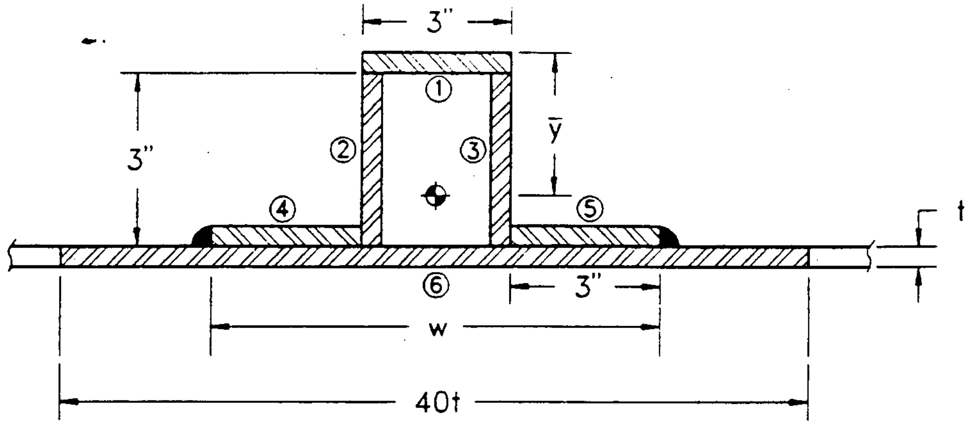


Figure 25. Cross section of ring stiffener and tank wall. If w is greater than 20t, the maximum length of wall usable is 40t.

w>20t, so length of shell usable is 40t.

t = 0.25 inches
w = 9 inches

Determine centroid of section.

Section #	A_i	$\bar{y}_i$	$A_i\bar{y}_i$
1	0.75	0.125	0.09375
2	0.75	1.75	1.3125
3	0.75	1.75	1.3125
4	0.75	3.125	2.34375
5	0.75	3.125	2.34375
6	2.5	3.375	8.4375
	6.25 in^2		15.84375

$\bar{x}_i$ = centerline of section (symmetrical)
$\bar{y}_i = \Sigma A_i\bar{y}_i/\Sigma A_i = 15.84375/6.25 = 2.54$ inches

Determine section modulus

$$I_{TOT} = I_0 + Ad^2$$

Section 1	$I = bh^3/12 = (3)(0.25)^3/12 = 0.00391$
	$Ad^2 = (0.75)(2.54-0.125)^2 = 4.3561\ in^4$
Section 2 & 3	$I = bh^3/12 = (0.25)(3)^3/12 = 0.5625$
	$Ad^2 = (0.75)(2.54-1.75)^2 = 0.4622\ in^4$
Section 4 & 5	$I = bh^3/12 = (3)(0.25)^3/12 = 0.00391$
	$Ad^2 = (0.75)(2.54-3.125)^2 = 0.2611\ in^4$
Section 6	$I = bh^3/12 = (10)(0.25)^3/12 = 0.0130$
	$Ad^2 = (2.5)(2.54-3.375)^2 = 1.7640\ in^4$

Section 1	$= I+Ad^2 = (0.00391+4.3561) = 4.3600\ in^4$
Section 2 & 3	$= I+Ad^2 = (0.5625+0.4622) = 1.0247\ in^4$
Section 4 & 5	$= I+Ad^2 = (0.00391+0.2611) = 0.2650\ in^4$
Section 6	$= I+Ad^2 = (0.0130+1.7640) = 1.7770\ in^4$
	$I_{TOT} = 4.3600+2(1.0247)+2(0.2650)+1.7770$
	$I_{TOT} = 8.7164\ in^4$
	$c = 2.65$ inches

Section modulus $Z_e = I/c = 8.7164/2.65 = 3.289\ in^3$

The required section modulus per §178.345-7 for structural steel is:

$I/c \quad = 0.00027w\ell$

$w = 57.25$ inches

$\ell = 60$ inches (max allowed)

$= 0.00027(57.25)(60) = 0.927\ in^3 < 3.289\ in^3$ OK

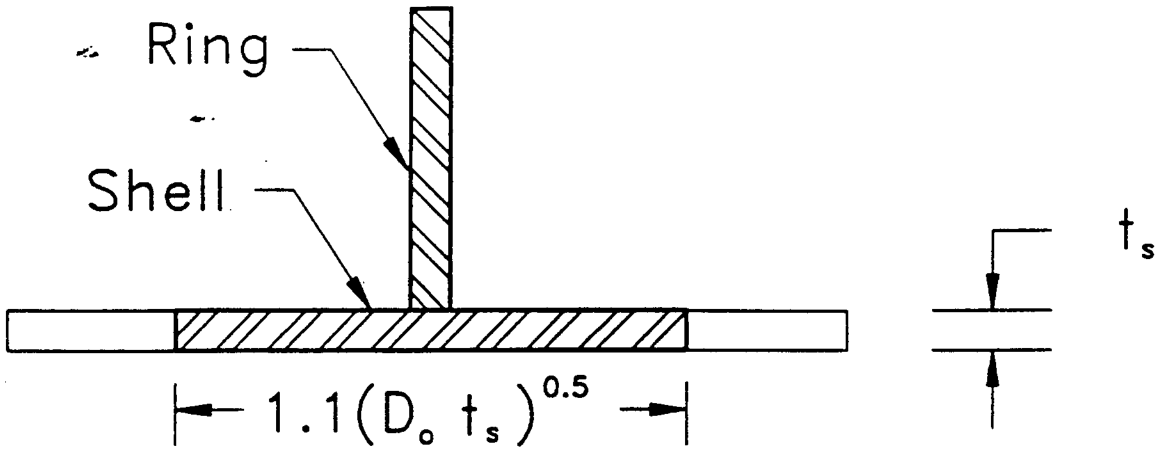


Figure 26. Definition of cross section for computation of moment of inertia of ring stiffener for ASME tanks.

Example 22. Determine adequacy of a circumferential stiffening ring for ASME tanks. (ASME Code UG-29)

Minimum Required Thickness t = 0.20″
Stiffening Rings are 3/8″ by 3″ strap
Tank Diameter = 60″
Length between straps = 72″ (equally spaced so L_s = 72″)
Thickness of tank shell = t_s = 0.25″
Cross sectional area of ring $A_s = 0.375 \times 3 = 1.125$ in^2
External Pressure = P = 14.7 psi (Vacuum tank)
Material - SA240/316L stainless steel
Design Temperature 300°F

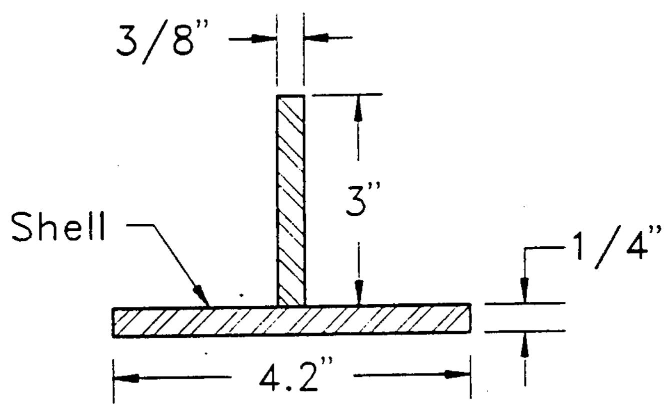


Figure 27. Cross section of ring stiffener and portion of tank for computation of moment of inertia.

The length of the shell allowed to be used is $1.10(D_o t_s)^{0.5}$ = 4.2″

The adequacy shall be determined by UG-29 as follows:

Step 1 Using A_s, t, D_o, P, and L_s calculate
$B = 0.75(PD_o)/[t+(A_s/L_s)]$
$B = .75(14.7)(60)/(.20+1.125/72)$
$B = 3068$

Step 2 Go to the applicable material chart in Subpart 3 of Section II, Part D. For SA240/316 stainless steel the material chart is found in FIG HA-4 on page 681 of Section II, Part D.

Steps 3&4 Use this chart to determine the value of A. For B=3068 and a design temperature of 300° F. The value of A can be determined to be 2.4×10^{-4}

Step 5 Does not apply since A can be determined by Step 4.

Step 6a For considering only the stiffening ring, the required moment of inertia is:

$I_s = [D_o^2 L_s(t+A_s/L_s)A]/14$

$I_s = [60^2(72)(.20+1.125/72)2.4 \times 10^{-4}]/14$

$I_s = 0.96$

Step 7a Moment of inertia for only the stiffening ring is:

$I = bh^3/12 = .375(3)^3/12 = 0.84 < 0.96$

Note: It is up to the designer whether to use the stiffener alone as calculated in steps 6a and 7a, or use the stiffener and the allowable shell contribution (as given in the definition of I′ in UG-29) which is calculated in steps 6b and 7b below:

Step 6b For considering the stiffening ring and the allowable shell contribution, the required moment of inertia is:

$I'_s = [D_o^2 L_s(t+A_s/L_s)A]/10.9$

$I'_s = [60^2(72)(.2+1.125/72)2.4 \times 10^{-4}]/10.9$

$I'_s = 1.23$

Step 7b Moment of inertia for the stiffening ring <u>and</u> the shell contribution is:

Section #	Area	$\bar{y}_i$	$A\bar{y}_i$
1 2	1.125 1.05	1.75 0.125	1.97 0.13
	$\sum A=2.175$		$\sum A\bar{y}=2.1$

$\bar{y} = \sum A\bar{y}/\sum A = 2.1/2.175 = 0.97$

$I_1 = bh^3/12+Ad^2 = .375(3)^3/12+1.125(1.75-.97)^2 = 1.528$

$I_2 = bh^3/12+Ad^2 = 4.2(.25)^3/12+1.05(.97-.125)^2 = .755$

$I' = I_1+I_2 = 1.528+.755 = 2.283 > I's$

Step 8 If either $I > I_s$ or $I' > I's$, the design is adequate.

In this case, $I' > I's$ so the stiffener design is adequate.

Design of tanks by ASME Code.

Design requirements of the ASME Code that are applicable to ASME certified cargo tanks are included in Section VIII-Division 1 and are as follows:

UG-16	GENERAL
UG-20	DESIGN TEMPERATURE
UG-21	DESIGN PRESSURE
UG-22	LOADINGS
UG-23	MAXIMUM ALLOWABLE STRESS VALUES
UG-25	CORROSION
UG-27	THICKNESS OF SHELLS UNDER INTERNAL PRESSURE
UG-28	THICKNESS OF SHELLS AND TUBES UNDER EXTERNAL PRESSURE
UG-29	STIFFENING RINGS FOR CYLINDRICAL SHELLS UNDER EXTERNAL PRESSURE
UG-30	ATTACHMENT OF STIFFENING RINGS
UG-32	FORMED HEADS, AND SECTIONS, PRESSURE ON CONCAVE SIDE
UG-33	FORMED HEADS, PRESSURE ON CONVEX SIDE
UG-36	OPENINGS IN PRESSURE VESSELS
UG-37	REINFORCEMENT REQUIRED FOR OPENINGS IN SHELLS AND FORMED HEADS
UW-12	JOINT EFFICIENCIES

Appendix 1 of Section VIII

Appendix 13 of Section VIII

Appendix G of Section VIII

Appendix L of Section VIII

The engineer is referred to Section VIII-Division 1 and Section II-Materials Part D-Properties for guidance on analysis procedures and design values.

CHAPTER 5. LOADING REQUIREMENTS

General requirements for DOT 406, 407, 412 cargo tank motor vehicles are given in Section 178.345.

Specific requirements for DOT 406 (§178.346).

The MAWP must be no lower than 2.65 psig and no higher than 4 psig.

Vacuum loaded cargo tanks must not be constructed to this specification.

DOT 406 tanks must be "constructed in accordance with the ASME Code" with exceptions.

Minimum thicknesses of materials are set forth in §178.346-2

Specific requirements for DOT 407 (§178.347).

The tank must be of circular cross-section and have an MAWP of at least 25 psig. Any tank with an MAWP greater than 35 psig and any tank designed to be loaded by vacuum must be "constructed and certified in accordance with the ASME Code". External pressure for a tank loaded by vacuum must be at least 15 psi.

Tanks with MAWP of 35 psig or less must be "constructed in accordance with ASME Code" with exceptions.

Minimum thicknesses of materials are prescribed in §178.347-2.

Specific requirements for DOT 412 (§178.348).

MAWP must be at least 5 psig. If loaded by vacuum, MAWP must be at least 25 psig internal and 15 psig external. If MAWP is greater than 15 psig, tank must be circular cross-section. If MAWP is greater than 15 psig, tank must be "constructed and certified in conformance with ASME Code". If

MAWP is 15 psig or less, tank must be "constructed in accordance with ASME Code" with exceptions.

Minimum thicknesses of tank materials are prescribed in §178.348-2.

For all three designs, the principal normal stresses should be computed for the appropriate combinations of loads using the stress transformation equation. All potential points of maximum stress should be considered.

Structural evaluation procedures.

The following paragraphs provide suggested guidance (and a checklist) for organizing a structural evaluation.

1. Identify geometry of tank and materials used.
2. Identify whether ASME or non-ASME design.
3. Check minimum thickness of tank heads and shell using minimum thickness tables in DOT regulations.
4. Check spacing of circumferential reinforcement.
   A. Check section modulus of ring stiffeners if they are used for circumferential reinforcement.
   B. Check adequacy of baffles and bulkheads if they are used for circumferential reinforcement.
5. Check structural integrity of tank.
   A. Check structural integrity of tank walls using loading combinations in the following section.
   B. Check adequacy of heads using procedures in parts UG-32 and UG-33 of the ASME Code.
6. Check adequacy of rollover protection.
7. Check adequacy of bottom damage protection.
8. Check adequacy of rear-end protection.

Loading Combinations.

The loading combination that should be considered for static design is:

COMB. SA - Combined stresses due to:

- pressure/vacuum
  S_{x1}, S_{y1}
- static head
  S_{x2}, S_{y2}
- static gravity loads
  S_{x3}, S_{s1}

The loading combinations that should be considered for normal operating loadings include the following:

COMB. NA - Combined stresses due to:

- pressure/vacuum (if appropriate)
  S_{x1}, S_{y1}
- static head
  S_{x2}, S_{y2}
- static gravity loads
  S_{x3}, S_{s1}
- vertical acceleration
  S_{x4}, S_{s2}
- longitudinal deceleration created by trailer braking
  S_{x5}, S_{x6}
- lateral acceleration
  S_{x9}, S_{s3}, S_{s4}

COMB. NB - Combined stresses due to:

- pressure/vacuum (if appropriate)
  S_{x1}, S_{y1}
- static head
  S_{x2}, S_{y2}
- static gravity loads
  S_{x3}, S_{s1}
- vertical acceleration
  S_{x4}, S_{s2}

- longitudinal acceleration
  S_{x7}, S_{x8}
- lateral acceleration
  S_{x9}, S_{s3}, S_{s4}

COMB. NC - Combined stresses due to:
- pressure/vacuum (if appropriate)
  S_{x1}, S_{y1}
- Static head
  S_{x2}, S_{y2}
- static gravity loads
  S_{x3}, S_{s1}
- vertical acceleration
  S_{x4}, S_{s2}
- longitudinal deceleration created by tractor braking
  S_{x10}, S_{x11}
- lateral acceleration
  S_{x9}, S_{s3}, S_{s4}

The loading combinations that should be considered for <u>extreme dynamic loadings</u> include:

COMB. EA - Combined stresses due to:
- pressure/vacuum (if appropriate)
  S_{x1}, S_{y1}
- static head
  S_{x2}, S_{y2}
- static gravity loads
  S_{x3}, S_{s1}
- longitudinal deceleration created by trailer braking
  S_{x5}, S_{x6}

COMB. EB - Combined stresses due to:
- pressure/vacuum (if appropriate)
  S_{x1}, S_{y1}
- static gravity loads
  S_{x3}, S_{s1}
- static head

S_{x2}, S_{y2}
- longitudinal acceleration
  S_{x7}, S_{x8}

COMB. EC - Combined stresses due to:
- pressure/vacuum (if appropriate)
  S_{x1}, S_{y1}
- static gravity loads
  S_{x3}, S_{s1}
- static head
  S_{x2}, S_{y2}
- longitudinal deceleration created by tractor braking
  S_{x10}, S_{x11}

COMB. ED - Combined stresses due to:
- pressure/vacuum (if appropriate)
  S_{x1}, S_{y1}
- static gravity loads
  S_{x3}, S_{s1}
- static head
  S_{x2}, S_{y2}
- vertical acceleration
  S_{x4}, S_{s2}

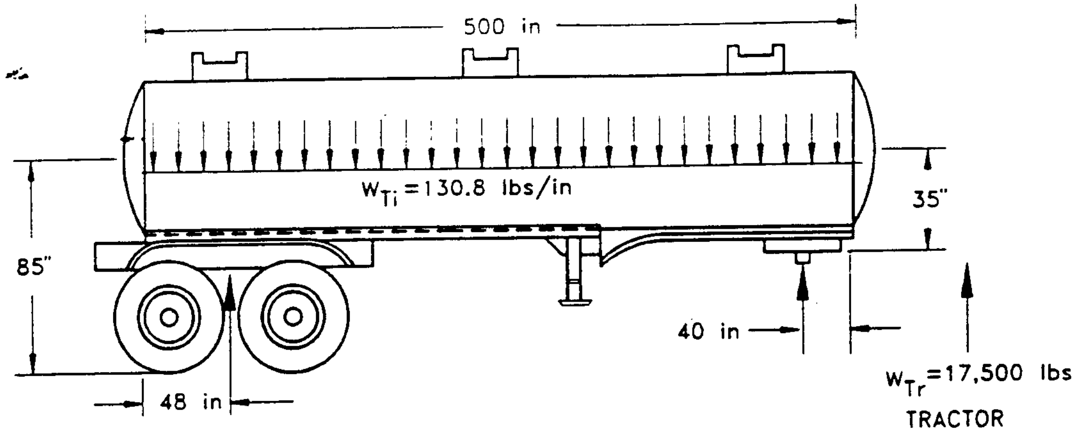
COMB. EE - Combined stresses due to:
- pressure/vacuum (if appropriate)
  S_{x1}, S_{y1}
- static head
  S_{x2}, S_{y2}
- static gravity loads
  S_{x3}, S_{s1}
- lateral acceleration
  S_{x9}, S_{s3}, S_{s4}

The effective stress (maximum/minimum principal stress) at any point must be determined for appropriate combinations of static and normal operating loadings, and combinations of static and extreme dynamic loadings using the following stress transformation equation in Section 178.345-3.

$$S = 0.5(S_x+S_y) \pm [0.25(S_x-S_y)^2+S_s^2]^{0.5}$$

Example 23. Structural integrity analysis of DOT 412 cargo tank.

Weight of Tank Shell = W_s = 5,400 lbs.
Weight of Tractor = W_{Tr} = 17,500 lbs.
Max. Weight of Lading = W_L = 60,000 lbs.
Max. Weight of Tank Plus Lading = W_T = 65,400 lbs.
Lading is Muriatic Acid With Specific Gravity γ = 1.2
Weight of Undercarriage = W_u = 8,000 lbs.
Thickness of Tank Wall = t = 0.219 inches
MAWP = P_m = 45 psi
Material = SA 240/316L
H_k = 35 inches
H_v = 85 inches
Outside diameter of tank = 60 inches



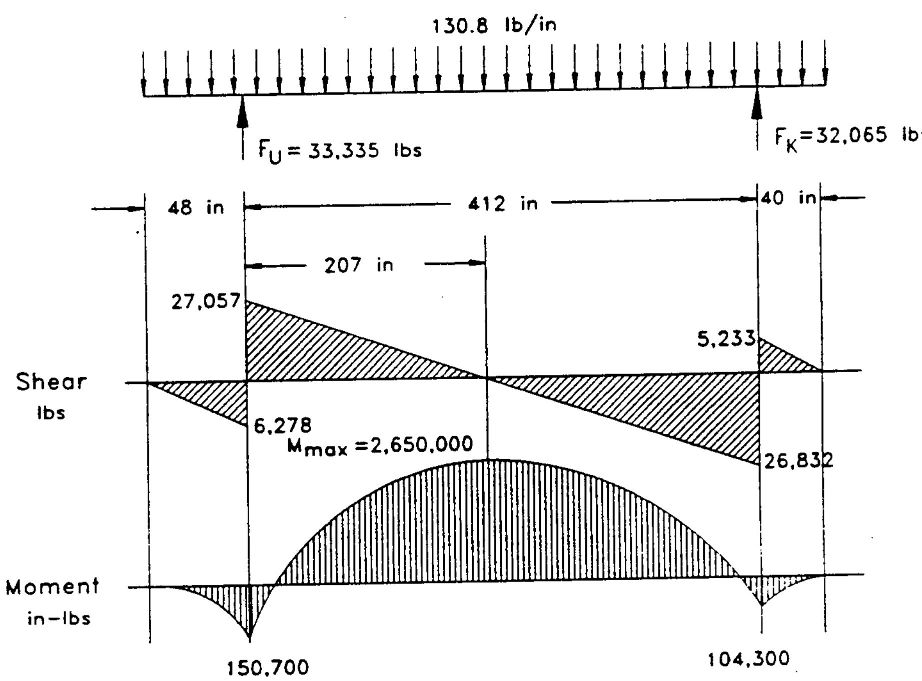


Figure 28. Load, shear and moment diagrams for example 23.

(1) Determine Reactions

- Weight uniformly distributed
- No major appurtances (>1,000 lbs.)
- Weight per inch, $W_{Ti} = W_T/L = 65{,}400/500 = 130.8$ lbs/in.

$\Sigma M_{FU} = 65{,}400\ (500/2-48) - F_k(500-40-48) = 0$

$F_k = 32{,}065$ lbs.

$F_u = 33{,}335$ lbs.

(2) Locate Point of Maximum Moment.

- at point where shear (V)=0

L_{max} at 33,335/130.8 = 255 inches from rear of tank
or 27,057/130.8 = 207 inches from rear support.

(3) If no weld is at or near the center of the tank or the joint efficiency is 1.0, stresses should be analyzed at the sections of maximum moment and maximum shear at the top, bottom and side centerline of the tank.

(4) Stresses Due to Internal Pressure

Circumferential Stress

$S_{y1} = PR_i/t + 0.6P = [45(29.78)/.219]+.6(45)$

$S_{y1} = 6{,}146$ psi

$S_{y1} = 6{,}146$ psi at top, bottom and side centerline

Longitudinal Stress

$S_{x1} = PR_i/2t - .2P = [45(29.78)/2(.219)]-.2(45)$

$S_{x1} = 3{,}051$ psi

$S_{x1} = 3{,}051$ psi and acts at top, bottom and side centerline of tank.

(5) Stress Due to Static Head

- At bottom of tank

$P = (D_i/12)(.433)\gamma = (59.56/12)(.433)(1.2) = 2.58$ psi

$S_{y2} = PR_i/t + .6P = [2.58(29.78)/.219]+.6(2.58) = 352$ psi

$S_{x2} = PR_i/2t - .2P = [2.58(29.78)/2(.219)]-.2(2.58) = 175$ psi

	S_{y2}	S_{x2}
At Bottom	352 psi	175 psi
At Side Centerline	352/2 = 176 psi	175/2 = 87 psi
At Top	0 psi	0 psi

(6) <u>Static Shear Stress Due to Static Weight.</u>

$S_{s1} = V_{max}/(0.5A)$

$A = \pi(R_o^2-R_i^2) = 41.3\ in^2$

$S_{s1} = 27{,}057/[0.5(41.3)]$

$S_{s1} = 1{,}310$ psi

$S_{s1} = 0$ at top and bottom

$S_{s1} = 1{,}310$ psi at side centerline

(7) <u>Bending Stress Due to Static Weight</u>

Maximum Moment

$M_{max} = 2{,}650{,}000$ in-lbs.

Section Properties

$I = (\pi/4)(R_o^4-R_i^4) = (\pi/4)(30^4-29.78^4)$

$I = 18{,}457\ in^4$

$c = 30$ inches

$Z_e = I/c = 18{,}457/30 = 615.2\ in^3$

$S_{x3} = M_{max}/Z_e = 2{,}650{,}000/615.2$

$S_{x3} = 4{,}308$ psi

$S_{x3} = 4{,}308$ psi on bottom

$S_{x3} = 0$ at side centerline

$S_{x3} = -4{,}308$ psi on top

Normal Operating Loads

(8) Axial Stress Due to Normal Longitudinal Acceleration.

Factor, F_l = 0.35

$S_{x7} = .35F_k/A = .35(32{,}065/41.3)$

S_{x7} = 272 psi at top, bottom and side centerline

(9) Bending Stress due to Normal Longitudinal Acceleration.

Factor, F_l = 0.35

$S_{x8} = M/Z_e = 0.35F_kH_k/Z_e$

$S_{x8} = 0.35(32{,}065)35/615.2$

S_{x8} = 638 psi

S_{x8} = 638 at bottom

S_{x8} = 0 at side centerline

S_{x8} = -638 at top

(10) Axial Stress Due to Normal Longitudinal Deceleration. Trailer braking only

Factor, F_l = 0.35

$S_{x5} = F/A = .35(F_u+W_u)/A$

$S_{x5} = .35(33{,}335+8{,}000)/41.3$

S_{x5} = 350 psi at top, bottom and side centerline

(11) Bending Stress Due to Normal Longitudinal Deceleration. Trailer braking only

Factor, F_l = .35

$S_{x6} = M/Z_e = [0.35(F_u+W_u)(H_v)]/Z_e$

$S_{x6} = [0.35(33{,}335+8{,}000)(85)]/615.2$

S_{x6} = 1,999 psi

S_{x6} = 1,999 psi at bottom

S_{x6} = 0 at side centerline

S_{x6} = -1,999 psi at top

(12) Axial Stress Due to Normal Longitudinal Deceleration. Tractor braking only.

Factor, F_1 = 0.35
S_{x10} = F/A = 0.35(F_k+W_{Tr})/A
S_{x10} = 0.35(32,065+17,500)/41.3
S_{x10} = -420 psi at top, bottom and side centerline

(13) Bending Stress Due to Normal Longitudinal Deceleration. Tractor braking only.

Factor, F_1 = 0.35
S_{x11} = M_{max}/Z_e
where: M_{max} = 0.35(F_k+W_{Tr})H_k
M_{max} = 0.35(32,065+17,500)(35)
M_{max} = 607,000 in-lbs.
S_{x11} = 607,000/615.2
S_{x11} = 987 psi
S_{x11} = -987 psi at bottom
S_{x11} = 0 psi at side centerline
S_{x11} = 987 psi at top

(14) Bending Stresses Due to Normal Vertical Acceleration.

Factor, F_2 = 0.35
S_{x4} = M_{max}/Z_e
where: M_{max} = (0.35)(2,650,000)
S_{x4} = 0.35(2,650,000)/615.2
S_{x4} = 1,508 psi
S_{x4} = 1,508 psi at bottom
S_{x4} = 0 at side centerline
S_{x4} = -1,508 psi at top

(15) Bending Stresses Due to Normal Lateral Acceleration.

Factor, F_3 = 0.2
S_{x9} = M_{max}/Z_e
where: M_{max} = (0.2)(2,650,000)

S_{x9} = (0.2)(2,650,000)/615.2
S_{x9} =
S_{x9} = ±862 psi (tension on one side and compression on other side)

For some tanks, the value of Z_e for lateral bending might differ from that for vertical bending.

(16) Shear Stress Due to Normal Vertical Acceleration.

Factor, F_2 = 0.35
$S_{s2} = V_{max}/(0.5A)$
S_{s2} = 0.35(27,057)/[0.5(41.3)]
S_{s2} = 459 psi
S_{s2} = 0 at top and bottom
S_{s2} = 459 psi at side centerline

(17) Shear Stress Due to Normal Lateral Acceleration.

Factor, F_3 = 0.2
$S_{s3} = V_{max}/(0.5A)$
S_{s3} = 0.2(27,057)/[0.5(41.3)]
S_{s3} = 262 psi
S_{s3} = 262 psi at top and bottom
S_{s3} = 0 at side centerline

(18) Torsional Shear Stress Due to Normal Lateral Accelerative Force.

Factor, F_3 = 0.2
$T = 0.2(W_u+F_u)H_v$
T = 0.2(8,000+33,335)(85)
T = 702,695 in-lbs.

$J = 2\pi R_{avg}^3 t$
where: R_{avg} = (30+29.78)/2 = 29.89 inches
$J = 2\pi(29.89)^3(.219)$
$J = 36{,}745\ in^4$

$S_{s4} = TR_o/J$

S_{s4} = 702,695(30)/36,745
S_{s4} = 574 psi
∴ S_{s4} = 574 psi at top, bottom and side centerline

Extreme Dynamic Loads

(19) <u>Axial Stress Due to Extreme Longitudinal Acceleration.</u>

Factor, F_1 = 0.70
$S_{x7} = 0.7F_k/A$
S_{x7} = 0.7(32,065)/41.3
S_{s7} = 543 psi at top, bottom and side centerline

(20) <u>Bending Stresses Due to Extreme Longitudinal Acceleration.</u>

Factor, F_1 = 0.70
$S_{x8} = M/Z_e = 0.7F_kH_k/Z_e$
S_{x8} = 0.7(32,065)(35)/615.2
S_{x8} = 1,277 psi
S_{x8} = 1,277 psi at bottom
S_{x8} = 0 at side centerline
S_{x8} = -1,277 psi at top

(21) <u>Axial Stress Due to Extreme Longitudinal Deceleration.</u>
<u>Trailer braking only</u>

Factor, F_1 = 0.7
$S_{x5} = F/A = 0.7(F_u+W_u)/41.3$
S_{x5} = 0.7(33,335+8,000)/41.3
S_{x5} = 701 psi at top, bottom and side centerline

(22) <u>Bending Stress Due to Extreme Longitudinal Deceleration.</u>
<u>Trailer braking only</u>

Factor, F_1 = 0.7
$S_{x6} = M/Z_e = [0.7(F_u+W_u)(H_v)]/Z_e$

S_{x6} = 0.7(33,335+8,000)(85)/615.2
S_{x6} = 3,998 psi
∴ S_{x6} = 3,998 psi at bottom
S_{x6} = 0 at side centerline
S_{x6} = -3,998 psi at top

(23) <u>Axial Stress Due to Extreme Longitudinal Deceleration. Tractor braking only.</u>

Factor, F_1 = 0.7
S_{x10} = F/A = 0.7(32,065+17,500)/41.3
S_{x10} = -840 psi (compression) at top, bottom and side centerline

(24) <u>Bending Stress Due to Extreme Longitudinal Deceleration. Tractor braking only.</u>

Factor, F_1 = 0.7
$S_{x11} = M_{max}/Z_e$
where: $M_{max} = 0.7(F_k+W_{Tr})H_k$
M_{max} = 0.7(32,065+17,500)(35)
M_{max} = 1,214,000 in-lbs.
S_{x11} = 1,214,000/615.2
S_{x11} = 1,973 psi
S_{x11} = -1,973 at bottom
S_{x11} = 0 psi at side centerline
S_{x11} = 1,973 psi at top

(25) <u>Bending Stresses Due to Extreme Vertical Acceleration.</u>

Factor, F_2 = 0.7
$S_{x4} = M_{max}/Z_e$
where: M_{max} = (0.7)(2,650,000) in-lbs.
S_{x4} = (0.7)(2,650,000)/615.2
S_{x4} = 3,015 psi
S_{x4} = 3,015 psi at bottom
S_{x4} = 0 at side centerline
S_{x4} = -3,015 psi at top

(26) <u>Shear Stress Due to Extreme Vertical Acceleration.</u>

Factor, $F_2 = 0.7$
$S_{s2} = V_{max}/0.5A = 0.7(27,057)/0.5(41.3)$
$S_{s2} = 917$ psi
$S_{s2} = 0$ at top and bottom
$S_{s2} = 917$ psi at side centerline

(27) <u>Shear Stress Due to Extreme Lateral Acceleration.</u>

Factor, $F_3 = 0.4$
$S_{s3} = 0.4(V_{max})/.5A$
$S_{s3} = 0.4(27,057)/.5(41.3)$
$S_{s3} = 524$ psi
$S_{s3} = 524$ psi at top and bottom
$S_{s3} = 0$ psi at side centerline

(28) <u>Torsional Shear Stress Due to Extreme Lateral Acceleration.</u>

Factor, $F_3 = 0.4$
$T = 0.4(W_u+F_u)H_v$
$T = 0.4(33,335+8,000)(85)$
$T = 1,405,390$ in-lbs.
$J = 36,745$ in^4 from Part 17
$S_{s4} = TR_o/J$
$S_{s4} = 1,405,390(30)/36,745$
$S_{s4} = 1,147$ psi
$S_{s4} = 1,147$ psi at top, bottom and side centerline

(29) <u>Bending Stress Due to Extreme Lateral Acceleration.</u>

Factor, $F_3 = 0.4$
$S_{x9} = M_{max}/Z_e$ (about vertical axis)
where: $M_{max} = 0.4(2,650,000)$
$M_{max} = 1,060,000$ in-lbs.
$Z_e = 615.2$ in^3 (assumed same as for vertical bending)
$S_{x9} = 1,060,000/615.2$
$S_{x9} = \pm 1,723$ psi at side centerlines

Table 2. Summary of Stresses Due to Static Loads

Load Description	Stress Designation	Normal Stress (psi)			Shear Stress (psi)		
		Bottom	Centerline	Top	Bottom	Centerline	Top
Internal Pressure	S_{y1}	6,146	6,146	6,146	—	—	—
	S_{x1}	3,051	3,051	3,051	—	—	—
Static Head	S_{x2}	175	87	0	—	—	—
	S_{y2}	352	176	0	—	—	—
Static Weight	S_{x3}	4,308	0	-4,308	—	—	—
	S_{s1}	—	—	—	0	1,310	—

Table 3. Summary of Stresses Due to Normal Operating Loads.

Load Description	Stress Designation	Normal Stress (psi)			Shear Stress (psi)		
		Bottom	Centerline	Top	Bottom	Centerline	Top
Longitudinal Acceleration	S_{x7}	272	272	272	—	—	—
	S_{x8}	638	0	-638	—	—	—
Longitudinal Deceleration (Trailer Braking Only)	S_{x5}	350	350	350	—	—	—
	S_{x6}	1,999	0	-1,999	—	—	—
Longitudinal Deceleration (Tractor Braking Only)	S_{x10}	-420	-420	-420	—	—	—
	S_{s11}	-987	0	987	—	—	—
Vertical Acceleration	S_{x4}	1,508	0	-1,508	—	—	—
	S_{s2}	—	—	—	0	459	0
Lateral Acceleration	S_{x9}	0	±862	0	—	—	—
	S_{s3}	—	—	—	262	0	262
	S_{s4}	—	—	—	574	574	574

Table 4. Summary of Stresses Due to Extreme Dynamic Loads.

Load Description	Stress Designation	Normal Stress (psi)			Shear Stress (psi)		
		Bottom	Centerline	Top	Bottom	Centerline	Top
Longitudinal Acceleration	S_{X7} S_{X8}	543 1,277	543 0	543 -1,277	--- ---	--- ---	--- ---
Longitudinal Deceleration (Trailer Braking Only)	S_{X5} S_{X6}	701 3,998	701 0	701 -3,998	--- ---	--- ---	--- ---
Longitudinal Deceleration (Tractor Braking Only)	S_{X10} S_{S11}	-840 -1,973	-840 0	-840 1,973	--- ---	--- ---	--- ---
Vertical Acceleration	S_{X4} S_{S2}	3,015 ---	0 ---	-3,015 ---	--- 0	--- 917	--- 0
Lateral Acceleration	S_{X9} S_{S3} S_{S4}	0 --- ---	±1,723 --- ---	0 --- ---	--- 524 1,147	--- 0 1,147	--- 524 1,147

Principal stresses for the appropriate and required combinations of loads can be computed using the stress transformation equation as illustrated on the following pages.

<u>Static Design</u> (Must comply with ASME Code)

COMB. SA:

At bottom of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x2} + S_{x3}$
$S_x = 3{,}051 + 175 + 4{,}308$
$S_x = 7{,}534$ psi
$S_y = S_{y1} + S_{y2}$
$S_y = 6{,}146 + 352$
$S_y = 6{,}498$ psi
$S_s = 0$ psi
$S = 7{,}534$ psi (tension)

At top of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x3}$
$S_x = 3{,}051 - 4{,}308$
$S_x = -1{,}257$ psi
$S_y = 6{,}146$ psi
$S_s = 0$ psi
$S = 6{,}147$ psi (tension)
$S = -1{,}257$ psi (compression)

At top of tank at maximum moment without internal pressure:

$S_x = S_{x3}$
$S_x = -4{,}308$ psi
$S_y = 0$ psi
$S_s = 0$ psi
$S = -4{,}308$ psi (compression)

Normal Operating Loadings

COMB. NA: (Usual loads with trailer braking)

At bottom of tank at maximum moment:

$S_x = S_{x1} + S_{x2} + S_{x3} + S_{x4} + S_{x5} + S_{x6}$
$S_x = 3{,}051 + 175 + 4{,}308 + 1{,}508 + 350 + 1{,}999$
$S_x = 11{,}391$ psi
$S_y = S_{y1} + S_{y2}$
$S_y = 6{,}146 + 352$
$S_y = 6{,}498$ psi
$S_s = S_{s4}$
$S_s = 574$ psi

$S = 0.5(S_x+s_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$
$S = 0.5(11{,}391+6{,}498) \pm (0.25(11{,}391-6{,}498)^2+574^2)^{0.5}$
$S = 8{,}945 \pm 2{,}513$
$S = 11{,}458$ psi (tension)

At top of tank at maximum moment without internal pressure:

$S_x = S_{x2} + S_{x3} + S_{x4} + S_{x5} + S_{x6}$
$S_x = 0 -4{,}308 -1{,}508 + 350 -1{,}999$
$S_x = -7{,}465$ psi
$S_y = 0$ psi
$S_s = S_{s4}$
$S_s = 574$ psi

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$
$S = 0.5(-7{,}465) \pm (0.25(-7{,}465)^2+574^2)^{0.5}$
$S = -3{,}733 \pm 3{,}776$
$S = -7{,}509$ psi (compression)

COMB. NB: (Usual loads with tractor accelerating)

At bottom of tank at maximum moment:

$S_x = S_{x1} + S_{x2} + S_{x3} + S_{x4} + S_{x7} + S_{x8}$
$S_x = 3{,}051 + 175 + 4{,}308 + 1{,}508 + 272 + 638$
$S_x = 9{,}952$ psi
$S_y = S_{y1} + S_{y2}$

$S_y = 6,146 + 352$
$S_y = 6,498$ psi
$S_s = S_{s4}$
$S_s = 574$ psi

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$
$S = 0.5(9,952+6,498) \pm (0.25(9,952-6,498)^2+574^2)^{0.5}$
$S = 8,225 \pm 1,820$
$S = 10,045$ psi (tension)

At top of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x3} + S_{x4} + S_{x7} + S_{x8}$
$S_x = 3,051 -4,308 -1,508 + 272 -638$
$S_x = -3,131$ psi
$S_y = S_{y1}$
$S_y = 6,146$
$S_s = S_{s4}$
$S_s = 574$

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$
$S = 0.5(-3,131+6,146) \pm (0.25(-3,131-6,146)^2+574^2)^{0.5}$
$S = 1,508 \pm 4,674$
$S = -3,166$ psi (compression)
$S = 6,182$ psi (tension)

At top of tank at maximum moment without internal pressure:

$S_x = S_{x3} + S_{x4} + S_{x7} + S_{x8}$
$S_x = -4,308 -1,508 + 272 -638$
$S_x = -6,182$ psi
$S_y = 0$ psi
$S_s = S_{s4}$
$S_s = 574$ psi

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$
$S = 0.5(-6,182) \pm (0.25(-6,182)^2+574^2)^{0.5}$
$S = -3,091 \pm 3,144$
$S = -6,235$ psi (compression)

COMB. NC: (Usual loads with tractor braking)

At bottom of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x2} + S_{x3} + S_{x4} + S_{x10} + S_{x11}$
$S_x = 3{,}051 + 175 + 1{,}508 + 4{,}308 - 420 - 987$
$S_x = 7{,}635$ psi
$S_y = S_{y1} + S_{y2}$
$S_y = 6{,}146 + 352$
$S_y = 6{,}498$ psi
$S_s = S_{s4}$
$S_s = 574$

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$
$S = 0.5(7{,}635+6{,}498) \pm (0.25(7{,}635-6{,}498)^2+574^2)^{0.5}$
$S = 7{,}067 \pm 808$
$S = 7{,}875$ psi (tension)

At bottom of tank at maximum moment without internal pressure:

$S_x = S_{x2} + S_{x3} + S_{x4} + S_{x10} + S_{x11}$
$S_x = 175 + 4{,}308 + 1{,}508 - 420 - 987$
$S_x = 4{,}584$ psi
$S_y = S_{y2}$
$S_y = 352$ psi
$S_s = S_{s4}$
$S_s = 574$

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$
$S = 0.5(4{,}584+352) \pm (0.25(4{,}584-352)^2+574^2)^{0.5}$
$S = 2{,}468 \pm 2{,}192$
$S = 4{,}660$ psi (tension)

At top of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x3} + S_{x4} + S_{x10} + S_{x11}$
$S_x = 3{,}051 - 4{,}308 - 1{,}508 - 420 + 987$
$S_x = -2{,}198$ psi
$S_y = S_{y1}$
$S_y = 6{,}146$ psi
$S_s = S_4$
$S_s = 574$

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$

$$S = 0.5(-2{,}198+6{,}146) \pm (0.25(-2{,}198-6{,}146)^2+574^2)^{0.5}$$
$$S = 1{,}974 \pm 4{,}211$$
$$S = 6{,}185 \text{ psi (tension)}$$
$$S = 2{,}237 \text{ psi (compression)}$$

At top of tank at maximum moment without internal pressure:

$$S_x = S_{x3} + S_{x4} + S_{x10} + S_{x11}$$
$$S_x = -4{,}308 - 1{,}508 - 420 + 987$$
$$S_x = -5{,}249 \text{ psi}$$
$$S_y = 0 \text{ psi}$$
$$S_s = S_{s4}$$
$$S_s = 574$$
$$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$$
$$S = 0.5(-5{,}249+0) \pm (0.25(-5{,}249-0)^2+574^2)^{0.5}$$
$$S = -2{,}625 \pm 2{,}686$$
$$S = -5{,}311 \text{ psi (compression)}$$

Extreme Dynamic Loadings

COMB. EA: (Static loads with trailer braking)

At bottom of tank at maximum moment:

$$S_x = S_{x1} + S_{x2} + S_{x3} + S_{x5} + S_{x6}$$
$$S_x = 3{,}051 + 175 + 4{,}308 + 701 + 3{,}998$$
$$S_x = 12{,}233 \text{ psi}$$
$$S_y = S_{y1} + S_{y2} = 6{,}146 + 352$$
$$S_y = 6{,}498 \text{ psi}$$
$$S_s = 0 \text{ psi}$$
$$S = 12{,}233 \text{ psi (tension)}$$

At top of tank at maximum moment with internal pressure:

$$S_x = S_{x1} + S_{x2} + S_{x3} + S_{x5} + S_{x6}$$
$$S_x = 3{,}051 + 175 - 4{,}308 + 701 + 3{,}998$$
$$S_x = 3{,}617 \text{ psi}$$
$$S_y = S_{y1}$$
$$S_y = 6{,}146 \text{ psi}$$

S_s = 0 psi

S = 3,617 psi (tension)

and S = 6,146 psi (tension)

At top of tank at maximum moment without internal pressure:

$S_x = S_{x3} + S_{x5} + S_{x6}$

S_x = -4,308 + 701 -3,998

S_x = -7,605 psi

S_y = 0 psi

S_s = 0 psi

S = -7,605 psi (compression)

COMB. EB: (Static loads with tractor accelerating)

At bottom of tank at maximum moment:

$S_x = S_{x1} + S_{x2} + S_{x3} + S_{x7} + S_{x8}$

S_x = 3,051 + 175+ 4,308 + 543 + 1,277

S_x = 9,354 psi

$S_y = S_{y1} + S_{y2}$ = 6,146 + 352

S_y = 6,498 psi

S_s = 0 psi

S = 9,354 psi (tension)

At top of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x3} + S_{x7} + S_{x8}$

S_x = 3,051 -4,308 + 543 -1,277

S_x = -1,991 psi

$S_y = S_{y1}$

S_y = 6,146 psi

S_s = 0 psi

S = 6,146 psi (tension)

or S = -1,991 psi (compression)

At top of tank at maximum moment without internal pressure:

$S_x = S_{x3} + S_{x7} + S_{x8}$

S_x = -4,308 + 543 -1,277

S_x = -5,042 psi

S_y = 0 psi

$S_s = 0$ psi

$S = 0$ psi (tension)

or $S = -5{,}042$ psi (compression)

COMB. EC: (Static loads with tractor braking)

At bottom of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x2} + S_{x3} + S_{x10} + S_{x11}$

$S_x = 3{,}051 + 175 + 4{,}308 - 840 - 1{,}973$

$S_x = 4{,}721$ psi

$S_y = S_{y1} + S_{y2} = 6{,}146 + 352$

$S_y = 6{,}498$ psi

$S_s = 0$ psi

$S = 6{,}498$ psi (tension)

At bottom of tank at maximum moment without internal pressure:

$S_x = S_{x2} + S_{x3} + S_{x10} + S_{x11}$

$S_x = 175 + 4{,}308 - 840 - 1{,}973$

$S_x = 1{,}670$ psi

$S_y = S_{y2} = 352$ psi

$S_s = 0$ psi

$S = 1{,}670$ psi (tension)

At top of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x3} + S_{x10} + S_{x11}$

$S_x = 3{,}051 - 4{,}308 - 840 + 1{,}973$

$S_x = -124$ psi

$S_y = S_{y1}$

$S_y = 6{,}146$ psi

$S_s = 0$ psi

$S = 6{,}146$ psi (tension)

At top of tank at maximum moment without internal pressure:

$S_x = S_{x3} + S_{x10} + S_{x11}$

$S_x = -4{,}308 - 840 + 1{,}973$

$S_x = -3{,}175$ psi

$S_y = 0$ psi

$S_s = 0$ psi

$S = -3{,}175$ psi (compression)

COMB. ED: (Static loads with vertical acceleration)

At bottom of tank at maximum moment with internal presssure:

$S_x = S_{x1} + S_{x2} + S_{x3} + S_{x4}$

$S_x = 3{,}051 + 175 + 4{,}308 + 3{,}015$

$S_x = 10{,}549$ psi

$S_y = S_{y1} + S_{y2} = 6{,}146 + 352$

$S_y = 6{,}498$ psi

$S_s = 0$ psi

$S = 10{,}549$ psi (tension)

At top of tank at maximum moment without internal pressure:

$S_x = S_{x3} + S_{x4}$

$S_x = -4{,}308 - 3{,}015$

$S_x = -7{,}323$ psi

$S_y = 0$ psi

$S_s = 0$ psi

$S = -7{,}323$ psi (compression)

COMB. EE: (Static loads with lateral acceleration)

At bottom of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x2} + S_{x3}$

$S_x = 3{,}051 + 175 + 4{,}308$

$S_x = 7{,}534$ psi

$S_y = S_{y1} + S_{y2} = 6{,}146 + 352$

$S_y = 6{,}498$ psi

$S_s = S_{s4}$

$S_s = 1{,}147$ psi

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$

$S = 0.5(7{,}534+6{,}498) \pm (0.25(7{,}534-6{,}498)^2+1{,}147^2)^{0.5}$

$S = 7{,}016 \pm 1{,}259$

$S = 8{,}275$ psi (tension)

At top of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x3}$

$S_x = 3{,}051 - 4{,}308$
$S_x = -1{,}257$ psi
$S_y = S_{y1}$
$S_y = 6{,}146$ psi
$S_s = S_{s4}$
$S_s = 1{,}147$

$$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$$
$$S = 0.5(-1{,}257+6{,}146) \pm (0.25(-1{,}257-6{,}146)^2+1{,}147^2)^{0.5}$$
$$S = 2{,}445 \pm 3{,}875$$
$$S = 6{,}320 \text{ psi (tension)}$$
or $$S = -1{,}431 \text{ psi (compression)}$$

At tension side of tank at maximum moment with internal pressure:

$S_x = S_{x1} + S_{x2} + S_{x9}$
$S_x = 3{,}051 + 87 + 1{,}723$
$S_x = 4{,}861$ psi
$S_y = S_{y1} + S_{y2} = 6{,}146 + 176$
$S_y = 6{,}322$ psi
$S_s = S_{s4}$
$S_s = 1{,}147$ psi

$$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$$
$$S = 0.5(4{,}861+6{,}322) \pm (0.25(4{,}861-6{,}322)^2+1{,}147^2)^{0.5}$$
$$S = 5{,}591 \pm 1{,}360$$
$$S = 6{,}951 \text{ psi (tension)}$$

At compression side of tank at maximum lateral moment with internal pressure:

$S_x = S_{x1} + S_{x2} + S_{x9}$
$S_x = 3{,}051 + 87 - 1{,}723$
$S_x = 1{,}415$ psi
$S_y = S_{y1} + S_{y2} = 6{,}146 + 176$
$S_y = 6{,}322$ psi
$S_s = S_{s4}$
$S_s = 1{,}147$ psi

$$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$$
$$S = 0.5(1{,}415+6{,}322) \pm (0.25(1{,}415-6{,}322)^2+1{,}147^2)^{0.5}$$
$$S = 3{,}869 \pm 2{,}708$$
$$S = 6{,}577 \text{ psi (tension)}$$

At compression side of tank at maximum lateral moment without internal pressure:

$S_x = S_{x2} + S_{x9}$

$S_x = 87 - 1{,}723$

$S_x = -1{,}636$ psi

$S_y = S_{y2} = 176$ psi

$S_s = S_{s4}$

$S_s = 1{,}147$ psi

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$

$S = 0.5(-1{,}636+176) \pm (0.25(-1{,}636-176)^2+1{,}147^2)^{0.5}$

$S = -730 \pm 1{,}462$

$S = -2{,}192$ psi (compression)

At bottom of tank adjacent to F_u with internal pressure:

$S_x = S_{x1} + S_{x2}$

$S_x = 3{,}051 + 175$

$S_x = 3{,}226$ psi

$S_y = S_{y1} + S_{y2} = 6{,}146 + 352$

$S_y = 6{,}498$ psi

$S_s = S_{s3} + S_{s4}$

$S_s = 524 + 1{,}147$

$S_s = 1{,}671$ psi

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$

$S = 0.5(3{,}226+6{,}498) \pm (0.25(3{,}226-6{,}498)^2+1{,}671^2)^{0.5}$

$S = 4{,}862 \pm 2{,}339$

$S = 7{,}201$ psi (tension)

At top of tank adjacent to F_u with internal pressure:

$S_x = S_{x1}$

$S_x = 3{,}051$ psi

$S_y = S_{y1}$

$S_y = 6{,}146$ psi

$S_s = S_{s3} + S_{s4}$

$S_s = 524 - 1{,}147$

$S_s = -623$ psi

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$

$S = 0.5(3{,}051+6{,}146) \pm (0.25(3{,}051-6{,}146)^2+623^2)^{0.5}$

$S = 4{,}599 \pm 1{,}668$

$S = 6{,}267$ psi (tension)

At side of tank adjacent to F_u with internal pressure:

$S_x = S_{x1} + S_{x2}$

$S_x = 3{,}051 + 87$

$S_x = 3{,}138$ psi

$S_y = S_{y1} + S_{y2} = 6{,}146 + 176$

$S_y = 6{,}322$ psi

$S_s = S_{s1} + S_{s4}$

$S_s = 1{,}310 + 1{,}147$

$S_s = 2{,}457$ psi

$S = 0.5(S_x+S_y) \pm (0.25(S_x-S_y)^2+S_s^2)^{0.5}$

$S = 0.5(3{,}138+6{,}322) \pm (0.25(3{,}138-6{,}322)^2+2{,}457^2)^{0.5}$

$S = 4{,}730 \pm 2{,}928$

$S = 7{,}658$ psi (tension)

CHAPTER 6. BOTTOM DAMAGE PROTECTION

Specifications for bottom damage protection are contained in §178.345-8(b). If bottom damage protection devices are used they must be able to withstand or deflect away from the cargo tank a force of 155,000 pounds (based on the ultimate strength of the material) from the front, side, or rear. Additional details are provided in the DOT Regulations.

The analysis procedure(s) to be employed to evaluate bottom damage protection devices is dependent upon the structure of the device. The device should be idealized appropriately and appropriate structural analysis procedures should be employed.

If an outlet, projection or piping is located in the lower 1/3 of the tank at the rear of the tank and it is protected by the rear-end protection device, the rear-end protection device must meet bottom damage protection requirements in addition to rear-end protection requirements.

CHAPTER 7. ROLLOVER DAMAGE PROTECTION

Criteria for DOT 406, DOT 407, DOT 412 cargo tank motor vehicles are included in §178.345-8(c). This section requires that the guards

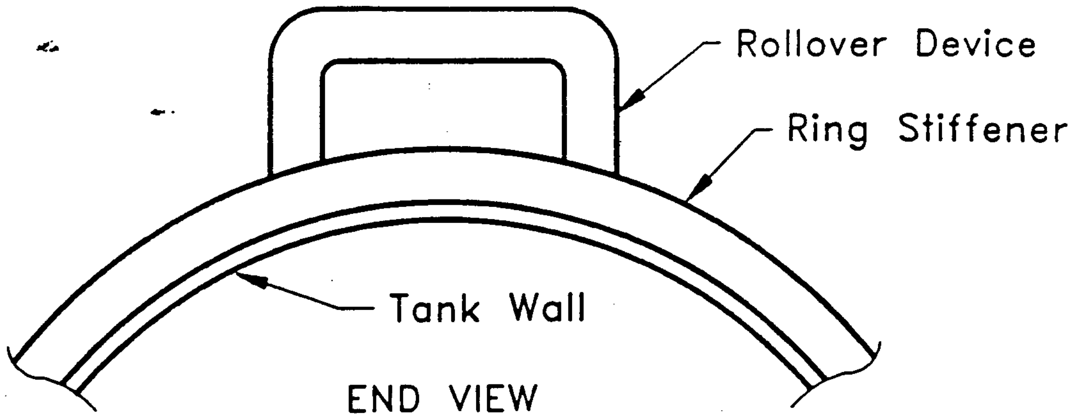
> "...must be designed and installed to withstand loads equal to twice the weight of the loaded cargo tank motor vehicle applied as follows: normal to the tank shell (perpendicular to the tank surface); and tangential (perpendicular to the normal load) from any direction. The stresses shall not exceed the ultimate strength of the material of construction. These design loads may be considered to be uniformly distributed and independently applied. If more than one rollover protection device is used, each device must be capable of carrying its proportionate share of the required loads and in each case at least one-fourth the total tangential load. The design must be proven capable of carrying the required loads by calculations, tests or a combination of tests and calculations."

Analysis Procedures

One common type of rollover damage protection device is an inverted "U" shaped member made of tubular elements as shown in figure 29. The devices are frequently installed at a stiffened cross section of the tank. In some designs, gusset plates are used between the tank wall and the rollover device to increase capacity to carry horizontal longitudinal load. Some designs have used a third leg to increase capacity.

The two-legged inverted "U" devices behave as a frame when subjected to vertical load or horizontal transverse load. Such frames can be analyzed using moment distribution procedures, frame analysis equations from Roarke and Young, approximate methods for frames, or finite element methods. For horizontal longitudinal load, these two-legged devices can be idealized as cantilever beams.

A difficulty, common to all of the suggested methods of analysis, is idealization of the strength and stiffness of the tank at its juncture with the rollover device.



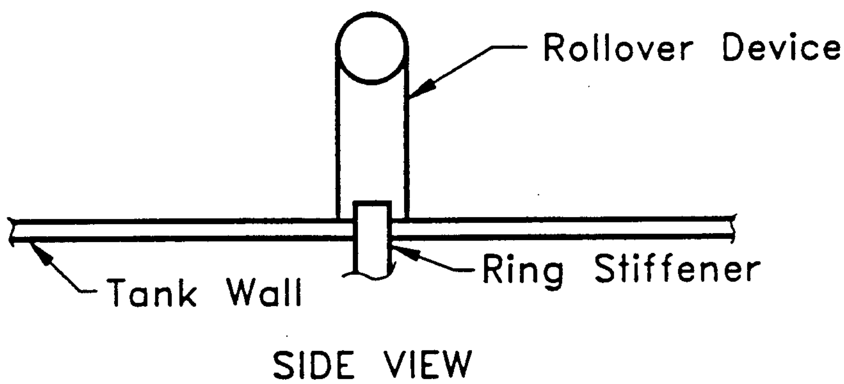


Figure 29. Inverted "U" rollover protection device.

Example 24. Analysis of inverted "U" using moment distribution.

The two-legged inverted "U" device shown in figure 30 is one of three devices used on a DOT 400 Series cargo tank motor vehicle whose total weight is 49,700 lbs. The device is made of 3-inch standard pipe with an ultimate strength of 65.2 ksi. Properties of the pipe cross section are given in figure 30. Each device must withstand a load of 2W/n which is 2(49,700)/3 = 33.13k.

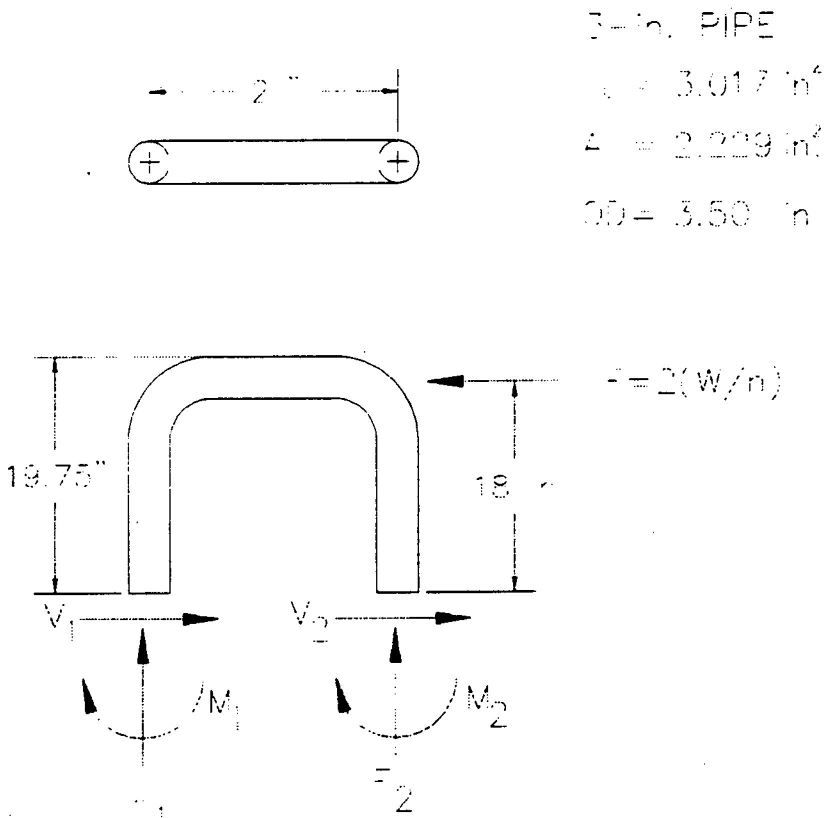


Figure 30. Two-legged, inverted "U" rollover device for DOT 400 Series cargo tank.

For the first analysis, it is assumed that the tank wall provides rigid support for the overturn device at the points of attachment. This is a reasonable assumption but is not accurate because the tank walls are flexible.

A moment distribution analysis table for "side sway" with assumed fixed-end moments of +100 in-k is presented in figure 32. The results show final joint moments of +77.9 in-k at joints A and D and 56.4 in-k at joints B and C. Further analysis to impose static equilibrium for the horizontal force of 33.13 kips is required.

For member AB:

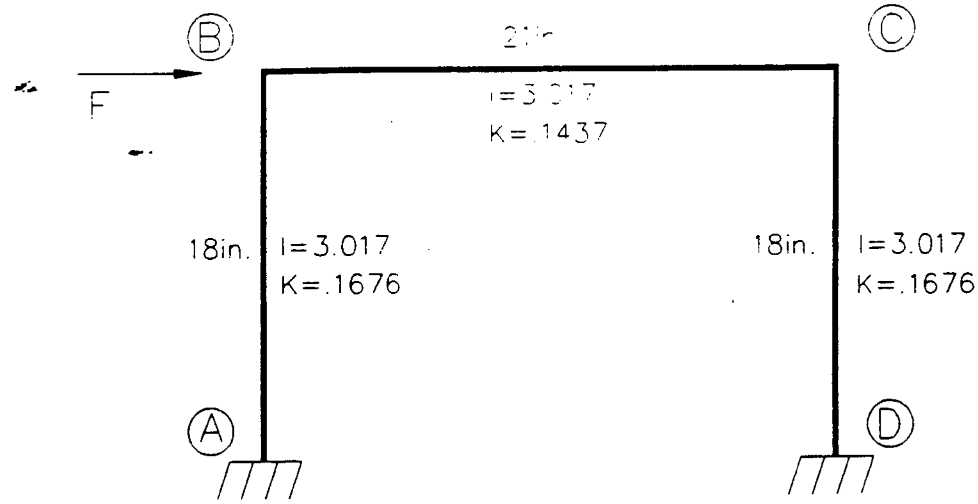
$\sum M_B=0;\ -H_A(18)+77.9+56.4 = 0$

$H_A = (77.9+56.4)/18$

$H_A = 7.46k$

also

$H_D = 7.46k$



Joint	A	B		C		D
Member	AB	BA	BC	CB	CD	DC
K	.1676	.1676	.1437	.1437	.1676	.1676
Dist. F.	0	.538	.462	.462	.538	0
F.E.M.	+100	+100	0	0	+100	+100
1st Dist	0	-53.8	-46.2	-46.2	-53.8	0
CO	-26.9	0	-23.1	-23.1	0	-26.9
2nd Dist	0	+12.4	+10.7	+10.7	+12.4	0
CO	+6.2	0	+5.3	+5.3	0	+6.2
3rd Dist	0	-2.8	-2.5	-2.5	-2.8	0
CO	-1.4	0	-1.2	-1.2	0	-1.4
4th Dist	0	+.6	+.6	+.6	+.6	0
Σ	+77.9	+56.4	-56.4	-56.4	+56.4	+77.9

Figure 31. Moment distribution analysis of rollover device with horizontal transverse load and bottom ends of vertical members fixed.

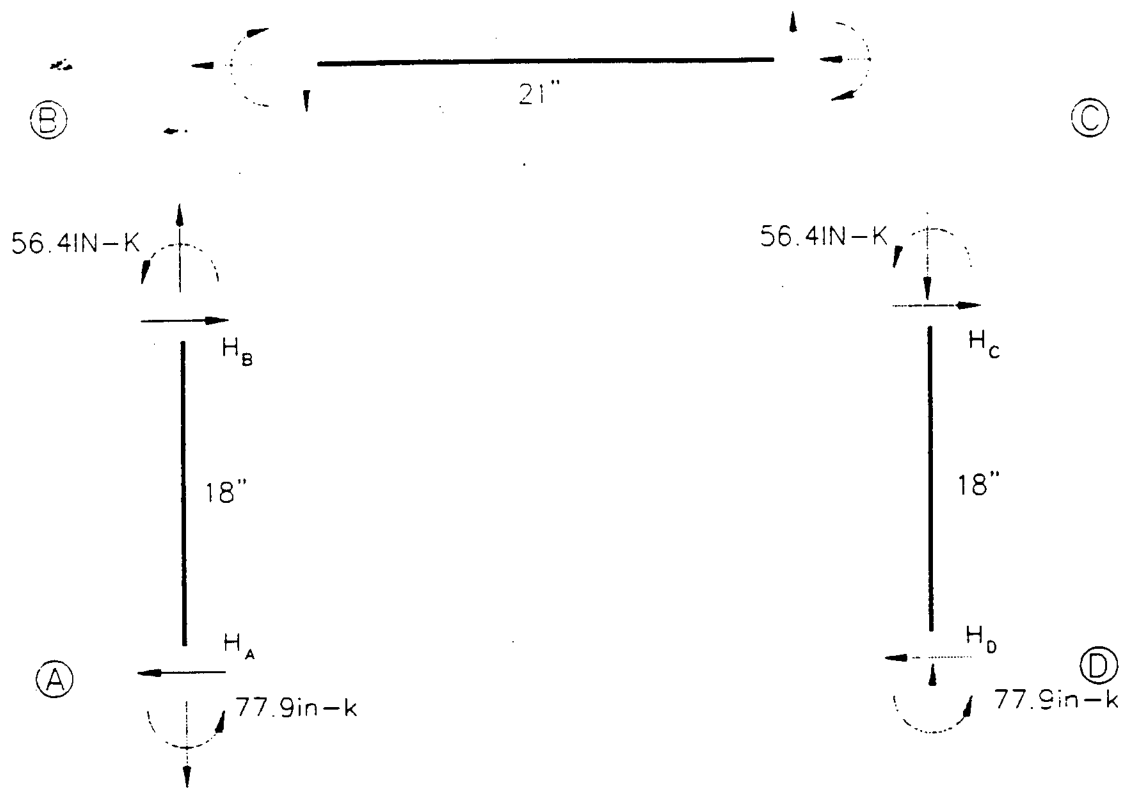


Figure 32. Preliminary results of moment distribution analysis with assumed fixed end moments.

For entire structure:

$\sum F_x=0$; F-7.46-7.46=0

F = 14.92k

The adjustment factor that must be applied to the results in figure 31 is:

33.13/14.92 = 2.22

All moments (and forces) for the solution presented in figure 31 must be multiplied by 2.22 for an actual load of 33.13 kips. The final adjusted moments at joints A and D are each 173 in-k as shown in figure 33.

Further, static analysis results in axial forces of 11.9 kips and shear forces of 16.56 kips in each of the vertical members.

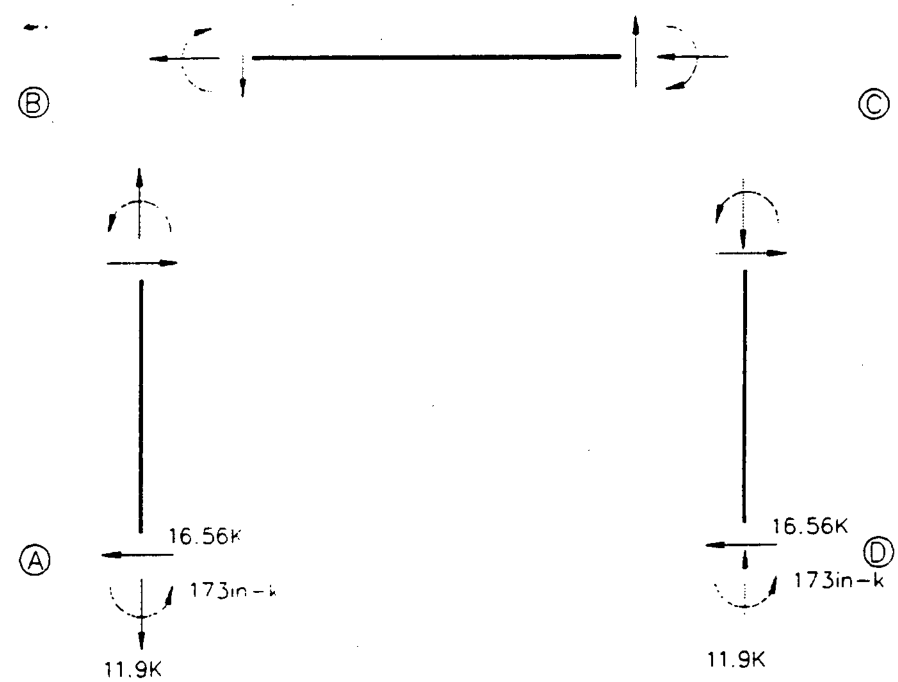


Figure 33. Final results of moment distribution analysis of structure in figure 31.

The maximum axial stress in member A-B, near joint A, is:

$$\text{Max } S = P_A/A + M/Z_e$$
$$\text{Max } S = 11.9/2.229 + 173(1.75)/3.017$$
$$\text{Max } S = 5.3 + 100.3$$
$$\text{Max } S = 105 \text{ ksi} > 65.2 \text{ ksi} \quad \text{(No good)}$$

The axial stress in the horizontal member near joints B and C would be:

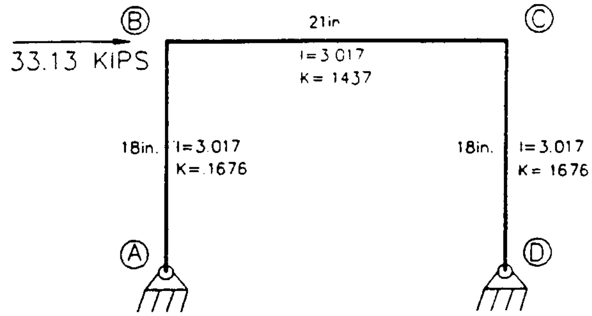
$$S = F/A + M/Z_e$$
$$M = 56.4(2.22)$$
$$M = 125.2$$
$$S = 16.56/2.229 + 125.2(1.75)/3.017$$
$$S = 7.4 + 72.6$$
$$S = 80.0 \text{ ksi}$$

The axial stress in the vertical members near joints B and C would be:

$$S = F/A + M/Z_e$$
$$S = 11.9/2.229 + 125.2(1.75)/3.017$$
$$S = 5.3 + 72.6$$
$$S = 77.9 \text{ ksi} > 65.2 \text{ ksi (No good)}$$

An analysis of the two leg-device for horizontal transverse load using the Moment Distribution Method and the assumption that the tank shell provides no stiffness in bending in the legs of the device is presented in figure 34. The final adjusted moments are shown in figure 35. The computed maximum tensile (or compressive) stresses in the vertical legs of the device occur near joints B and C and is 186 ksi.

If the effective stiffness of the tank wall were known and could be included in the analysis, the results would be expected to be somewhere between the two Moment Distribution solutions presented herein.



Joint	A	B		C		D
Member	AB	BA	BC	CB	CD	DC
K	.1676	.1676	.1437	.1437	.1676	.1676
Dist. F	1	.538	.462	.462	.538	1
FEM	+100	+100	0	0	+100	+100
1st Dist	-100	-53.8	-46.2	-46.2	-53.8	-100
CO	-26.9	-50	-23.1	-23.1	-50	-26.9
2nd Dist	+26.9	+39.3	+33.8	+33.8	+36.3	+26.9
CO	+19.6	+13.4	+16.9	+16.9	+13.4	+19.6
3rd Dist	-19.6	-16.3	-14.0	-14.0	-16.3	-19.6
CO	-8.1	-9.8	-7.0	-7.0	-9.8	-8.1
4th Dist	+8.1	-9.0	+7.8	+7.8	+9.0	+8.1
CO	4.5	4.0	3.9	3.9	4.0	4.5
5th Dist	-4.5	-4.2	-3.6	-3.6	-4.2	-4.5
CO	-2.1	-2.2	-1.8	-1.8	-2.2	-2.1
6th Dist	+2.1	+2.1	+1.8	+1.8	+2.1	+2.1
Σ	0	31.5	-31.5	-31.5	31.5	0

Figure 34. Moment distribution analysis of rollover device with horizontal transverse load and bottom ends of vertical members hinged.

It is noted that for each of the two solutions by Moment Distribution presented on the previous pages, the sum of moments for the two ends of a vertical member of the device is 298 in-k. For the first solution (with the ends of the legs fixed) the moments are 173 in-k at the bottom and 125 in-k at the top. These total 298 in-k. For the second solution (with the ends of the legs pinned) the moments are zero at the bottom and 298 in-k at the top.

For a more accurate solution wherein partial fixity of the tank wall would be accurately included, the moments would also total 298 in-k. The degree of fixity that would result in the lowest maximum stresses in the legs of the device would be one that caused the moments to be equal at each end of the legs. (i.e. 298 ÷ 2 = 149 in-k) In this case, the stresses in each end of the legs would be:

$$S = F/A + M/Z_e$$
$$S = F/A + 149(1.75)/3.017$$
$$S = F/A + 86 \text{ ksi}$$

F/A would be somewhere between 5.3 and 12.7 ksi which would make the maximum stress over 90 ksi.

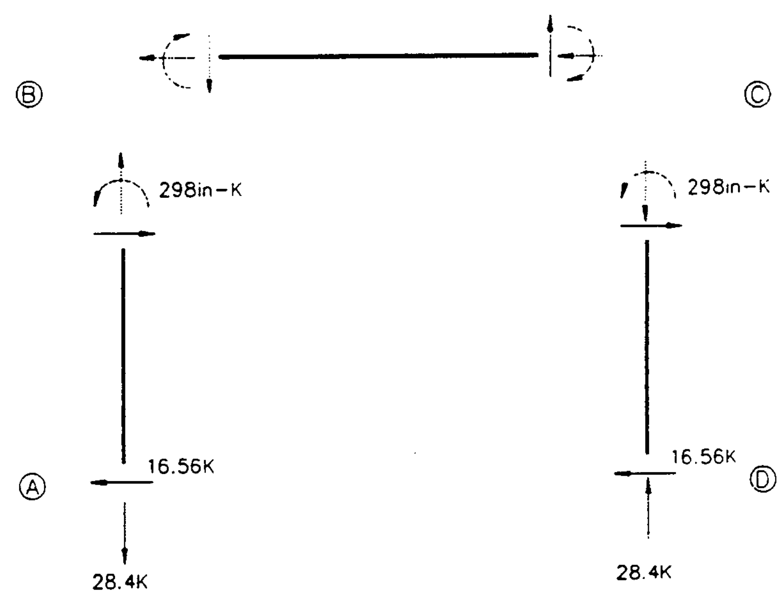


Figure 35. Results of moment distribution analysis of structure in figure 34.

Roark and Young Analysis

Equations for structural analysis of single bent frames are given in Roark and Young for various combinations of member sizes, member lengths and support conditions. For a frame subjected to a concentrated side load and with fixed supports as shown in figure 36, the following six general equations for coefficients are given:

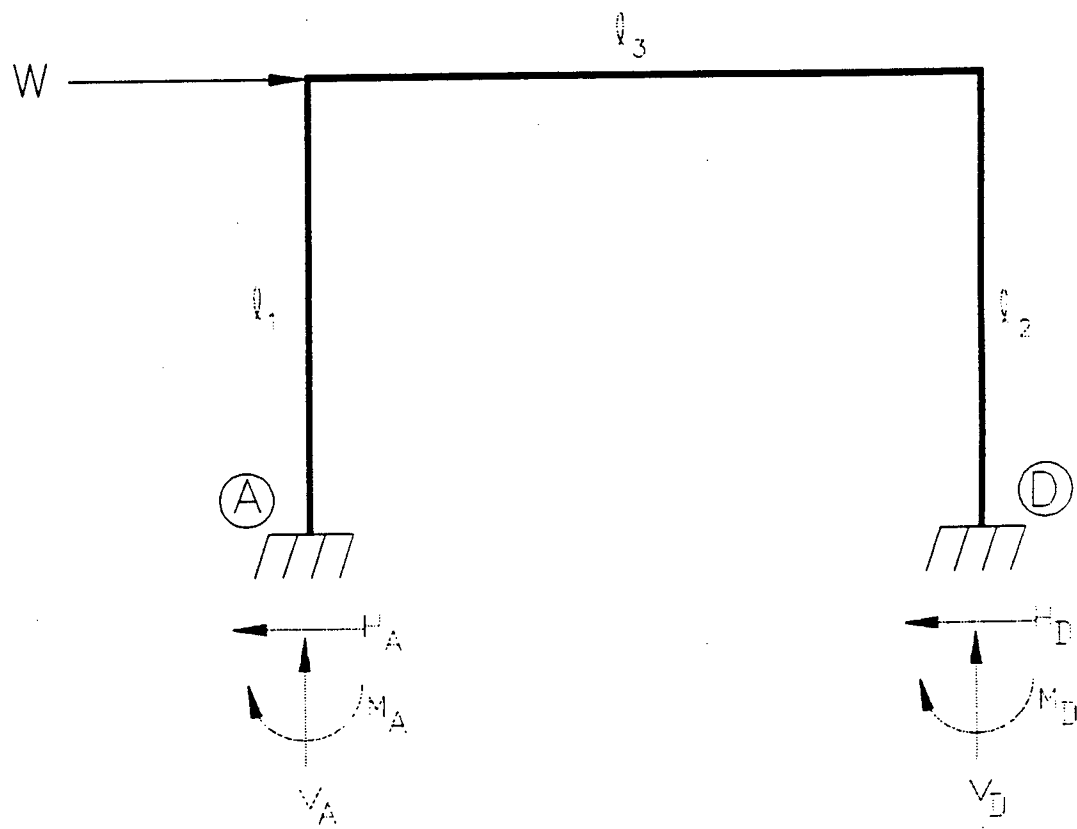


Figure 36. Idealization of rollover protection device for analysis using formulas from Roark and Young.

$$C_{HH} = \ell_1^3/3E_1I_1 + [\ell_1^3-(\ell_1-\ell_2)^3]3E_2I_2 + \ell_1^2\ell_3/E_3I_3$$

$$C_{HV} = C_{VH} = \ell_2\ell_3(2\ell_1-\ell_2)/2E_2I_2 + \ell_1\ell_3^2/2E_3I_3$$

$$C_{HM} = c_{MH} = \ell_1^2/2E_1I_1 + \ell_2(2\ell_1-\ell_2)/2E_2I_2 + \ell_1\ell_3/E_3I_3$$

$$C_{VV} = \ell_2\ell_3^2/E_2I_2 + \ell_3^3/3E_3I_3$$

$$C_{VM} = C_{MV} = \ell_2\ell_3/E_2I_2 + \ell_3^2/2E_3I_3$$

$$C_{MM} = \ell_1/E_1I_1 + \ell_2/E_2I_2 + \ell_3/E_3I_3$$

For the frame shown in figure 36, the equations reduce to:

$$C_{HH} = 2\ell_1^3/3EI + \ell_1^2\ell_3/EI$$

$$C_{HV} = \ell_1^2\ell_3/2EI + \ell_1\ell_3^2/2EI$$

$$C_{HM} = \ell_1^2/EI + \ell_1\ell_3/EI$$

$$C_{VV} = \ell_2\ell_3^2/EI + \ell_3^3/3EI$$

$$C_{VM} = \ell_2\ell_3/EI + \ell_3^2/2EI$$

$$C_{MM} = \ell_1/EI + \ell_2/EI + \ell_3/EI$$

Factors for loads are computed using the three following equations:

$$LF_H = W(C_{HH} - \ell_1C_{HM} + \ell_1^3/6E_1I_1)$$

$$LF_V = W(C_{VH} - \ell_1C_{VM})$$

$$LF_M = W(C_{MH} - \ell_1C_{MM} + \ell_1^2/2E_1I_1)$$

Coefficients and factors for loads can then be used in the following equations to solve for reactions at the left support. Reactions at the right support can then be evaluated using equations of static equilibrium.

$$C_{HH}H_A + C_{HV}V_A + C_{HM}M_A = LF_H$$

$$C_{VH}H_A + C_{VV}V_A + C_{VM}M_A = LF_V$$

$$C_{MH}H_A + C_{MV}V_A + C_{MM}M_A = LF_M$$

Example 25. Structural analysis of frame using equations from Roark and Young.

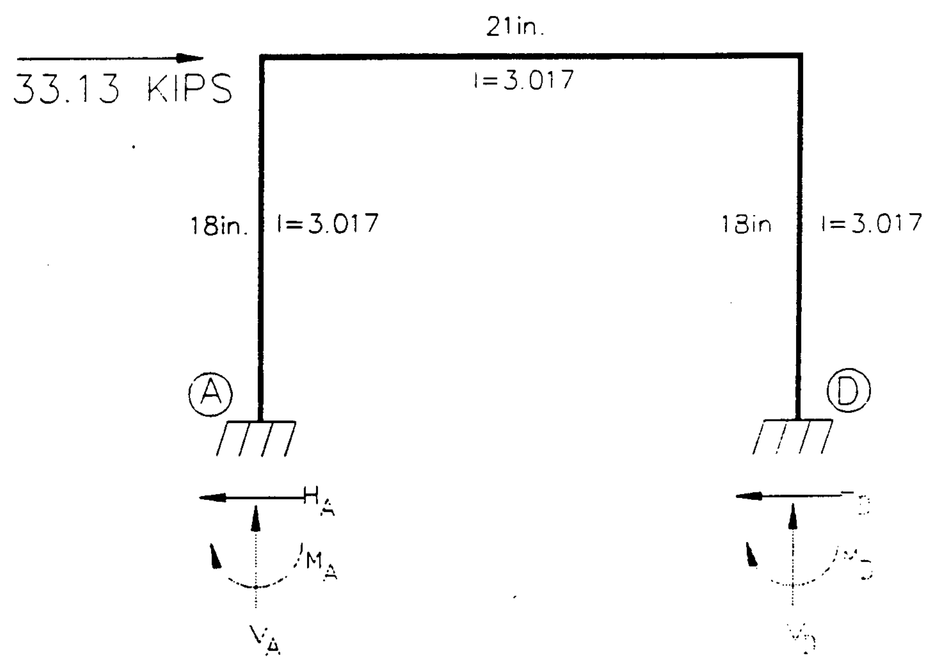


Figure 37. Idealization of frame for analysis using formulas from Roark and Young.

Use $E = 29x10^6$ psi
Then $EI = 29x10^6(3.017)$
$EI = 87.5x10^6$

$C_{HH} = 2\ell_1^3/3EI + \ell_1^2\ell_3/EI$

$C_{HH} = 2(18)^3/(3)(87.5x10^6) + (18)^2(21)/(87.5x10^6)$

$C_{HH} = 122.2x10^{-6}$

$C_{HV} = \ell_1^2\ell_3/2EI + \ell_1\ell_3^2/2EI$

$C_{HV} = (18)^2(21)/2EI + 18(21)^2/2EI$

$C_{HV} = 84.24x10^{-6}$

$C_{HM} = \ell_1^2/EI + \ell_1\ell_3/EI$

$C_{HM} = (18)^2/(87.5x10^6) + 18(21)/(87.5x10^6)$

$C_{HM} = 8.023x10^{-6}$

$C_{VV} = \ell_2\ell_3^2/EI + \ell_3^3/3EI$

$C_{VV} = (18)(21)^2/87.5x10^6 + (21)^3/3(87.5x10^6)$

$C_{VV} = 126.0x10^{-6}$

$C_{VM} = \ell_2\ell_3/EI + \ell_3^2/2EI$

$C_{VM} = (18)(21)/87.5x10^6 + (21)^2/2(87.5x10^6)$

$C_{VM} = 6.84x10^{-6}$

$C_{MM} = \ell_1/EI + \ell_2/EI + \ell_3/EI$

$C_{MM} = 18/87.5x10^6 + 18/87.5x10^6 + 21/87.5x10^6$

$C_{MM} = 0.651x10^{-6}$

$LF_H = W(C_{HH} - \ell_1 C_{HM} + \ell_1^3/6EI$

$LF_H = W[(122.2x10^{-6}) - 18(8.023x10^{-6}) + (18)^3/(6)(87.5x10^6)]$

$LF_H = W(-11.09x10^{-6})$

$LF_V = W(C_{VH} - \ell_1 C_{VM})$

$LF_V = W[84.24x10^{-6} - 18(6.84x10^{-6})]$

$LF_V = W(-38.88x10^{-6})$

$LF_M = W(C_{MH} - \ell_1 C_{MM} + \ell_1^2/2EI)$

$LF_M = W(8.023x10^{-6}) - 18(.651x10^{-6}) + (18)^2/(2)(87.5x10^6)$

$LF_M = W(-1.847x10^{-6})$

$C_{HH}H_A + C_{HV}V_A + C_{HM}M_A = LF_H$

$122.2x10^{-6}H_A + 84.24x10^{-6}V_A + 8.023x10^{-6}M_A = -11.09x10^{-6}W$

$C_{HV}H_A + C_{VV}V_A + C_{VM}M_A = LF_V$

$84.24x10^{-6}H_A + 126.0x10^{-6}V_A + 6.84x10^{-6}M_A = -38.88x10^{-6}W$

$C_{MH}H_A + C_{MV}V_A + C_{MM}M_A = LF_M$

$8.023x10^{-6}H_A + 6.84x10^{-6}V_A + .651x10^{-6}M_A = -1.847x10^{-6}W$

These equations are solved simultaneously to result in:

$H_A = +16.6k$
$V_A = -11.9k$
$M_A = -173$ in-k

The combined stress adjacent to joint A due to axial force and bending moment is:

Max $S = V_A/A + M_A/Z_e$
Max $S = 11.9/2.229 + 173/1.724$

Max S = 5,34+100.34
Max S = 105 ksi

Approximate Methods.

Approximate methods of structural analysis can be used to analyze two-legged inverted "U" rollover protection devices. One such method is the portal method. In the portal method, it is assumed that points of inflection (zero moment) occur at mid-height of the vertical members. The horizontal shear will be the same value in each of the two vertical members and the bending moment at the base of each of the two vertical members will be the same.

The assumptions stated above make the structure statically determinate and an analysis can be completed using statics.

Example 26. Analysis of inverted "U" using portal (approximate) method.

$V_1 = V_2$
$F_1 = F_2$
$M_1 = M_2$
$F_1 = (9/21)F$
$V_1 = F/2$
$M_1 = 9V_1 = 9(F/2)$
$\max S = F_1/A + M_1/Z_e$

For the values given in figure 32, the solution would be:

$V_1 = F/2 = 33.13/2 = 16.57k$
$F_1 = (9/21)F = (9/21)(33.13) = 14.2k$
$M_1 = 9(V_1) = 9(16.57) = 149$ in-k
$\max S = F_1/A + M_1/Z_e$
$\max S = 14.2/2.229 + 149/1.724$
$\max S = 6.4 + 86.4$
$\max S = 92.8$ ksi

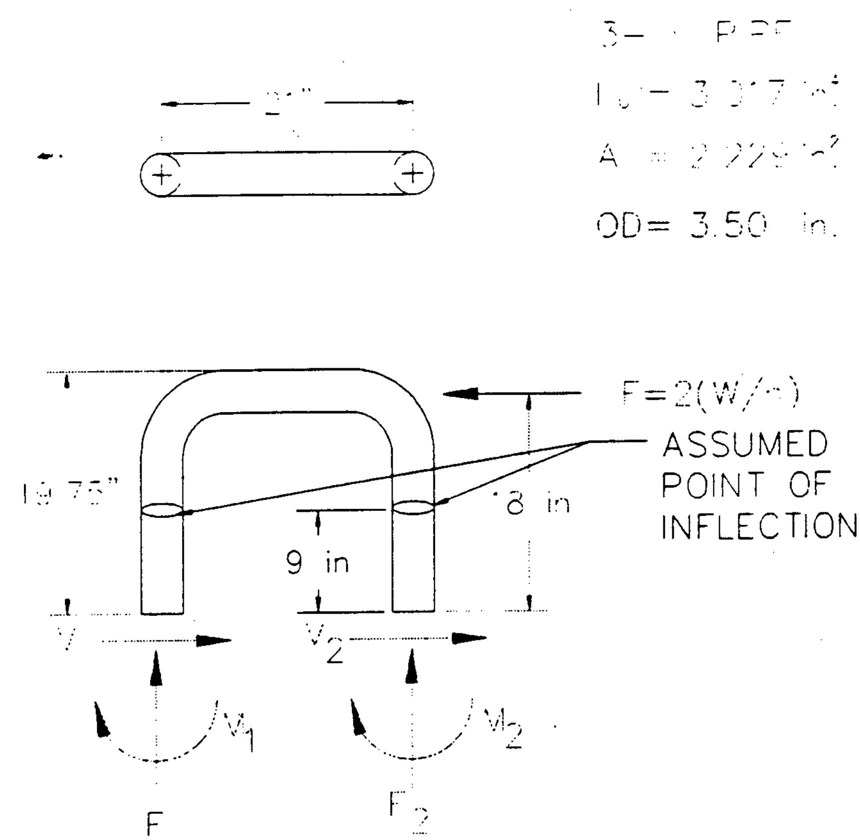


Figure 38. Idealization of rollover protection device for approximate analysis.

Example 27. Analysis of inverted "U" using finite elements.

Stresses in the rollover devices were further evaluated using a finite element analysis procedure. For these analyses, the ends of the legs of the devices were assumed fixed against rotation at the points of attachment to the tank wall.

For the two-leg device subjected to a horizontal transverse load of 33.13 kips the maximum stress is:

$$\max S = F/A + M/Z_e$$
$$\max S = 11.37/2.229 + 181.5/1.724$$
$$\max S = 5.1 + 105$$
$$\max S = 110 \text{ ksi}$$

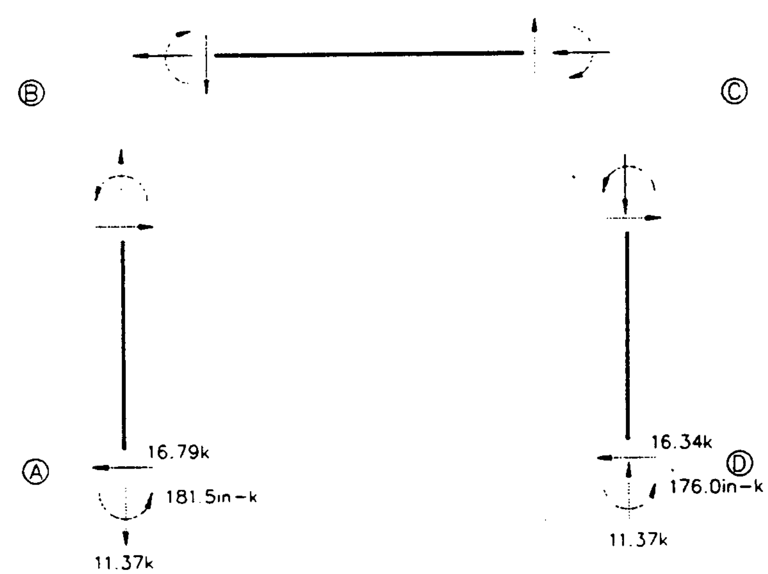
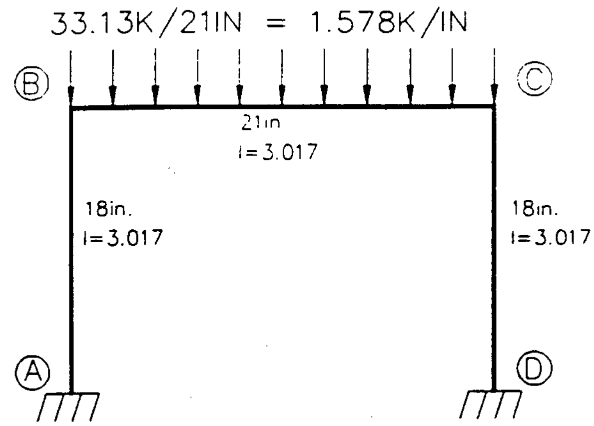


Figure 39. Results of finite element analysis of device subjected to horizontal transverse load of 33.13 kips.

Table 5. Comparison of results of various analysis procedures for inverted "U" rollover protection device subjected to horizontal transverse load.

Method of Analysis	Maximum Normal Stress, ksi	Location of Maximum Normal Stress
Moment Distribution w/ends fixed	105	Bottom end of vertical members
Moment Distribution w/ends hinged	186	Top end of vertical members
Roark and Young w/ends fixed	105	Bottom end of vertical members
Portal Method	92.8	Both ends at vertical members
Finite Element Method w/ends fixed	110	Bottom end of vertical members

A moment distribution analysis can be performed for vertical load on the device. Idealization of the structure and load, and the moment distribution table are shown in figure 40. Further static analysis gives the results shown in figure 41.



Joint	A	B		C		D
Member	AB	BA	BC	CB	CD	DC
K	.1676	.1676	.1437	.1437	.1676	.1676
Dist. F	0	.538	.462	.462	.538	0
FEM	0	0	+58	-58	0	0
1st Dist	0	-31.2	-26.8	+26.8	+31.2	0
CO	-15.6	0	+13.4	-13.4	0	+15.6
2nd Dist	0	-7.2	-6.2	+6.2	+7.2	0
CO	-3.6	0	+3.1	-3.1	0	+3.6
3rd Dist	0	-1.7	-1.4	+1.4	+1.7	0
CO	-.9	0	+.7	-.7	0	+.9
4th Dist	0	-.4	-.3	+.3	+.4	0
Σ	-20.1	-40.5	+40.5	-40.5	+40.5	+20.1

Figure 40. Moment distribution analysis of rollover device with vertical load and bottom ends of vertical members fixed.

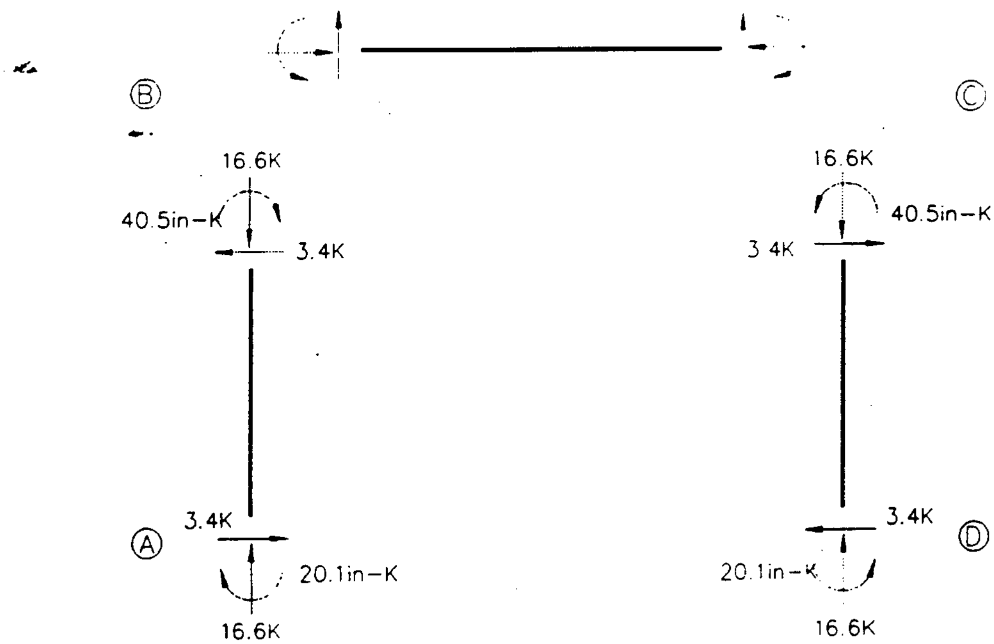


Figure 41. Results of moment distribution analysis for vertical load.

The maximum normal stresses in members A-B and C-D occur near joints B and C and are:

$$\max S = F/A + M/Z_e$$
$$\max S = 16.6/2.229 + 40.5/1.724$$
$$\max S = 7.5 + 23.5$$
$$\max S = 31.0 \text{ ksi}$$

It is noted that an approximate analysis such as S = P/A would result in:

$$S = 33.13/(2)(2.229)$$
$$S = 7.4 \text{ ksi}$$

which is much less than 31.0 ksi computed using moment distribution.

A finite element analysis of the rollover device shown in figure 40 and subjected to vertical load was performed and the results are presented in figure

42. The maximum normal stress in vertical member A-B occurs near joint B and is:

$$\max S = F/A + M/Z_e$$
$$\max S = 16.565/2.229 + 38.086/1.724$$
$$\max S = 7.43 + 22.09$$
$$\max S = 29.5 \text{ ksi}$$

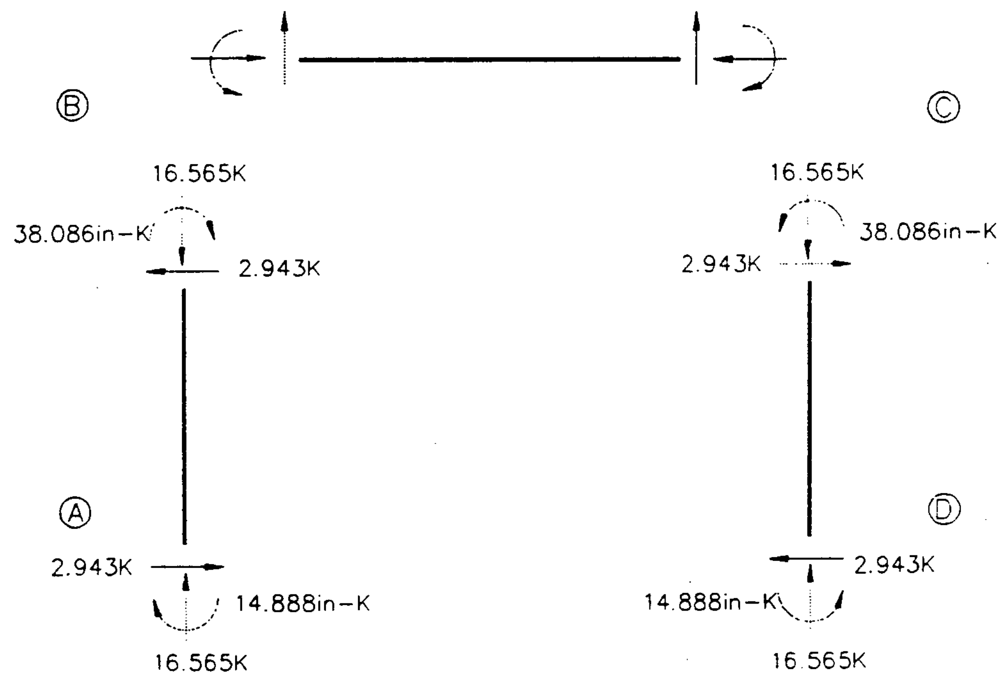


Figure 42. Results of finite element analysis for structure in figure 40.

Rectangular box-like structures fabricated from flat plates and resembling "tombstones" have been used for rollover protection devices. One such design used on a DOT 400 series trailer is shown in figure 43.

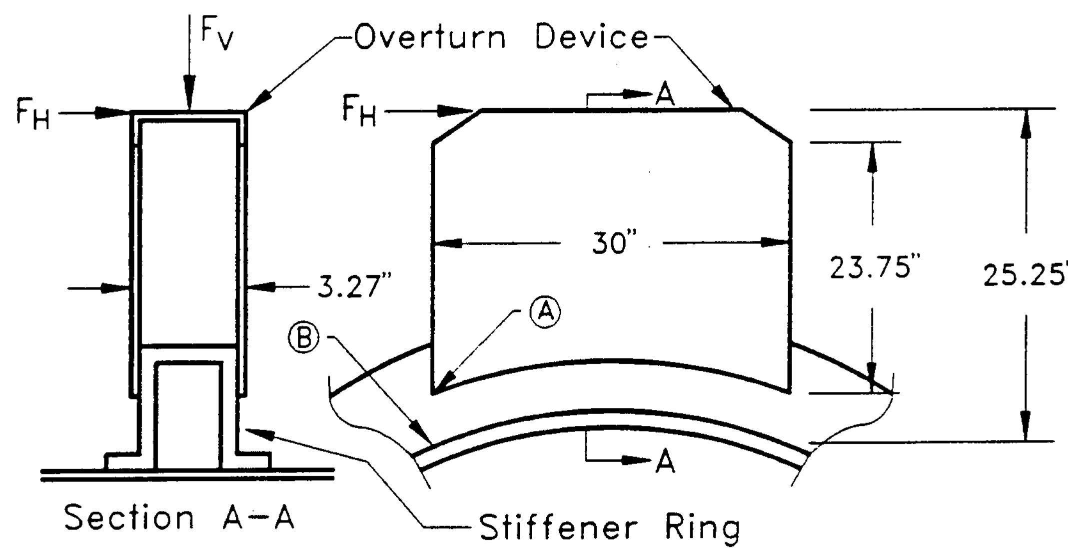


Figure 43. Box-like or "tombstone" rollover protection device.

The box is made from 10 gage (0.135 in.) Type 304 stainless steel. The device is welded to a stiffener ring made from 12 gage (0.105 in.) stainless steel. Two of these devices are used on a trailer having a total weight of 61,500 lbs. The vertical load that must be supported by the two devices is twice the total weight of the loaded tank (i.e., 2(61,500) = 123,000 lbs.). Each device must support half of this load or 61,500 lbs. The direct compressive stress in the device due to vertical load would be:

$$S = F/A = 61{,}500/8.91 = 6{,}900 \text{ psi}$$

The devices are required to support a horizontal longitudinal load and a horizontal transverse load of twice the total weight. Each device would be required to support 61,500 lbs. For horizontal load, the device will behave as a cantilever beam. The elastic section modulus for bending in the longitudinal direction is 12.55 in.^{3} and the bending stress is:

$$S = M/Z_e$$
$$S = 61{,}500(23.75)/12.55$$
$$S = 116{,}400 \text{ psi.}$$

The elastic section modulus for bending in the transverse direction is 52.5 in^3 and the bending stress is:

$$S = M/Z_e$$
$$S = 61{,}500(23.75)/52.5$$
$$S = 27{,}820 \text{ psi}$$

The average shear stress in the walls of the box when subjected to horizontal transverse load would be:

$$S_s = V/A$$
$$S_s = 61{,}500/(2)(30)(.135)$$
$$S_s = 7{,}593 \text{ psi}$$

The average shear stress in the walls of the box when subjected to horizontal longitudinal load would be:

$$S_s = V/A$$
$$S_s = 61{,}500/(2)(3.27)(.135)$$
$$S_s = 69{,}657 \text{ psi}$$

Buckling of Plates in Compression - (Roark and Young)

Elastic buckling of the thin plates used in "tombstone" devices should be checked. The compression "flange" for horizontal longitudinal bending can be checked using formulas from Table 35 in Roark and Young. Case 1 in that table is for a rectangular plate loaded with uniform compression on two opposite edges.

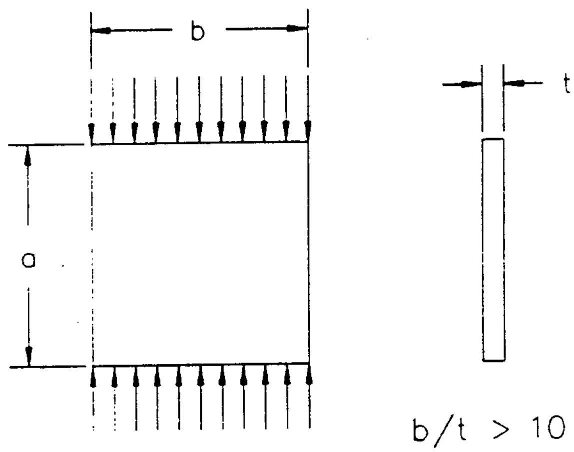


Figure 44. Rectangular plate under equal uniform compression on two opposite edges b with b/t>10.

Case 1a is for all edges of the plate simply supported and the formula for critical buckling stress is:

$$S_{cr} = [KE/(1-v^2)](t/b)^2$$

where K depends on the ratio of a/b and b is the length of each loaded edge.

a/b =	0.2	0.3	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.7	3	∞
K =	22.2	10.9	6.92	4.23	3.45	3.29	3.40	3.68	3.45	3.32	3.29	3.32	3.40	3.32	3.29	3.29

For the example at hand a/b would be 24.75/30 = 0.825, which would give a value of K = 3.43.

The critical bulking stress would be:

$$S_{cr} = [KE/1-\nu^2)](t/b)^2$$
$$S_{cr} = [(3.43)(28x10^6)/(1-.3^2)](.135/30)^2$$
$$S_{cr} = 2,137 \text{ psi}$$

For all edges of the plate clamped, the formula for critical buckling stress is the same as given above and values of K are:

a/b =	1	2	3	∞
K =	7.7	6.7	6.4	5.73

If a value of K = 7.7 is used for the device being considered the critical buckling stress would be:

$$S_{cr} = [KE/1-\nu^2)](t/b)^2$$
$$S_{cr} = [7.7(28x10^6)/(1-.3^2)](.135/30)^2$$
$$S_{cr} = 4,798 \text{ psi}$$

It is noted that both of the values computed above are extremely small in comparison with both the yield strength of the material and the computed stress due to load.

Buckling of Plates in Shear - (Roark and Young)

Case 4 in Table 35 of Roark and Young is for a rectangular plate under uniform shear on all edges.

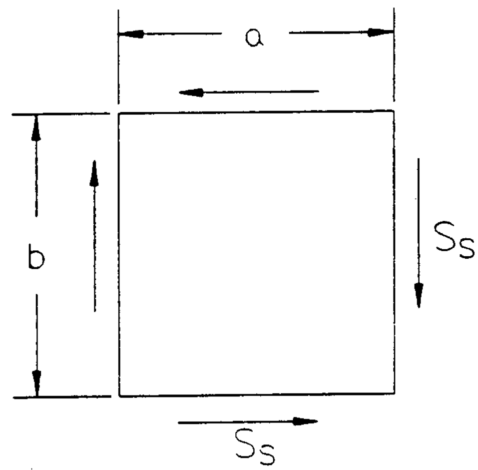


Figure 45. Rectangular plate under uniform shear on all edges.

For all edges of the plate simply supported, the critical shear stress is:

$$S_{scr} = [KE/(1-\nu^2)](t/b)^2$$

where K depends on the ratio a/b and b is the length of the shorter sides.

a/b =	1.0	1.2	1.4	1.5	1.6	1.8	2.0	2.5	3.0	∞
K =	7.75	6.58	6.00	5.84	5.76	5.59	5.43	5.18	5.02	4.40

For an a/b ratio of 30/24.75=1.21, K would be 6.55. The critical shear stress would be:

$$S_{scr} = [KE/(1-\nu^2)](t/b)^2$$
$$S_{scr} = [6.55(28\text{x}10^6)/(1-.3^2)](.135/24.75)^2$$
$$S_{scr} = 5{,}996 \text{ psi}$$

For all edges of the plate clamped, the formula for critical shear stress is the same as above but the following values of K are given:

a/b =	1	2	∞
K =	12.7	9.5	7.38

For an a/b ratio of 1.21, K would be 12.0 and the critical shear stress would be:

$$S_{scr} = [KE/(1-\nu^2)](t/b)^2$$
$$S_{scr} = [12.0(28\text{x}10^6)/(1-.3^2)](.135/24.75)^2$$
$$S_{scr} = 10{,}985 \text{ psi}$$

Buckling of Plates in Shear-(Salmon and Johnson).

Salmon and Johnson give the following formula for critical buckling shear stress for a thin rectangular plate simply supported on all edges and subjected to shear:

$$S_{scr} = [K\pi^2E]/[12(1-\nu^2)(b/t)^2]$$

where:

$K = 5.34+4.0(b/a)^2$
E = modulus of elasticity, psi
ν = poisson's ratio
t = thickness of plate, in.
b = short dimension of plate, in.
a = long dimension of plate, in.

From the previous computations, the ratio of b/a would be 0.825 and the value of K would be:

$$K = 5.34+4.0(b/a)^2$$
$$K = 5.34+4.0(.825)^2$$
$$K = 8.06$$

The critical buckling shear stress would be:

$$S_{scr} = [K\pi^2E]/[12(1-\nu^2)(b/t)^2]$$
$$S_{scr} = [8.06\pi^2(28x10^6)]/[12(1-.3^2)(24.75/.135)^2$$
$$S_{scr} = 6,068 \text{ psi}$$

This compares to 5,996 psi from the Roark and Young formula.

Buckling of Plates in Shear-(Guide for Stability Design).

The Structural Stability Research Council, in their Guide to Stability Design-Criteria for Metal Structures, give the following formula for critical buckling shear stress for a thin rectangular plate subjected to shear with all edges fixed:

$$S_{scr} = [K\pi^2 E]/[12(1-v^2)(b/t)^2]$$

where:

$$K = 8.98 + 5.6(b/a)^2$$

and all other variables as defined by Salmon and Johnson in the previous section.

From the previous computation, the ratio of b/a would be 0.825 and the value of K would be:

$$K = 8.98 + 5.6(.825)^2$$
$$K = 12.79$$

The critical buckling shear stress would be:

$$S_{scr} = [12.79\pi^2(28X10^6)]/12(1-0.3^2)(24.75/.135)^2$$
$$S_{scr} = 9{,}630 \text{ psi}$$

This compares to 10,985 psi from the Roark and Young formula.

Example 28. Analysis of continuous overturn rails.

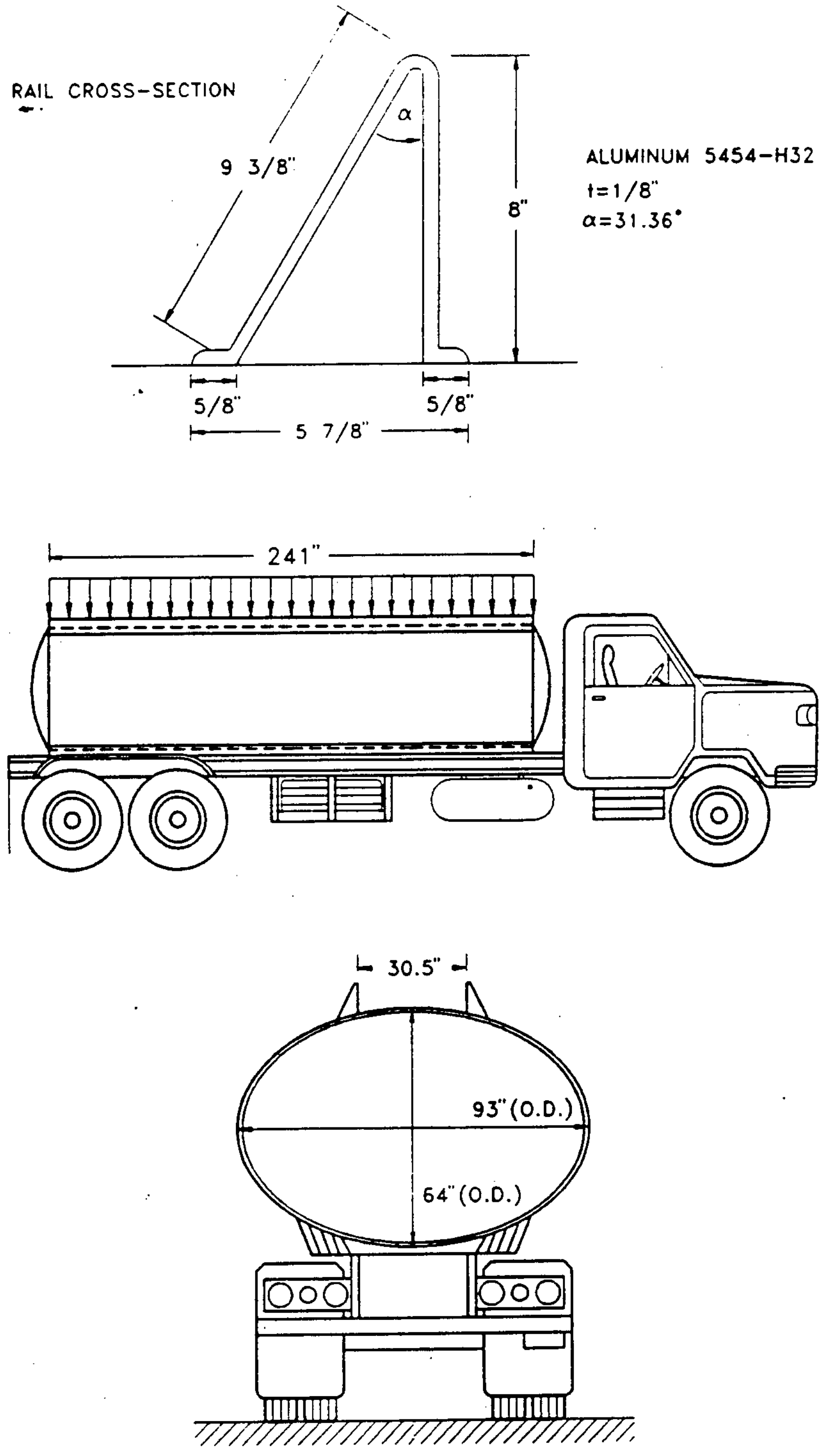


Figure 46. Continuous overturn protection rail on single unit cargo tank truck.

The continuous overturn rails shown on the vehicle in figure 46 are to be analyzed for the required loads of 2g's in the vertical, horizontal transverse and horizontal longitudinal directions.

The analysis presented serves to illustrate the manner in which overturn rails have been analyzed. IT IS NOT A RECOMMENDED METHOD. The method is based on assumed uniform support at the rail throughout its length and that assumption has not been substantiated. The authors think that concentrations of force at bulkheads and baffles make that assumption inappropriate. Work is being performed to address that question.

Gross weight of vehicle = 51,000 lbs.
Vertical load: F = 2G = 2(51,000 lbs.) = 102,000 lbs.
102,000/241 = 423.25 lbs/inch (uniformly distributed)
Horizontal load: W = 2G = (2)(51,000 lbs.) = 102,000 lbs.
102,000/241 = 423.25 lbs/inch (uniformly distributed)

A free body diagram for the rail for vertical load is shown in figure 47.

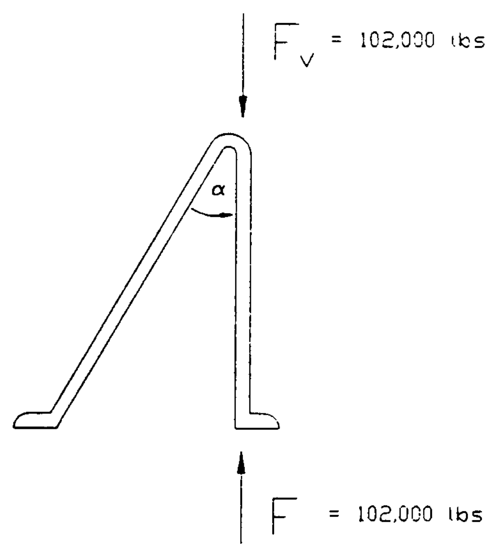


Figure 47. Free body diagram for overturn rail for vertical load.

Strength of the rail element in compression is controlled by either the yield strength or the compressible buckling strength. The compressive buckling strength is computed using Euler's formula.

$$P_{cr} = \pi^2 EI/L^2$$

where:

$E = 10.2 \text{ X } 10^6$ psi

$I = bh^3/12 = (241)(0.125)^3/12 = 0.0392 \text{ in}^4$

For the 8″ leg:

$P_{cr} = \pi^2(10.2 \text{ X } 10^6)(0.0392)/8^2 = 61{,}660$ lbs.

$S_{cr} = P_{cr}/A = 61{,}600/(241)(0.125) = 2{,}047$ psi

For the 9 3/8″ leg:

$P_{cr} = \pi^2(10.2 \text{ X } 10^6)(0.0392)/9.375^2 = 44{,}900$ lbs.

$S_{cr} = P_{cr}/A = 44{,}900/(241)(0.125) = 1{,}490$ psi

The actual compressive stress in the 8″ leg is:

$S = F/A$

$S = 102{,}000/[241(0.125)(2)]$ (2 devices)

$S = 1{,}693 \text{ psi} < 2{,}047$ psi OK

A free body diagram for the overturn rail subjected to horizontal transverse load is shown in figure 48.

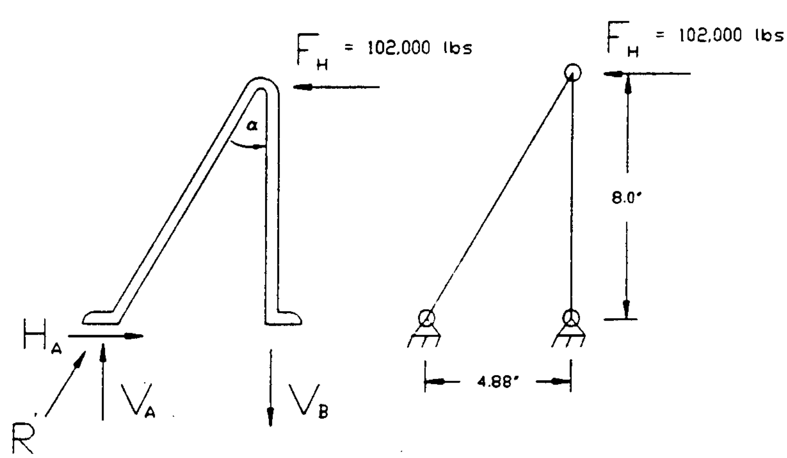


Figure 48. Free body diagram for overturn rail for horizontal transverse load.

Computation of stresses for this loading yields the following:

H_A = 102,000 lbs.
V_B = 102,000(8)/4.88
V_B = 167,213 lbs.
S_B = 167,213/(241)(.125)
S_B = 5,551 psi (tension)
V_A = 167,213 lbs.
$R = (167{,}213^2+102{,}000^2)^{0.5}$
R = 195,868 lbs.
S_A = 195,868/(241)(.125)
S_A = 6,502 psi (compression) > 1,490 psi FAILS

Note this analysis assumes that the overturn rails are fully supported by the shell along their entire length.

Analysis of Local Stresses.

Local stresses which occur at pads, cradles, or other supports must be considered in accordance with appendix G of the ASME Code. Appendix G references several other publications for analyzing different types of local stresses. For local stresses in cylinders due to external loads it references the Welding Research Council (WRC) Bulletin No. 107.

One common loading situation which must be considered are the stresses in the tank wall due to loads on overturn protection devices.

8. REAR-END PROTECTION

Requirements for rear-end protection for 400 Series cargo tank vehicles are given in §178.345-8(d). Geometric requirements are the same as current requirements for MC 300 Series. Structural strength requirements specify a load from 2 g's deceleration but no factor of safety is required and the material strength to be used is not addressed. The load from 2 g's deceleration would be twice the total weight of the fully loaded cargo tank motor vehicle. §178.345-8(b) addresses bottom damage protection and requires that a bottom damage protection device be able to withstand a force of 155,000 lbs. regardless of the weight of the vehicle. If rear-end protection devices protect an outlet, projection or piping, it must also be designed to withstand a minimum force of 155,000 lbs. although twice the weight of the vehicle may be less than 155,000 lbs. In other words, under these conditions, rear-end protection must also meet the bottom damage protection requirement. In addition, §178.345-8(d) states that the load should be uniformly applied in the horizontal plane at an angle of 10 degrees or less to the longitudinal axis of the vehicle as shown in figure 49.

This regulation seems to imply that performance is the key requirement. It is required that the bumper/device prevent contact with lading containing components of the cargo tank motor vehicle at the specified load. The bumper/device may be sacrificial so long as it serves the required function. Yield strength of the material is an appropriate limit state because the device would have few or no repetitions of the maximum load. An ultimate strength failure mechanism type of analysis is also appropriate.

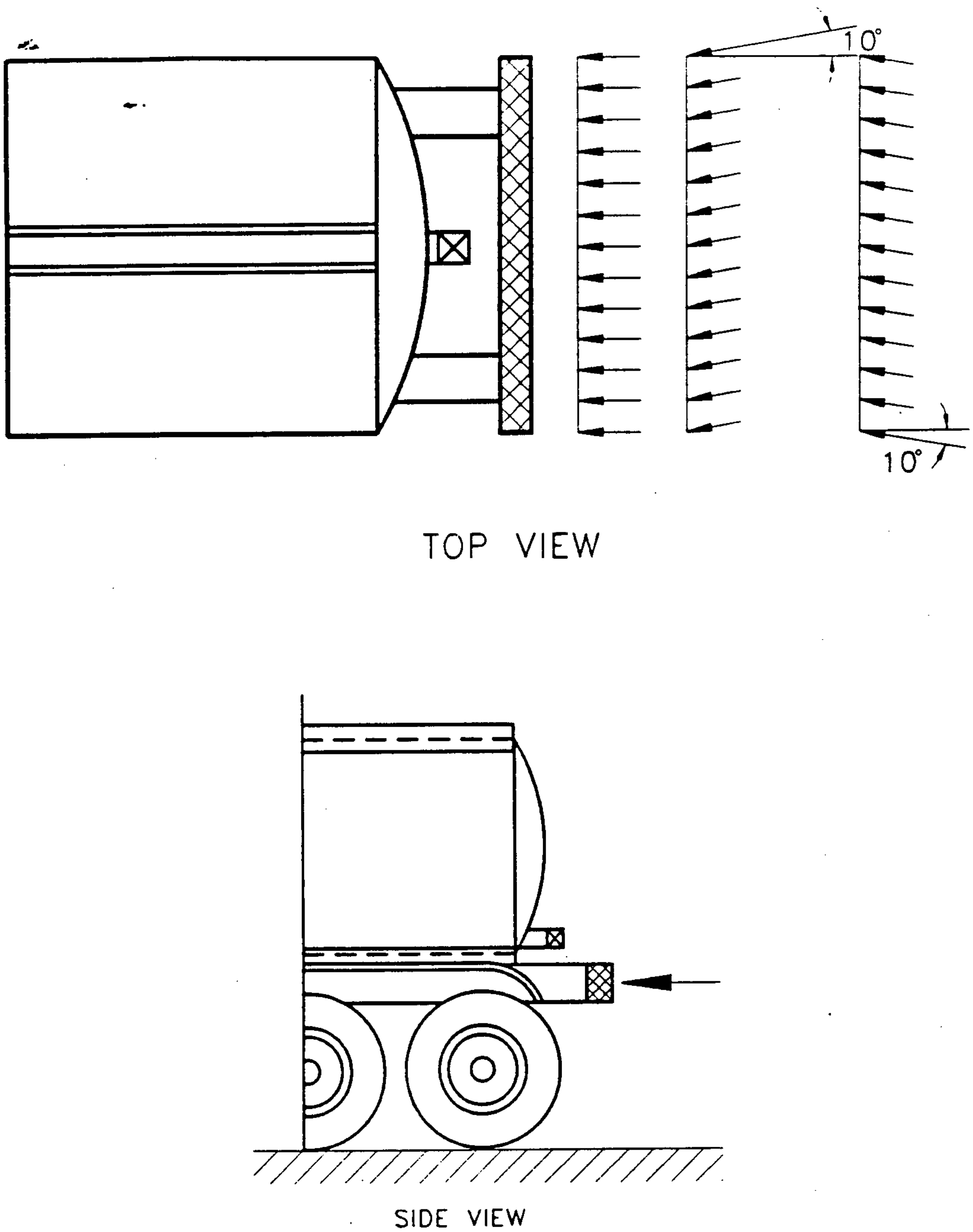


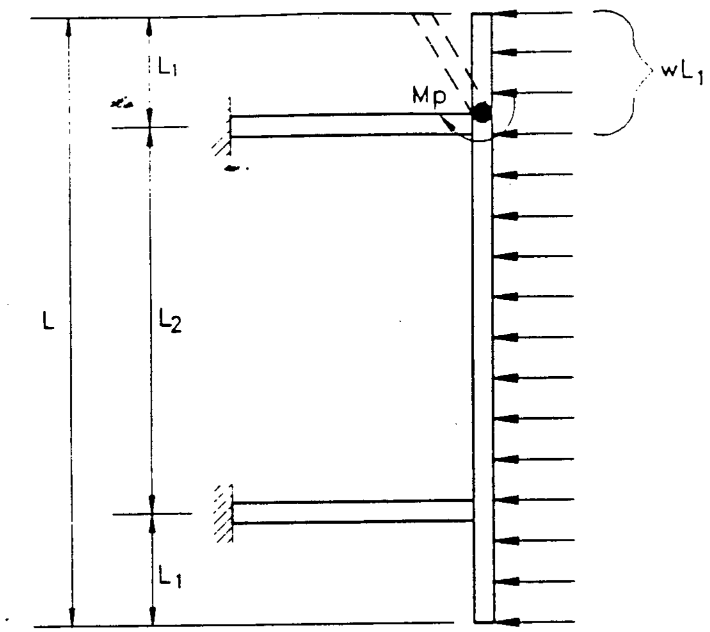
Figure 49. Loading for rear-end protection devices for DOT 400 Series cargo tank motor vehicles.

Possible ultimate strength, plastic hinge, failure mechanisms for rear-end protection devices consisting of two longitudinal and one transverse structural members are shown in figures 50 through 52.

In figure 50, the failure mechanism consists of the overhanging portion of the transverse member being deflected forward with a plastic hinge formed in the transverse member at its juncture with the longitudinal member.

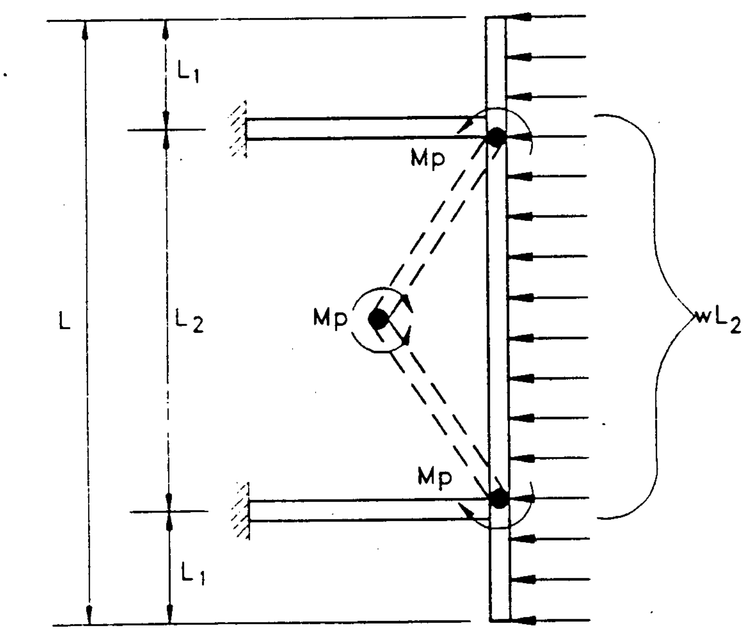
In figure 51, the failure mechanism typically forms in the transverse member between the two longitudinal members and has three plastic hinges in the transverse member. However, if the moment capacity of the longitudinal members is small, another failure mechanism might form.

The plastic hinge failure mechanism shown in figure 52 is for the side load component when the load is applied at 10 deg. to the longitudinal axis of the vehicle. The parallel component will be less critical than the case where full load is applied parallel to the longitudinal axis. For the transverse component, plastic hinges would form in the longitudinal struts at their front ends if they are suitably attached to the vehicle. Plastic hinges at the rear end of the longitudinal members would form either in the longitudinal member or the transverse member depending on which member has the smaller plastic moment value.



$$wL_1 = 2M_p/L_1$$
so that
$$wL = (2M_p/L_1)(L/L_1)$$
where
$$wL > 2w_{TOTAL}$$
then
$$2w_{TOTAL} \le (2M_P/L_1)(L/L_1)$$

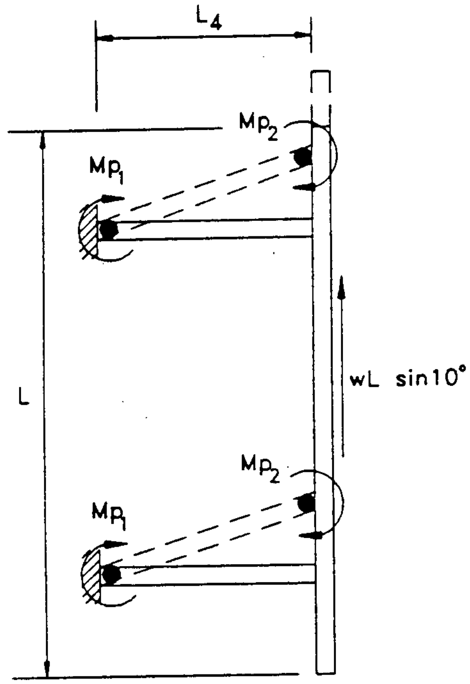
Figure 50. Plastic hinge failure mechanism for cantilevered portion of horizontal bar.



$$wL_2 = 16M_p/L_2$$
so that
$$wL = (16M_p/L_2)(L/L_2)$$
where
$$wL \ge 2w_{TOTAL}$$
then
$$2w_{TOTAL} \le (16M_p/L_2)(L/L_2)$$

Figure 51. Plastic hinge failure mechanism for interior portion of horizontal bar.

The parallel component will be less critical than the case where full load is applied parallel to the longitudinal axis. For the transverse component, plastic hinges would form in the longitudinal struts at their front ends if they are suitably attached to the vehicle. Plastic moments at the rear end of the longitudinal members would form either in the longitudinal member or the transverse member depending on which value is lesser.



$$wL = (2M_{p1}+2M_{p2})/L_4\sin 10°$$
where
$$wL \geq 2w_{TOTAL}$$
then
$$2w_{TOTAL} \leq (2M_{p1}+2M_{p2})/L_4\sin 10°$$

Figure 52. Possible plastic hinge failure mechanism for side load. Other mechanisms may be possible depending upon relative strengths of members and connections.

Example 29. Analysis of stresses in rear-end protection device.

The rear-end protection device shown in figure 53 is used on a cargo tank trailer. Total weight of the vehicle is 80,000 lbs. Adequacy of the strength of the device is to be checked against DOT series requirements. The device must withstand a force of twice the weight of the vehicle. 2 X 80,000 = 160,000 lbs. This load acts in the horizontal plane anywhere within $\pm 10°$ of the longitudinal axis of the vehicle. For the device shown:

L = 88 in.
L_1 = 25 in.
L_2 = 38 in.
L_4 = 34 in.

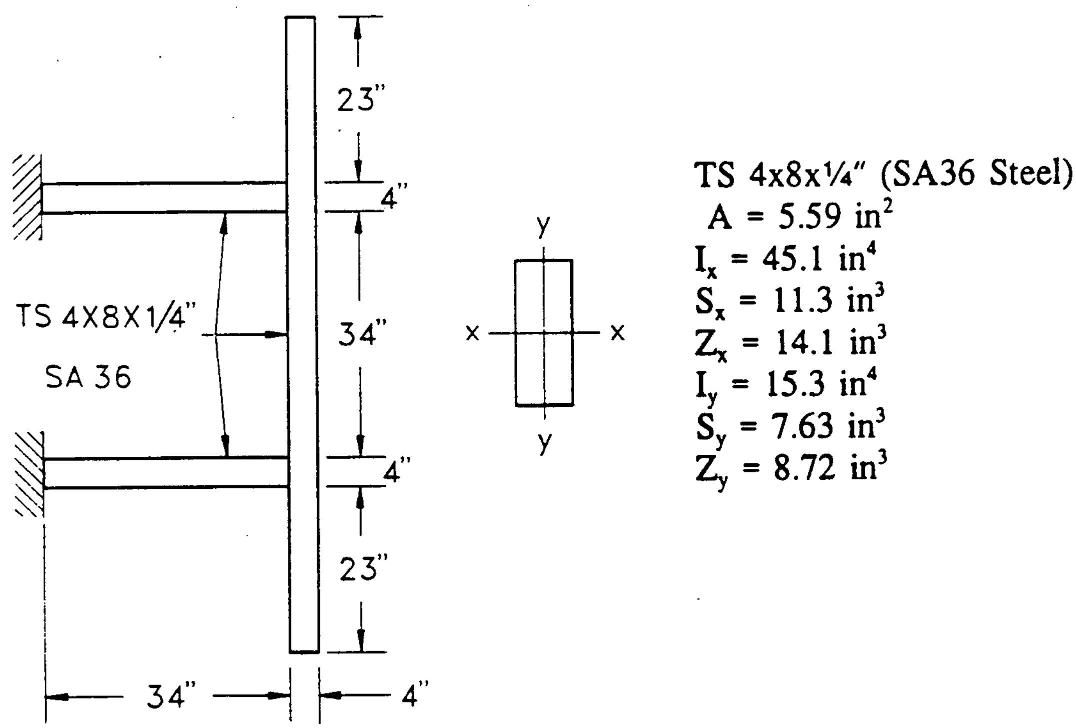


Figure 53. Rear-end protection device for cargo tank trailer.

The plastic moment capacity of the TS section about its vertical axis is:

$$M_p = F_y Z_p$$
$$M_p = 36(8.72)$$
$$M_p = 314 \text{ in-k}$$

For longitudinal loading, the failure mechanism shown in figure 47 yields a total strength of :

$$wL = (2M_p/L_1)(L/L_1)$$
$$wL = [(2)(314)/25](88/25)$$
$$wL = 88 \text{ kips} < 160 \text{ kips} \quad \underline{\text{FAIL}}$$

The failure mechanism shown in figure 48 yields a total strength of:

$$wL = (16M_p/L_2)(L/L_2)$$
$$wL = [(16)(314)/38](88/38)$$
$$wL = 306 \text{ kips} > 160 \text{ kips} \quad \underline{\text{OK}}$$

The failure mechanism shown in figure 49 yields a strength of:

$$wL = (2M_{p1}+2M_{p2})/L_4 \sin 10^\circ$$
$$wL = [(2)(314)+2(314)]/34 \sin 10^\circ$$
$$wL = 212 \text{ kips} > 160 \text{ kips} \quad \underline{\text{OK}}$$

The analysis shows that the overhanging end of the transverse member does not have adequate strength to resist the required load.

Many rear-end protection devices will be more complicated because diagonal braces, web plates and other structural members have been provided to increase strength of the device. Such devices can not be analyzed accurately with simple idealizations and methods of analysis. Appropriate approximate analysis or finite element analysis procedures are in order for such devices.

Some rear-end protection devices are constructed such that the longitudinal members slope downward to the rear (e.g. the horizontal bumper bar is lower than the front attach points of the longitudinal members). For such devices, a bending moment about the transverse axis will exist at the front attach points of the underride device and stresses due to that moment plus axial load should be evaluated.

Some devices on single unit trucks consist of a horizontal bumper bar attached to the truck frame with two vertical members and with two diagonal struts extending from the horizontal bar forward and up to the truck frame. For such devices, the horizontal bar can be analyzed as a beam as illustrated in the previous example. For longitudinal load, the attaching members (verticals and diagonals) can be analyzed as trusses if member alignment is suitable. For side load, the device is a three dimensional structure and finite element analysis procedures are appropriate. However, a plastic hinge failure mechanism might also be appropriate.

REFERENCES

1. Pressure Vessel Handbook, Ninth Edition, Eugene F. Megyesy, Pressure Vessel Handbook Publishing Inc., P. O. Box 35365, Tulsa, Oklahoma, 74153.

2. Formulas for Stress and Strain, Fifth Edition, Raymond J. Roark and Warren C. Young, McGraw-Hill Book Company.

3. ASME Boiler and Pressure Vessel Code, Section VIII-Division 1, Pressure Vessels, American Society of Mechanical Engineers, 1992 Edition.

4. ASME Boiler and Pressure Vessel Code, Section II-Part D, Materials-Properties, American Society of Mechanical Engineers, 1992 Edition.

5. Title 49, Code of Federal Regulations, Part 178.

6. Theory of Elasticity, Second Edition, Timoshenko and Goodier, McGraw-Hill Book Company, 1951.

7. Structural Analysis for Moment Distribution

8. Strength of Materials book

9. Design of Welded Structures, Omer W. Blodgett, The James T. Lincoln Arc Welding Foundation, Cleveland, Ohio, 1966.

10. Alcoa Structural Handbook

11. Engineering Data for Aluminum Structures, Construction Manual Series, Section 3, The Aluminum Association, 3rd Edition, January, 1975.

12. Strength of Materials, Part II, S. Timoshenko, D. Van Nostrand Company, Inc., New York, N.Y.

13. Manual of Steel Construction, Eighth Edition, American Institute of Steel

Construction, Chicago, Illinois, 1980.

14. Theory of Plates and Shells, Second Edition, S. Timoshenko and S. Woinowsky-Krieger, McGraw-Hill Book Company, 1959.

15. Local Stresses In Spherical & Cylindrical Shells due to External Loadings, March 1979 Revision of WRC Bulletin 107/August 1965. K. R. Wichman, A. G. Hopper and J. L. Mershon, Welding Research Council, New York, N.Y.

16. Local Stresses In Cylindrical Shells due to External Loadings on Nozzles, Supplement to WRC Bulletin No. 107-(Revision 1), J. L. Mershon, K. Mokhtarian, C. G. Ranjan and E. C. Rodabaugh, Welding Research Council, New York, N.Y.

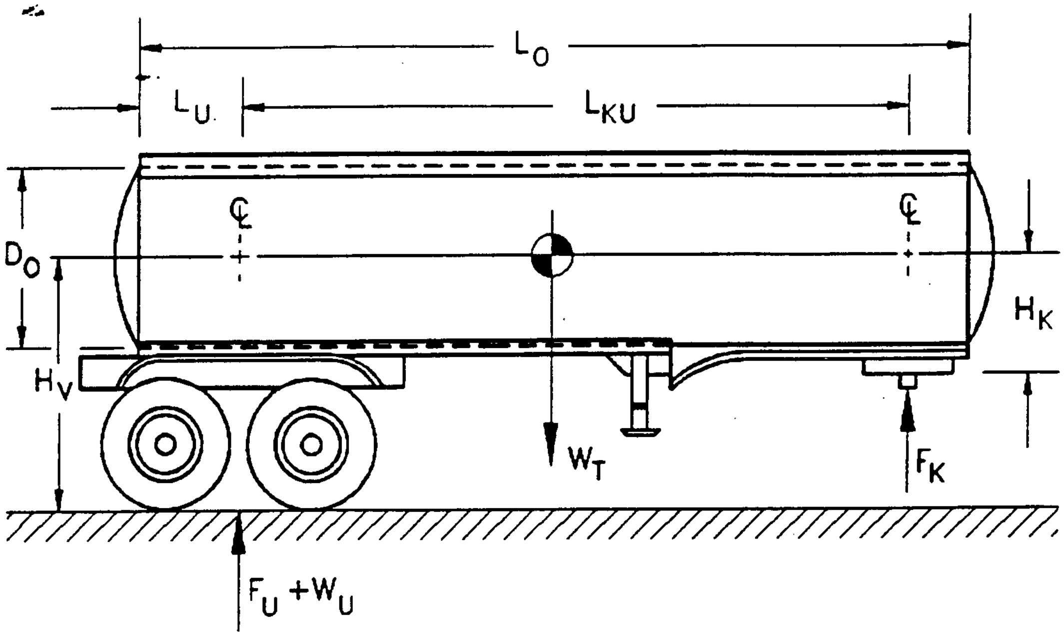
17. Steel Structures: Design and Behavior, Third Edition, Salmon and Johnson, Harper & Row Publishers, 1990.

18. Guide to Stability Design: Criteria for Metal Structures, Fourth Edition, Structural Stability Research Council, Edited by T. V. Galambos, A Wiley-Interscience Publication, John Wiley & Sons, 1988.

19. Manual of Steel Construction-Load & Resistance Factor Design, Volume I, Second Edition, American Institute of Steel Construction, 1994.

20. Mechanics of Materials, Second Edition, R. C. Hibbeler, MacMillan College Publishing Company, 1994.

APPENDIX A. NOMENCLATURE



A = Cross sectional area of tank shell or other structural member, in^2.

A_m = Area enclosed by median line of vessel wall, in^2.

a_g = Acceleration due to gravity (32.2 ft/sec/sec).

a = Acceleration, ft/sec/sec.

b = Number of baffles in cargo tank compartment.

c = Distance from neutral axis to extreme fiber of flexural member, inches.

C_i = Inside circumference of shell, inches.

C_o = Outside circumference of shell, inches.

D = Inside diameter of head at its juncture with shell, inches.

D_i	=	Inside diameter of shell, inches.
D_{iy}	=	Inside width of non-circular pattern at minor axis, inches.
e	=	Joint efficiency for, or the efficiency of, appropriate joint in cylindrical or spherical shells, or the efficiency of ligaments between openings, whichever is less. For welded vessels, use the efficiency specified in UW-12 of the ASME code. The ASME Code uses E for joint efficiency.
E	=	Modulus of elasticity of material at design temperature, psi. The ASME Code uses E for joint efficiency.
F_1	=	Factor for longitudinal decelerative/accelerative force.
	=	0.35 for normal operating loadings.
	=	0.70 for extreme dynamic loadings.
F_2	=	Factor for vertical acceleration.
	=	0.35 for normal operating loadings.
	=	0.70 for extreme dynamic loadings.
F_3	=	Factor for lateral acceleration.
		0.20 for normal operating loadings.
		0.40 for extreme dynamic loadings.
F_u	=	Reaction load at undercarriage, lbs.
F_k	=	Reaction load at kingpin, lbs.
g	=	Acceleration due to gravity, 32.2 ft/sec/sec.
h	=	Inside depth of ellipsoidal head (from point of tangency of flange to inside surface at center), inches.
H_k	=	Height from transverse hinge point of fifth wheel to centroid of transverse cross section of tank, inches.
H_v	=	Height from road to centerline of tank cross section, inches.

I = Moment of inertia, in^4.

For elliptical cross sections which are true conic sections of uniform wall thickness, the moment of inertia is given as follows:

$$I = \frac{\pi}{4} [R_{xo}R_{yo}^3 - R_{xI}R_{yI}^3]$$

Since many vessels of non-circular cross section are not true conic sections, the designer of the shell may find it necessary to utilize alternate formulas more appropriate to the section under consideration.

J = Polar moment of inertia, in^4.

L = Dish radius of major axis of non-circular pattern, inches.

L_k = Distance from cargo tank front head seam to center of kingpin, inches.

L_o = Overall length of shell from front head seam to rear head seam, inches.

L_s = Center to center spacing of ring stiffeners, inches.

L_u = Distance from cargo tank rear head seam to center of undercarriage, that is, halfway between the axles of a tandem, the second axle of a tridem, or the center of the axle of a single axle trailer, inches.

L_{ku} = Distance from centerline of kingpin to center of undercarriage, inches.

L_x = Distance from cargo tank front head seam, inches.

L_{max} = Distance from cargo tank rear head seam to cross section where maximum bending moment occurs, inches.

M = Bending moment, in-lbs.

P = Internal or external pressure, psi.

P_d = Dynamic impact pressure, psi.

P_h = Static head pressure, psi.

P_m = Maximum allowable working pressure (MAWP), psi.

MAWP = The MAWP for each cargo tank must be greater than or equal to the largest of the following: (The MAWP derived is the pressure to be used as prescribed in the ASME Code in the design of the tank).

(1) The pressure prescribed for the lading in part 173:

(2) Vapor pressure of the most volatile lading, at 115° F (expressed in psig), plus the maximum static pressure exerted by the lading at the maximum lading density, plus any pressure exerted by a gas padding (including air in the ullage space or dome, if used, or

(3) The maximum pressure in the tank during loading or unloading.

P_t = Total static pressure = $P_d + P_h$, psi.

Q = Dimensionless factor from the ASME Code.
A factor in the formulas for torispherical heads depending on the head proportion (R_h/R_k). Based on the ratio on the inside crown radius to the inside knuckle radius.

R_k = Knuckle radius, inches.

R_h = Inside radius of head (crown radius), inches.

R_i = Inside radius of shell, inches.

R_o = Outside radius of shell, inches.

R_{xi} = Half the length of the inside major axis for true conic elliptical sections, inches.

R_{xo} = Half the length of the outside major axis for true conic elliptical sections, inches.

R_{yi} = Half the length of the inside minor axis for true conic elliptical sections, inches.

R_{yo} = Half the length of the outside minor axis for true conic elliptical sections, inches.

$\bar{r}$ = Radius of curvature, inches.

S = Effective stress, at any given point under the most severe combination of static and dynamic loadings that can occur at the same time, psi.

S_a = Allowable tensile stress from DOT Regulation or ASME Code, psi.

S_b = Allowable compressive buckling stress, psi.

S_{bA} = Critical compressive buckling stress - Alcoa Formula, psi.

S_{bY} = Critical compressive buckling stress - Roark & Young Formula, psi.

S_c = Compressive stress due to static bending loads, psi.

S_d = Effective tensile stress generated by a 2 "g" deceleration of liquid cargo, psi.

S_{dx} = Longitudinal tensile stress generated by a 2 "g" deceleration of liquid cargo, psi.

S_{dy} = Circumferential tensile stress generated by a 2 "g" deceleration of liquid cargo, psi.

S_m = Maximum allowable tensile stress for a particular material per Table UHA-23, UCS-23, or UNF-23 of the ASME Code, psi.

S_{scr} = Critical shear buckling stress for rectangular plate.

S_u = Ultimate tensile strength for a particular material, psi.

S_x = Net longitudinal stress, in psi, generated by appropriate combinations of the following stresses:

$$S_{x1}, S_{x2}, S_{x3}, S_{x4}, S_{x5}, S_{x6}, S_{x7}, S_{x8}, S_{x9}, S_{x10} \text{ and } S_{x11}$$

S_{x1} = Longitudinal stresses resulting from the MAWP or the lowest pressure at which the cargo tank may operate, psi.

S_{x2} = Longitudinal tensile stress due to static head of liquid lading, psi.

S_{x3} = Tensile or compressive stress resulting from the bending moment caused by static weight of the fully loaded cargo tank, all structural elements, equipment and appurtenances supported by the cargo tank wall, psi.

S_{x4} = Tensile or compressive stress generated by the bending moment resulting from a vertical accelerative force, psi.

S_{x5} = Tensile stress generated by the axial load resulting from longitudinal decelerative force from application of trailer brakes, psi.

S_{x6} = Tensile or compressive stress generated by the bending moment resulting from a longitudinal decelerative force from application of trailer brakes, psi.

S_{x7} = Tensile stress generated by the axial load resulting from a longitudinal accelerative force, psi.

S_{x8} = Tensile or compressive stress generated by the bending moment resulting from a longitudinal accelerative force, psi.

S_{x9} = Tensile or compressive stress generated by the bending moment resulting from a lateral accelerative force, psi.

S_{x10} = Compressive stress generated by the axial load resulting from a longitudinal force from application of tractor brakes, psi.

S_{x11} = Tensile or compressive stress generated by the bending moment resulting from a longitudinal decelerative force from application of tractor brakes, psi.

S_y = Net circumferential stress, in psi, resulting from the appropriate combinations of the following stresses:

S_{y1} and S_{y2}

S_{y1} = Circumferential stress generated by internal or external pressure, when applicable, psi.

S_{y2} = Circumferential stress due to static head of liquid lading, psi.

S_s = Net shear stress, in psi, resulting from the appropriate combinations of the following stresses:

S_{s1}, S_{s2}, S_{s3} and S_{s4}

S_{s1} = Vertical (flexural) shear stress generated by gravity of fully loaded cargo tank, psi.

S_{s2} = Vertical (flexural) shear stress generated by a vertical accelerative force, psi.

S_{s3} = Lateral (flexural) shear stress generated by a lateral accelerative force, psi.

S_{s4} = Torsional shear stress generated by a lateral accelerative force, psi.

t = Actual thickness of tank wall, inches.

t_{min} = Minimum thickness for the shell and heads must be such that the maximum stress levels specified in §178.345-3 are not exceeded, inches.

t_h = Minimum thickness of head, inches.

t_s = Minimum thickness of shell (corrosion allowance is added to this value), inches.

T = Torque, inch-lbs.

V = Volume of tank, gallons.

w = Weight per unit length, lbs/inch.

W_b = Maximum density of product which can be carried in a cargo tank which may not be fully loaded, lbs/gal.

W_c = Maximum density of tank contents for a fully loaded cargo tank, lbs/gal.

W_k = Weight of kingpin/upper coupler, lbs.

W_L = Maximum total weight of product which can be carried in the vessel, lbs.

W_s = Weight of empty vessel (Total weight of empty trailer minus W_k minus W_u), lbs.

W_T = Weight of vessel and contents = W_s+W_L, lbs.

W_{Ti} = Weight of vessel and contents per unit length, lbs/inch.

W_{Tr} = Weight of tractor, lbs.

W_u = Weight of trailer undercarriage, lbs.

Z_e = Elastic section modulus = I/R_o = I/c, in^3.

Z_p = Plastic section modulus, in^3.

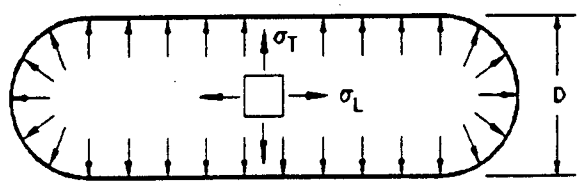
α = Coefficient of thermal expansion for material, inches/inch/deg F.

ν = Poisson's ratio

APPENDIX B. SELECTED FORMULAS
for Section Properties and Stresses

Stresses in Thin-wall Pressure Vessels:

The basic formulas for stresses in thin-wall pressure vessels (due to either internal or external pressure) come from basic strength of materials. For a cylindrical vessel:

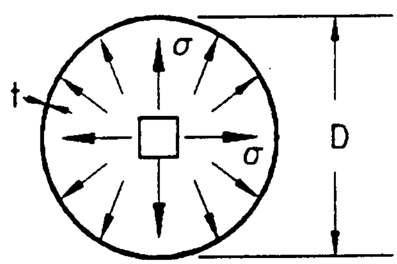


P=pressure

$$\sigma_L = \frac{PD}{2t}$$

$$\sigma_T = \frac{PD}{4t}$$

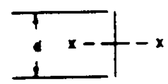
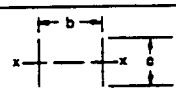
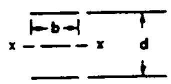
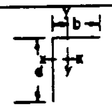
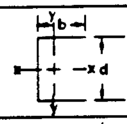
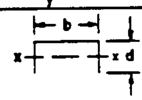
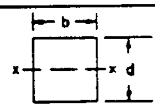
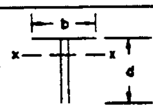
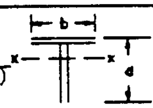
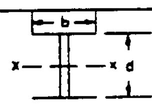
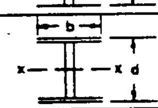
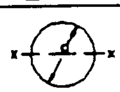
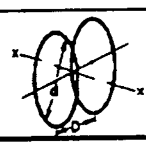
For a spherical vessel:



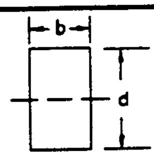
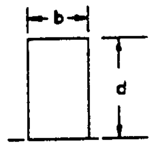
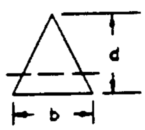
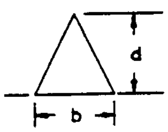
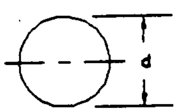
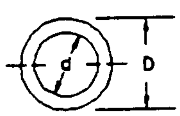
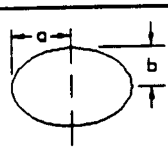
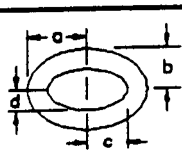
$$\sigma = \frac{PD}{4t}$$

As stated above, these formulas are also applicable to vessels subjected to external pressure (vacuum inside). However, the thin walls of a vessel may buckle elastically when loaded in compression and the strength of the vessel must be based on the compressive buckling strength of the thin walls. Bulkheads baffles or rings which have adequate stiffness/strength and are suitably attached to the vessel walls, serve to reduce the effective length of vessel wall subjected to compressive buckling and thereby increase the strength of the vessel.

Properties of Weld Treated as Line

Outline of Welded Joint b=width d=depth	Bending (about horizontal axis x−x)	Twisting
	$S_w = \frac{d^2}{6}$	$J_w = \frac{d^3}{12}$
	$S_w = \frac{d^2}{3}$	$J_w = \frac{d(3b^2+d^2)}{6}$
	$S_w = bd$	$J_w = \frac{b^3+3bd^2}{6}$
 $N_y = \frac{b^2}{2(b+d)}$ $N_x = \frac{d^2}{2(b+d)}$	$S_w = \underset{\text{top}}{\frac{4bd+d^2}{6}} = \underset{\text{bottom}}{\frac{d^2(4b+d)}{6(2b+d)}}$	$J_w = \frac{(b+d)^4-6b^2d^2}{12(b+d)}$
$N_y = \frac{b^2}{2b+d}$ 	$S_w = bd + \frac{d^2}{6}$	$J_w = \frac{(2b+d)^3}{12} - \frac{b^2(b+d)^2}{(2b+d)}$
$N_x = \frac{d^2}{b+2d}$ 	$S_w = \underset{\text{top}}{\frac{2bd+d^2}{3}} = \underset{\text{bottom}}{\frac{d^2(2b+d)}{3(b+d)}}$	$J_w = \frac{(b+2d)^3}{12} - \frac{d^2(b+d)^2}{(b+2d)}$
	$S_w = bd + \frac{d^2}{3}$	$J_w = \frac{(b+d)^3}{6}$
$N_y = \frac{d^2}{b+2d}$ 	$S_w = \underset{\text{top}}{\frac{2bd+d^2}{3}} = \underset{\text{bottom}}{\frac{d^2(2b+d)}{3(b+d)}}$	$J_w = \frac{(b+2d)^3}{12} - \frac{d^2(b+d)^2}{(b+2d)}$
$N_y = \frac{d^2}{2(b+d)}$ 	$S_w = \underset{\text{top}}{\frac{4bd+d^2}{3}} = \underset{\text{bottom}}{\frac{4bd^2+d^3}{6b+3d}}$	$J_w = \frac{d^3(4b+d)}{6(b+d)} + \frac{b^3}{6}$
	$S_w = bd + \frac{d^2}{3}$	$J_w = \frac{b^3+3bd^2+d^3}{6}$
	$S_w = 2bd + \frac{d^2}{3}$	$J_w = \frac{2b^3+6bd^2+d^3}{6}$
	$S_w = \frac{\pi d^2}{4}$	$J_w = \frac{\pi d^3}{4}$
	$I_w = \frac{\pi d}{2}\left(D^2+\frac{d^2}{2}\right)$ $S_w = \frac{I_w}{c}$ where $c = \frac{\sqrt{D^2+d^2}}{2}$	

Properties of Standard Sections

	Area A	Moment of Inertia I	Section Modulus S	Radius of Gyration r
	bd	$\frac{bd^3}{12}$	$\frac{bd^2}{6}$	$\frac{d}{\sqrt{12}}$
	bd	$\frac{bd^3}{3}$	$\frac{bd^2}{3}$	$\frac{d}{\sqrt{3}}$
	$\frac{bd}{2}$	$\frac{bd^3}{36}$	$\frac{bd^2}{24}$	$\frac{d}{\sqrt{18}}$
	$\frac{bd}{2}$	$\frac{bd^3}{12}$	$\frac{bd^2}{12}$	$\frac{d}{\sqrt{6}}$
	$\frac{\pi d^2}{4}$	$\frac{\pi d^4}{64}$	$\frac{\pi d^3}{32}$	$\frac{d}{4}$
	$\frac{\pi}{4}(D^2-d^2)$	$\frac{\pi}{64}(D^4-d^4)$	$\frac{\pi}{32}\frac{(D^4-d^4)}{D}$	$\frac{\sqrt{D^2+d^2}}{4}$
	πab	$\frac{\pi a^3 b}{4}$	$\frac{\pi a^2 b}{4}$	$\frac{a}{2}$
	$\pi(ab-cd)$	$\frac{\pi}{4}(a^3 b-c^3 d)$	$\frac{\pi(a^3 b-c^3 d)}{4a}$	$\frac{1}{a}\sqrt{\frac{a^3 b-c^3 d}{ab-cd}}$

APPENDIX C. STRENGTHS OF SELECTED MATERIALS
AT NORMAL TEMPERATURES

STEEL

ASME DESIGN.	ASTM DESIGN.	PRODUCT FORM	F_y, ksi YIELD STR.	F_u, ksi TENSILE STR.
SA36	A36	PLATE AND ROLLED SHAPES	36	58
SA53-B		PIPE	35	60
SA105		FORGINGS	36	70
SA106-A	A106	PIPE	30	48
SA193-B7		BOLTING 4-7 IN.	75	100
SA193-B7		BOLTING 2.5-4 IN.	95	115
SA193-B7		BOLTING < 2.5 IN.	105	125
SA194-2H		------	-----	-----
SA240-304				
SA240-304L		PLATE	25	70
SA240-316				
SA240-316L		PLATE	25	70
SA285-C		PLATE	30	55
SA285-C		PLATE/SHT	30	55
SA414				
SA516				
	A569			
	A570			
	A572			
	A607			
	A656			
	A715			

ALUMINUM

SPEC. NO.	ALLOY DESIGN.	PRODUCT FORM	F_y, ksi YIELD STR.	F_u, ksi TENSILE STR.
SB209	5052-0	PLATE/SHT	9.5	25
SB209	5052-32	PLATE/SHT	23	31
SB209	5052-H34	PLATE/SHT	26	34
SB209	5086-0	PLATE/SHT	14	35
SB209	5086-H32	PLATE/SHT	28	40
SB209	5086-H34	PLATE/SHT	34	44
SB209	5154-0	PLATE/SHT	11	30
SB209	5154-H32	PLATE/SHT	26	36
SB209	5154-H34	PLATE/SHT	29	39
SB209	5254-0	PLATE/SHT	11	30
SB209	5254-H32	PLATE/SHT	26	36
SB209	5254-H34	PLATE/SHT	29	39
SB209	5454-0	PLATE/SHT	12	31
SB209	5454-H32	PLATE/SHT	26	36
SB209	5454-H34	PLATE/SHT	29	39
SB209	5652-0	PLATE/SHT	9.5	25
SB209	5652-H32	PLATE/SHT	23	31
SB209	5652-H34	PLATE/SHT	26	34

APPENDIX D. DIMENSIONS OF PIPE

Nominal Pipe Size	Outside Diameter	NOMINAL WALL THICKNESS											
		Sched. 20	Sched. 30	Std. Weight	Sched. 40	Sched. 60	Extra Strong	Sched. 80	Sched. 100	Sched. 120	Sched. 140	Sched. 160	XX Strong
1/8	0.405	---	---	0.068	0.068	---	0.095	0.095	---	---	---	---	---
1/4	0.540	---	---	0.088	0.088	---	0.119	0.119	---	---	---	---	---
3/8	0.675	---	---	0.091	0.091	---	0.126	0.126	---	---	---	---	---
1/2	0.840	---	---	0.109	0.109	---	0.147	0.147	---	---	---	0.187	0.294
3/4	1.050	---	---	0.113	0.113	---	0.154	0.154	---	---	---	0.218	0.308
1	1.315	---	---	0.133	0.133	---	0.179	0.179	---	---	---	0.250	0.358
1-1/4	1.660	---	---	0.140	0.140	---	0.191	0.191	---	---	---	0.250	0.382
1-1/2	1.900	---	---	0.145	0.145	---	0.200	0.200	---	---	---	0.281	0.400
2	2.375	---	---	0.154	0.154	---	0.218	0.218	---	---	---	0.343	0.436
2-1/2	2.875	---	---	0.203	0.203	---	0.276	0.276	---	---	---	0.375	0.552
3	3.500	---	---	0.216	0.216	---	0.300	0.300	---	---	---	0.438	0.600
3-1/2	4.000	---	---	0.226	0.226	---	0.318	0.318	---	---	---	---	0.636
4	4.500	---	---	0.237	0.237	---	0.337	0.337	---	0.438	---	0.531	0.674
5	5.563	---	---	0.258	0.258	---	0.375	0.375	---	0.500	---	0.625	0.750
6	6.625	---	---	0.280	0.280	---	0.432	0.432	---	0.562	---	0.718	0.864
8	8.625	0.250	0.277	0.322	0.322	0.406	0.500	0.500	0.593	0.718	0.812	0.906	0.875
10	10.750	0.250	0.307	0.365	0.365	0.500	0.500	0.593	0.718	0.843	1.000	1.125	---
12	12.750	0.250	0.330	0.375	0.406	0.562	0.500	0.687	0.843	1.000	1.125	1.312	---

APPENDIX E. MOTOR CARRIER REGULATIONS FOR MC 306/307/312 CARGO TANKS.

NOTE: Construction of MC 306, 307 and 312 cargo tanks was allowed until 31 August 1995 to specifications in effect on 30 December 1990.

§178.340 General design and construction requirements applicable to specifications MC 306 (§178.341), MC 307 (§178.342), and MC 312 (§178.343) cargo tanks.

§178.340-1 Specification requirements for MC 306, MC 307 and MC 312 cargo tanks.

(a) Specification MC 306, MC 307 and MC 312 cargo tanks constructed on or after December 1, 1967 for the bulk transportation of hazardous commodities must meet the requirements contained in this section in addition to the requirements of each applicable specification as contained in §178.341 (MC 306), §178.342 (MC 307) and (MC 312).

(b) All of these specification requirements are minimum requirements.

§178.340-2 General requirements.

(a) Every cargo tank and vessel shall be designed and constructed in accordance with the best known and available practices in addition to the other applicable cargo tank specification requirements.

(b) Those requirements relating to parts and accessories applicable to all motor vehicles engaged in interstate commerce as contained in Part 393 of the Motor Carrier Safety Regulations are an integral part of this specification.

(c) Where applicable the additional requirements prescribed in Part 173 to accommodate specific commodities are considered an integral part of these specifications.

(d) Multi-purpose cargo tank.

(1) A single cargo tank may be divided into compartments of different specification construction. Each such compartment shall conform to specification requirements concerned.

(2) A single cargo tank may be physically altered to comply with another cargo tank specification in these regulations; or altered to accommodate a commodity not requiring a DOT specification tank.

§178.340-3 Material

(a) All sheet and plate material for shell, heads, bulkheads and baffles for cargo tanks which are not required to be constructed in accordance with the American Society of Mechanical Engineers' Boiler and Pressure Vessel Code shall meet the following minimum applicable requirements:

(1) ALUMINUM ALLOYS (AL). Only aluminum alloy material suitable for fusion welding and in compliance with one of the following ASTM specifications shall be used:

ASTM B-209 Alloy 5052
ASTM B-209 Alloy 5086
ASTM B-209 Alloy 5154
ASTM B-209 Alloy 5254
ASTM B-209 Alloy 5454
ASTM B-209 Alloy 5652

All heads, bulkheads, baffles, and ring stiffeners may use 0 temper (annealed) or stronger tempers. All shell shall be made of materials with properties equivalent to H 32 or H 34 tempers, except that lower ultimate strength tempers may be used if the minimum shell thicknesses in Table II in §§178.341-2, 178.342-2, or 178.343-2 are increased in inverse proportion to the lesser ultimate strength.

(2) STEEL.

	Mild steel (MS)	High strength low alloy steel (HSLA)	Austenitic stainless steel (SS)
Yield point-p.s.i.------------	25,000	45,000	25,000
Ultimate strength-p.s.i.-----	45,000	60,000	70,000
Elongation, 2-inch samples-%	20	25	30

§178.340-4 Structural integrity.

(a) Maximum stress values. The maximum calculated stress value must not exceed 20 percent of the minimum ultimate strength of the material as authorized in 340-3, except when ASME Code pressure vessel design requirements apply.

(b) Loadings. Cargo tanks shall be provided with additional structural elements as necessary to prevent resulting stresses in excess of those permitted in paragraph (a) of this subsection. Consideration shall be given to forces imposed by each of the following loads individually, and where applicable a vector summation of any combination thereof:

(1) Dynamic loading under all product load configurations.
(2) Internal pressure.
(3) Superimposed loads such as operating equipment, insulation, linings, hose tubes, cabinets and piping.
(4) Reactions of supporting lugs and saddles or other supports.
(5) Effect of temperature gradients resulting from product and ambient temperature extremes. Thermal coefficients of dissimilar materials where used should be accommodated.

§178.340-5 Joints.

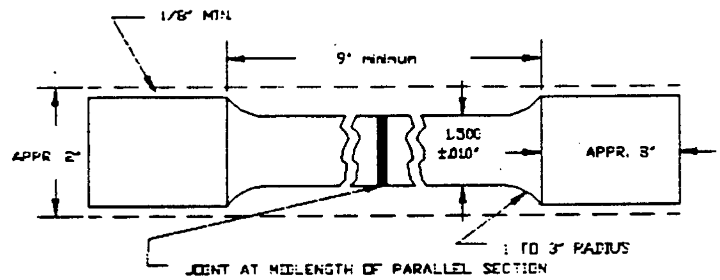
(a) Method of joining. All joints between tank shells, heads, baffles (or baffle attaching rings), and bulkheads shall be welded in accordance with the requirements contained in this section.

(b) Strength of joints (Aluminum Alloy (AL)). All welded aluminum alloy joints shall be made in accordance with recognized good practice, and the efficiency of a joint shall be not less than 85 percent of the properties of the adjacent material. Aluminum alloys shall be joined by an inert gas arc welding process using aluminum-magnesium type of filler metals which are consistent with the material suppliers recommendations.

(c) Strength of joints (Mild Steel(MS), High Strength Low Alloy (HSLA), Austenitic Stainless Steel (SS)). Joints shall be welded in accordance with recognized good practice and the efficiency of any joint shall be not less than 85 percent of the mechanical properties of the adjacent metal in the tank.

(1) Combinations of mild steel (MS), high strength low alloy (HSLA) and/or austenitic stainless steel (SS), may be used in the construction of a single tank, provided that each material, where used, shall comply with the minimum requirements specified in §178.340-3(a) for the material used in the construction of that section of the tank. Whenever stainless steel sheets are used in combination with steels of other types of steel, joints made by welding shall be formed by the use of stainless steel electrodes or filler rods and the stainless steel electrodes or filler rods used in the welding shall be suitable for use with the grade of stainless steel concerned, according to the recommendations of the manufacturer of the stainless steel electrodes or filler rods.

(d) Compliance test. Compliance with the requirements contained in paragraph (b) or (c) of this subsection for the welded joints indicated in paragraph (a) of this subsection shall be determined by preparing from materials representative of those to be used in tanks subject to this specification and by the same technique of fabrication, 2 test specimens conforming to figure as shown below and testing them to failure in tension. One pair of test specimens may represent all the tanks to be made of the same combination of materials by the same technique of fabrication, and in the same shop, within six months after the tests on such samples have been completed. The butt welded specimens tested shall be considered qualifying other types or combinations of types of weld using the same filler material and welding process as long as parent metals are of the same types of material.



§178.340-6 Supports and anchoring.

(a) Cargo tanks with frames not made integral with the tank as by welding, shall be provided with restraining devices to eliminate any relative motion between the tank and frame which may result from the stopping, starting or turning of the vehicle. Such restraining devices shall be readily accessible for inspection and maintenance, except that insulation and jacketing are permitted to cover the restraining devices.

(b) Any cargo tank designed and constructed so that it constitutes in whole or in part the structural member used in lieu of a frame, shall be supported in such a manner that the resulting stress levels in the cargo tank do not exceed those specified in §178.340-4(a). The design calculations of the support elements shall include loadings imposed by stopping, starting and turning in addition to those imposed as indicated in §178.340-4(b) using 20 percent of the minimum ultimate strength of the support material.

§178.340-7 Circumferential reinforcement.

(a) Tanks with shell thicknesses less than 3/8 of an inch shall in addition to the tank heads be circumferentially reinforced with either bulkheads, baffles, or ring stiffeners. It is permissible to use any combination of the aforementioned reinforcements in a single cargo tank.

(1) Location. Such reinforcement shall be located in such a manner that the maximum unreinforced portion of the shell be as specified in Table II of the applicable specification and in no case more than 60 inches. Additionally such circumferential reinforcement shall be located within one inch of points where discontinuity in longitudinal shell sheet alignment exceeds 10 degrees unless otherwise reinforced with structural members capable of maintaining shell stress levels permitted in §178.340-4(a).

(b) Baffles. Baffles or baffle attaching rings if used as reinforcement members shall be circumferentially welded to the tank shell. The welding must not be less than 50 percent of the total circumference of the vessel and the maximum unwelded space on this joint shall not exceed 40 times the shell thickness.

(c) Double bulkheads. Tanks designed to transport different commodities which if combined during transit will cause a dangerous condition or evolution of hear or gas shall be provided with compartments separated by an air space. This air space shall be vented and be equipped with drainage facilities which shall be kept operative at all times.

(d) Ring stiffeners. Ring stiffeners when used to comply with this section shall be continuous around the circumference of the tank shell and shall have a section modulus about the neutral axis of the ring section parallel to the shell at least equal to that determined by the following formula:

I/C(Min)=0.00027WL (MS, HSLA, and SS) Steel
I/C(Min)=0.000467WL (AL) Aluminum Alloy

where:

I/C= Section modulus (inches^3).
W = Tank width or diameter (inches)
L = Ring spacing (inches):i.e., the maximum distance from the midpoint of the unsupported shell on one side of the ring stiffener to the midpoint of the unsupported shell on the opposite side of the ring stiffener.

(1) If a ring stiffener is welded to the tank shell (with each

circumferential weld not less than 50 percent of the total circumference of the vessel and the maximum unwelded space on this joint not exceeding 40 times the shell thickness) a portion of the shell may be considered as part of the ring section for purposes of computing the ring section modulus. The maximum portion of the shell to be used in these calculations is as follows:

Circumferential ring stiffener to tank shell welds	Distance between parallel circumferential ring stiffener to shell welds	Shell section credit
1	--	20t
2	Less than 20t	20t+W
2	20t or more	40t

where:

t = Shell thickness

W = Distance between parallel circumferential ring stiffener to shell welds.

(2) If configuration of internal or external ring stiffener encloses an air space, this air space shall be arranged for venting and be equipped with drainage facilities which shall be kept operative at all times.

§178.340-8 Accident damage protection.

(a) Appurtenances: The term "appurtenance" means any cargo tank accessory attachment that has no liquid product retention or other liquid containment function, and provides no structural support to the tank.

(1) The design, construction, and installation of any appurtenance to the shell or head of the cargo tank must be such as to minimize the possibility of appurtenance damage or failure adversely affecting the product retention integrity of the tank.

(2) Structural members, such as the suspension subframe, overturn protection and external rings, when practicable, should be utilized as sites for attachment of appurtenances and any other accessories to a cargo tank.

(3) Except as prescribed in subparagraph (5) of this paragraph, the welding of any appurtenance to a shell or head must be made by attachment to a

mounting pad. The thickness of a mounting pad must not be less than that of the shell or head to which it is attached. A pad must extend at least 2 inches in each direction from any point of attachment of an appurtenance. Pads must have rounded corners or otherwise be shaped in a manner to preclude stress concentration on the shell or head. The mounting pad must be attached by a continuous weld around the pad.

(4) The appurtenance must be attached to the mounting pad so there will be no adverse affect upon the product-retention integrity of the tank if any force is applied to the appurtenance, in any direction, except normal to the tank, or within 45° of normal.

(5) Shirting structures, conduit clips, brakeline clips, and similar lightweight attachments, which are of a metal thickness, construction, or material, appreciably less strong but not more than 72 percent of the thickness of the tank shell or head to which such a device is attached, may be secured directly to the tank shell or head if each device is so designed and installed that damage to it will not affect the product retention integrity of the tank. These lightweight attachments must be secured to the tank shell by continuous weld or in such manner as to preclude formation of pockets, which may become sites for incipient corrosion.

(b) Rear bumpers. Every cargo tank shall be provided with a rear bumper to protect the tank and piping in the event of a rear end collision and minimize the possibility of any part of the colliding vehicle striking the tank. The bumper shall be located at least 6 inches to the rear of any vehicle component which is used for loading or unloading purposes or may at any time contain lading while in transit. Dimensionally, the bumper shall conform to 49 CFR 393.86. Structurally, the bumper shall be designed to successfully absorb (no damage which will cause leakage of product) the impact of the vehicle with rated payload, with a deceleration of 2 "g" using a factor of safety of two based on the ultimate strength of the bumper material. For purposes of these regulations such impact shall be considered uniformly distributed and applied horizontally (parallel to the ground) from any direction at an angle not exceeding 30° to the longitudinal axis of the vehicle.

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(c) Overturn protection. All closures for filling, manhole or inspection openings shall be protected from damage which will result in leakage of lading in the event of overturning of the vehicle by being enclosed within the body of the tank or dome attached to the tank or by guards.

(1) When guards are required, they shall be designed and installed to withstand a vertical load of twice the weight of the loaded tank and a horizontal load in any direction equivalent to one-half the weight of the loaded tank. These design loads may be considered independently. Ultimate strength of the material shall be used as a calculation base. If more than one guard is used each shall carry its proportionate share of the load. If protection other than guards are considered the same design load criteria is applicable.

(2) Except for pressure actuated vents no overturn protection is required for non-operating nozzles or fittings less than five inches in diameter (which do not contain product while in transit) that project a distance less than the inside diameter of the fitting. This projected distance may be measured either from the shell or the top of an adjacent ring stiffener provided such stiffener is within 30 inches of the center of the nozzle or fitting.

(3) If the overturn protection is so constructed as to permit accumulation of liquid on the top of the tank, it shall be provided with drainage facilities directed to a safe point of discharge.

(d) Piping.

(1) Product discharge piping shall be provided with protection in such a manner as to reasonably assure against the accidental escape of contents. Such protection may be provided by:

(i) A shear section located outboard of each emergency valve sear and within 4 inches of the vessel which will break under stain and leave the emergency valve sear and its attachment to the vessel and the valve head intact and capable of retaining product. The shear section shall be machined in such a manner as to abruptly reduce the wall thickness of the adjacent piping (or valve) material by at least 20 percent; or

(ii) By suitable guards capable of successfully absorbing a concentrated horizontal force of at least 8000 pounds applied from any horizontal direction, without damage to the discharge piping which will adversely affect the product retention integrity of the discharge valve.

(2) Minimum road clearance. The minimum allowable road clearance of any cargo tank component or protection device located between any two adjacent axles on a vehicle or vehicle combination shall be at least 1/2 inch for each foot separating such axles and in no case less than 12 inches.

(3) Strength of piping, fittings, hose and hose couplings. Hose, piping and fittings for tanks to be unloaded by pressure shall be designed for a bursting pressure of at least 100 psig and not less than four times the pressure to which,

in any instance, it may be subjected in service by the action of any vehicle mounted pump or other device (not including safety relief valves), the action of which may be to subject certain portions of the tank piping and hose to pressures greater than the design pressure of the tank. Any coupling used on hose to make connections shall be designed for a working pressure not less than 20 percent in excess of the design pressure of the hose and shall be so designed that there will be no leakage when connected.

(4) Provision for expansion and vibration. Suitable provisions shall be made in every case to allow for and prevent damage due to expansion, contraction, jarring and vibration of all pipe. Slip joints shall not be used for this purpose.

(5) Heater coils. Heater coils, when installed, shall be so constructed that the breaking-off of their external connections will not cause leakage of contents of tank.

(6) Gauging, loading, and air-inlet devices. Gauging, loading and air-inlet devices, including their valves, shall be provided with adequate means for their secure closure, and means shall also be provided for the closing of pipe connections of valves.

On May 2, 1989, §178.340-8(b) was revised to read as follows:

§178.340-8 Accident damage protection.

(b) *Rear-end Protection.* Each cargo tank shall be provided with a rear accident damage protection device to protect the tank and piping in the event of a rear-end collision and reduce the likelihood of damage which could result in the loss of lading. The rear-end protection device must be in the form of a rear bumper or rear-end tank protection device meeting the following:

(1) *Rear bumper.* (i) The bumper shall be located at least 6 inches to the rear of any vehicle component used for loading or unloading or that may contain lading while the vehicle is in transit.

(ii) The dimensions of the bumper shall conform to §393.86 of this title.

(iii) The structure of the bumper shall be designed to withstand, without leakage of lading, the impact of the vehicle with rated payload, at a deceleration of 2 "g" using the safety factor of two based on the ultimate strength of the bumper material. Such impact shall be considered uniformly distributed and applied horizontally (parallel to the ground) from any direction at any angle not exceeding 30 degrees to the longitudinal axis of the vehicle.

(2) *Rear-end tank protection device.* (Noting in this paragraph shall be construed to relieve a manufacturer of responsibility for complying with the requirements of §393.86 of this title.)

(i) The inboard surface of the rear-end tank protection device shall be located at least 6 inches to the rear of any vehicle component used for loading or unloading or that may contain lading while the vehicle in is transit, in order to prevent the device from applying force upon the cargo tank or tank components in the event of an accident.

(ii) The dimensions of the rear-end protection device shall conform to the following:

(A) The bottom surface of the rear-end protection device must be at least 4 inches below the lower surface of any valve, fitting, or piping at the rear of the tank and not more than 60 inches from the ground with the vehicle empty.

(B) The maximum width of a notch, indentation, or separation between sections of a rear-end tank protection device may not exceed 24 inches. A notched, indented, or separated rear-end protection device may be used only when the piping at the rear of the tank is equipped with a sacrificial device outboard of a shutoff valve. (a sacrificial device is an element, such as a shear section, designed to fail under load in order to prevent damage to any lading retention part or device. The device must break under strain at no more than 70 percent of the strength of the weakest piping element between the tank and the sacrificial device. Operation of the sacrificial device must leave the remaining piping and its attachment to the tank intact and capable of retaining lading.)

(C) The widest part of the motor vehicle at the rear may not extend more than 18 inches beyond the outermost ends of the device or (if separated) devices on either side of the vehicle.

(iii) The structure of the rear-end tank protection device and its attachment to the vehicle must be designed to withstand without leakage of lading, the impact of the cargo tank motor vehicle at rated payload, at a deceleration of 2 "g" using a safety factor of two based on the ultimate strength of the materials used. Such impact shall be considered uniformly distributed and applied horizontally (parallel to the ground) from any direction at an angle not to exceed 30 degrees to the longitudinal axis of the vehicle.

§178.341 Specification MC 306; cargo tanks

§178.341-1 General requirements.

(a) Specification MC 306 cargo tanks must comply with the general design and construction requirements in §178.340 in addition to the specific requirements contained in this section.

(b) Design pressure. The design pressure of each cargo tank shall not be less than that pressure exerted by the static head of the fully loaded tank in the upright position.

§178.341-2 Thickness of shell, heads, bulkheads and baffles.

(a) Material thickness. The minimum thicknesses of tank material authorized in §178.340-3 shall be predicated on not exceeding the maximum allowable stress level (§178.340-4(a) but in no case less than those indicated in Tables I and II below.

(1) Product density. The material thicknesses contained in Tables I and II are minimums based on a maximum 7.2 pounds per gallon product weight. If the tank is designed to haul products weighing more than 7.2 pounds per gallon, the gallon per inch value used to determine the minimum thickness of heads, bulkheads, baffles or shell sheets shall be the actual section capacity required in gallons per inch multiplied by the actual product density in pounds per gallon divided by 7.2.

Table I.--Minimum Thickness of Heads, Bulkheads and Baffles. Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) in United States Standard Gauge--Aluminum (AL)--Expressed in Decimals of an Inch.

	Volume Capacity in Gallons Per Inch											
	10 or Less			Over 10 to 14			Over 14 to 18					
	MS	HSLA SS	AL	MS	HSLA SS	AL	MS	HSLA SS	AL	MS	HSLA SS	AL
Thickness--	14	15	.096	13	14	.109	12	13	.130	11	12	.151

Table II.--Minimum Thickness of Shell Sheets. (Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) in United States Standard Gauge--Aluminum Alloy (AL)--Expressed in Decimals of an Inch.

Maximum Shell Radius	Distance Between Bulkheads, Baffles or Ring Stiffener	Volume Capacity in Gallons Per Inch											
		10 or Less			Over 10 to 14			Over 14 to 18			18 and over		
		MS	HSLA SS	AL	MS	HSLA SS	AL	MS	HSLA SS	AL	MS	HSLA SS	AL
Less than 70"	36" or less------	14	16	.087	14	16	.087	14	15	.096	13	14	.109
	Over 36" to 54"--	14	16	.087	14	15	.096	13	14	.109	12	13	.130
	54" Thru 60"-----	14	15	.096	13	14	.109	12	13	.130	11	12	.151
70" or more Less than 90"	36" or less------	14	16	.087	14	15	.096	13	14	.109	12	13	.130
	Over 36" to 54"--	14	15	.096	13	14	.109	12	13	.130	11	12	.151
	54" Thru 60"-----	13	14	.109	12	13	.130	11	12	.151	10	11	.173
90" or more Less than 125"	36" or less------	14	15	.096	13	14	.109	12	13	.130	11	12	.151
	Over 36" to 54"--	13	14	.109	12	13	.130	11	12	.151	10	11	.173
	54" Thru 60"-----	12	13	.130	11	12	.151	10	11	.173	9	10	.194
125" or more	36" or less------	13	14	.109	12	13	.130	11	12	.151	10	11	.173
	Over 36" to 54"--	12	13	.130	11	12	.151	10	11	.173	9	10	.194
	54" Thru 60"-----	11	12	.151	10	11	.173	9	10	.194	8	9	.216

§178.342 Specification MC 307; cargo tanks.

§178.342-1 General requirements.

(a) Specification MC 307 cargo tanks must comply with the general design and construction requirements in §178.340, in addition to the specific design requirements contained in this section.

(b) The design pressure (maximum allowable working pressure) of each cargo tank shall be not less than 25 psig. For working pressures in excess of 50 psig, the tank must be designed in accordance with the requirements of the ASME Code.

(c) Tanks shall be of circular cross-section.

§178.342-2 Thickness of shell, heads, bulkheads, and baffles.

(a) Material thickness. The minimum thickness of tank material authorized by §178.340-3 shall be not less than those obtained by applying the following formulas nor less than those specified in Table I and II below:

Thickness of shell = $T_s = PD/2SE_s$

Thickness of heads = $T_h = 0.885PL/SE_h$ (for pressure on concave side only)

Where:

T_s = Maximum thickness of shell material, exclusive of allowance for corrosion or other loadings;
T_h = Minimum thickness of head material, after forming, exclusive of allowance for corrosion and other loadings;
P = Design pressure, pounds per square inch;
D = Inside diameter of shell, inches;
L = Inside crown radius of head, inches;
S = Maximum allowable stress value, pounds per square inch equals one-fourth of specified minimum ultimate tensile strength. (One-fourth of aluminum alloy's annealed minimum ultimate strength.);
E_s = Lowest efficiency of any longitudinal joint in shell.

(85% max.);

E_h = Lowest efficiency of any joint in head. (85% max.).

(1) The knuckle radius of the head shall not be less than three times the material thickness. The straight flange shall not be less than three times the material thickness for butt-welded heads.

(2) For heads with pressure on the convex side, the material thickness as obtained by the above formula shall be increased by 67 percent unless such heads are adequately braced to prevent excessive distortion.

(b) Corrosion allowance. Vessels or part of vessels subject to thinning by corrosion , erosion or mechanical abrasion shall have provision made to withstand the intended life and service by a suitable increase in the thickness of the material over that determined by the design formulas, or by using some other suitable method of protection. Material added for these purposed need not be of the same thickness for all parts of the vessel if different rates of attack are expected for the various parts.

Table I.--Minimum Thickness of Heads, Bulkheads and Baffles. Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) in United States Standard Gauge--Aluminum (AL) in Decimals of an Inch

	Volume Capacity in Gallons Per Inch																				
	10 or Less			Over 10 to 14			Over 14 to 18			18 to 22			22 to 26			26 to 30			30 and Over		
	ML	HSLA SS	AL	ML	HSLA SS	AL	ML	HSL A SS	AL	ML	HSLA SS	AL	ML	HSLA SS	AL	ML	HSLA SS	AL	ML	HSLA SS	AL
Thickness	14	15	.109	13	14	.130	12	13	.151	11	12	.173	10	11	.194	9	10	.216	8	9	.237

Table II.--Minimum Thickness of Shell Sheets. (Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) in United States Standard Gauge--Aluminum Alloy (AL)--Expressed in Decimals of an Inch.

Distance Between Bulkheads, Baffles or other Shell Stiffeners	Volume Capacity in Gallons Per Inch																				
	10 or Less			Over 10 to 14			14 to 18			18 to 22			22 to 26			26 to 30			30 and Over		
	MS	HSLA SS	AL	MS	HSLA SS	AL	MS	HSLA SS	AL	MS	HSLA SS	AL	ML	HSLA SS	AL	ML	HSLA SS	AL	ML	HSLA SS	AL
36" or Less----------	14	16	.109	14	16	.109	14	15	.109	13	14	.130	12	13	.151	11	12	.173	10	11	.194
Over 36" to 54"------	14	16	.109	14	15	.109	13	14	.130	12	13	.151	11	12	.173	10	11	.194	9	10	.216
54" Thru 60"---------	14	15	.109	13	14	.130	12	13	.151	11	12	.173	10	11	.194	9	10	.216	8	9	.237

§178.343 Specification MC 312; cargo tanks.

§178.343-1 General requirements.

(a) Specification MC 312 cargo tanks must comply with the general design and construction requirements in §178.340 in addition to the specific requirements contained in this section.

(b) Tank design. Cargo tanks built under this specification that are unloaded by pressure in excess of 15 psig must be designed and constructed in accordance with and fulfill all requirements of the ASME Code. No tank shall have head, bulkheads, baffles or shell thicknesses less than that specified in §178.343-2, Tables I and II, nor shall the spacing of bulkheads, baffles or shell stiffeners exceed that specified in §178.340-7.

(c) Design pressure shall be not less than pressure used for unloading.

§178.343-2 Thickness of shell, heads, bulkheads and baffles of the non-ASME Code tanks.

(a) Material thickness. The minimum thicknesses of tank material authorized in §178.340-3 shall be predicated on not exceeding the maximum allowable stress level in §178.340(a) but in no case less than those indicated in Tables I and II listed below, or the accompanying aluminum alloy formula:

(1) Aluminum alloy formula:

$$\text{Thickness of Aluminum Alloy Materials} = \text{Steel Thickness from Tables I \& II} \times (3\mathrm{x}10^7/E)^{1/3}$$

Where E = Modulus of Elasticity of the material to be used.

(b) Lining. Except as provided in paragraph (c) of this subsection, cargo tanks shall be lined and the material used for lining each cargo tank subject to this specification shall be homogenous, nonporous, imperforate when applied, not less elastic than the metal of the tank proper, and substantially immune to attach by the commodities to be transported therein. It shall be directly bonded or attached by other equally satisfactory means. Joints and seams in the lining shall be made by fusing the material together, or by other equally satisfactory means.

(c) Conditions under which tanks need not be line. Tanks need not be lined as provided in paragraph (b) of this subsection, if:

(1) The material of the tank is thick enough to withstand 10 years normal service-without being reduced at any point to less thickness than that specified in paragraph (a) of this subsection corresponding to its type; or,

(3) The chemical reaction between the material of the tank and the commodity to be transported therein is such as to allow the tank to be properly passivated or neutralized and if the tank is not frequently cleaned and not used in the transportation of other commodities.

Table I.--Minimum Thickness of Heads, Bulkheads and Baffles. Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) in United States Standard Gauge--unless otherwise expressed in fractions of an inch

	Volume Capacity in Gallons Per Inch											
	10 or Less			Over 10 to 14			14 to 18			18 and Over		
	Product Weight in Pounds Per Gallon @ 60° F											
	10 lbs and Less	Over 10 to 13 lbs	13 lbs to 16	10 lbs and Less	Over 10 to 13 lbs	13 lbs to 16	10 lbs and Less	Over 10 to 13 lbs	13 lbs to 16	10 lbs and Less	Over 10 to 13 lbs	13 lbs to 16
Thickness-	12	10	8	10	8	3/16	9	3/16	1/4	8	1/4	1/4

Table II.--Minimum Thickness of Shell Sheets. (Mild Steel (MS), High Strength Low Alloy Steel (HSLA), Austenitic Stainless Steel (SS) in United States Standard Gauge--unless otherwise expressed in fractions of an inch.

Maximum Shell Radius	Distance Between Bulkheads, Baffles or Ring Stiffener	Volume Capacity in Gallons Per Inch											
		10 or Less			Over 10 to 14			14 to 18			18 and over		
		Product Weight in Pounds Per Gallon @ 60° F											
		10 lbs. and Less	Over 10 to 13 lbs.	13 lbs. to 16	10 lbs. and Less	Over 10 to 13 lbs.	13 lbs. to 16	10 lbs. and Less	Over 10 to 13 lbs.	13 lbs. to 16	10 lbs. and Less	Over 10 to 13 lbs.	13 lbs. to 16
Less than 70"	36" or less------	12	10	8	12	10	8	12	10	8	10	8	3/16
	Over 36" to 54"--	12	10	8	12	10	8	10	8	3/16	9	3/16	1/4
	54" Thru 60"-----	12	10	8	10	8	3/16	9	3/16	1/4	8	1/4	1/4
70" or more Less than 90"	36" or less------	12	10	8	12	10	8	10	8	3/16	9	3/16	1/4
	Over 36" to 54"--	12	10	8	10	8	3/16	9	3/16	1/4	8	1/4	1/4
	54" Thru 60"-----	10	8	3/16	9	3/16	1/4	8	1/4	1/4	3/16	1/4	5/16
90" or more Less than 125"	36" or less------	12	10	8	10	8	3/16	9	3/16	1/4	1/4	8	1/4
	Over 36" to 54"--	10	8	3/16	9	3/16	1/4	8	1/4	1/4	3/16	1/4	5/16
	54" Thru 60"-----	9	3/16	1/4	8	1/4	1/4	3/16	1/4	5/16	3/16	1/4	5/16
125" or more	36" or less------	10	8	3/16	9	3/16	1/4	8	1/4	1/4	3/16	1/4	5/16
	Over 36" to 54"--54"	9	3/16	1/4	8	1/4	1/4	3/16	1/4	5/16	3/16	1/4	5/16
	Thru 60"-----	8	1/4	1/4	3/16	1/4	5/16	3/16	1/4	5/16	3/16	5/16	3/8

APPENDIX F. STRUCTURAL EVALUATION OF MC 306/307/312 CARGO TANKS.

MC regulations for 300 series cargo tanks do not give detailed load combinations or analysis procedures for evaluating structural integrity of cargo tanks but do give detailed loadings for overturn protection and rear-end protection. Also, some 307/312 cargo tanks must be designed in accordance with the ASME Code.

As a result of lack of detailed requirements for non-ASME, MC 300 series cargo tanks, manufacturers have developed dynamic load factors and analysis procedures through their experiences.

For an MC 306 cargo tank, structural integrity of the tank is frequently based on analysis of the tank idealized as a simple beam subjected to static gravity weights of the tank, lading and appurtenances. Bending stresses computed for this situation are then multiplied by a factor of 1.7 to account for "dynamic effects". (This would be the same as the computation of bending stresses due to extreme vertical acceleration for an MC 406 cargo tank.) This is considered to be a surrogate for vertical acceleration, lateral acceleration, braking, etc. The computed tensile stress is compared to that allowed by the regulations (20% of ultimate). Weld joint efficiencies must be at least 85% and are already accounted for in the allowable stress value. Computed compressive stress is compared to allowable values from the Alcoa or Roark and Young formulas. Minimum shell thickness must conform to values given in the regulations. Ring stiffener spacing and section modulus must also conform to the regulations. The design of many MC 306 cargo tanks has been based on this simplified approach.

The analysis/design of non-ASME, MC 307/312 cargo tanks would also be based on the approach described above.

Structural analysis procedures for MC 307/312 cargo tanks required to meet portions of the ASME Code, would be the same as those for DOT 407/412 cargo tanks except for differences in the MC and DOT Regulations.