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Nelson et al.

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[54] **LARGE VOLUME TANKS**

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[52] **U.S. Cl.** ..... **220/5 A**; 220/1 B; 220/72

[51] **Int. Cl.** **B65d 7/02**; B65d 7/42

[58] **Field of Search** ..... 220/5 A, 71, 1 B, 72; 217/95

[56] **References Cited**

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3,471,053	10/1969	Endicott et al.	220/5 A

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Attorney, Agent, or Firm—Harold N. Wells

[57] **ABSTRACT**

Large storage tanks are made possible by placing support bands around shell courses which otherwise would be too thin to support the design load. The location of the support bands takes advantage of the resistance of the shell courses to bending in order to provide a nearly fully-stressed design as contrasted to the normal design which does not fully utilize available metal thickness.

4 Claims, 4 Drawing Figures

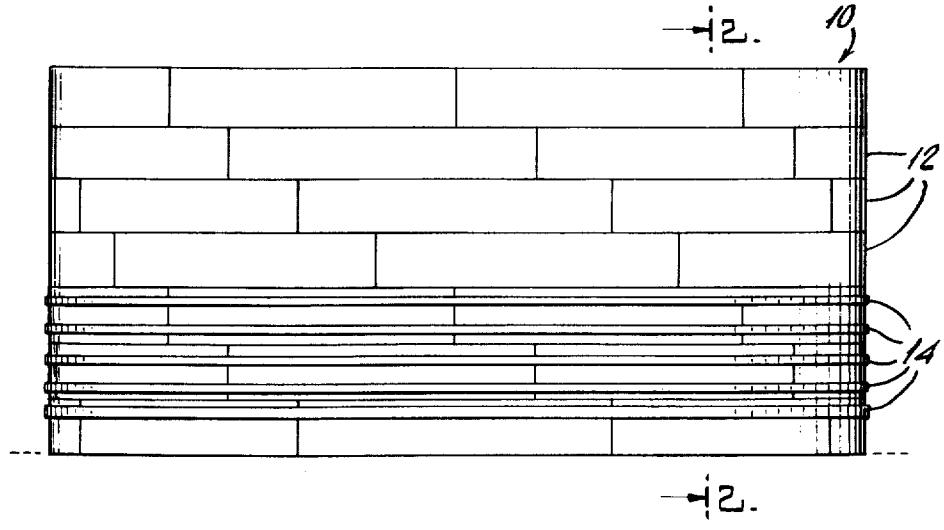


Fig. 1.

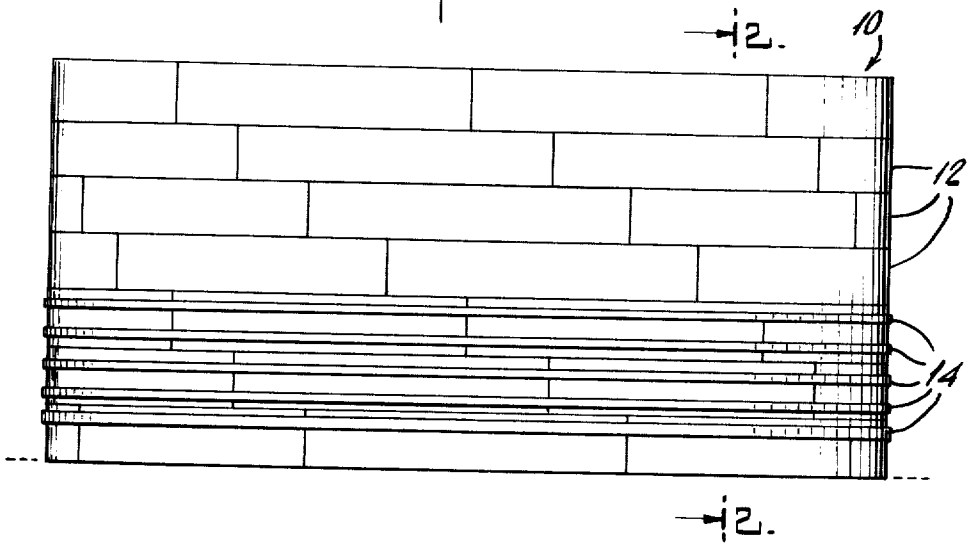


Fig. 2.

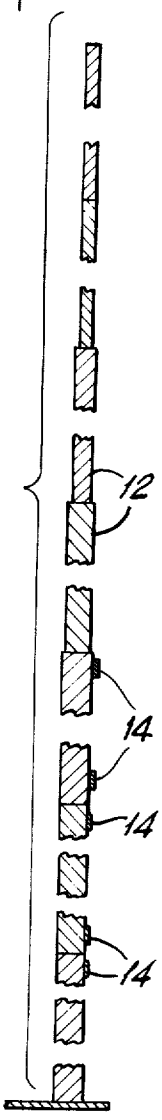


Fig. 3.

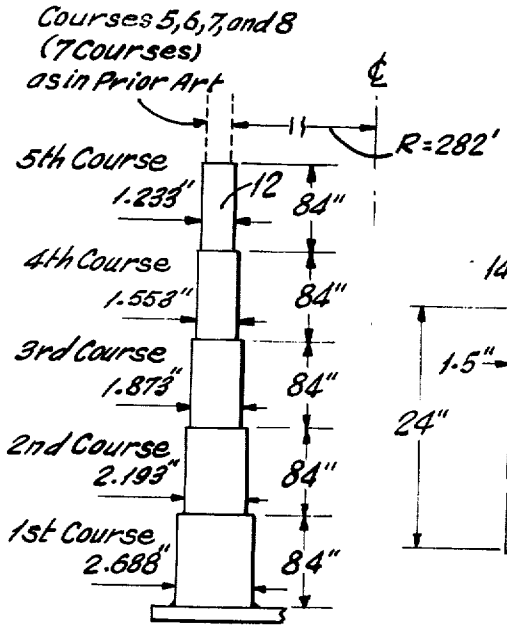
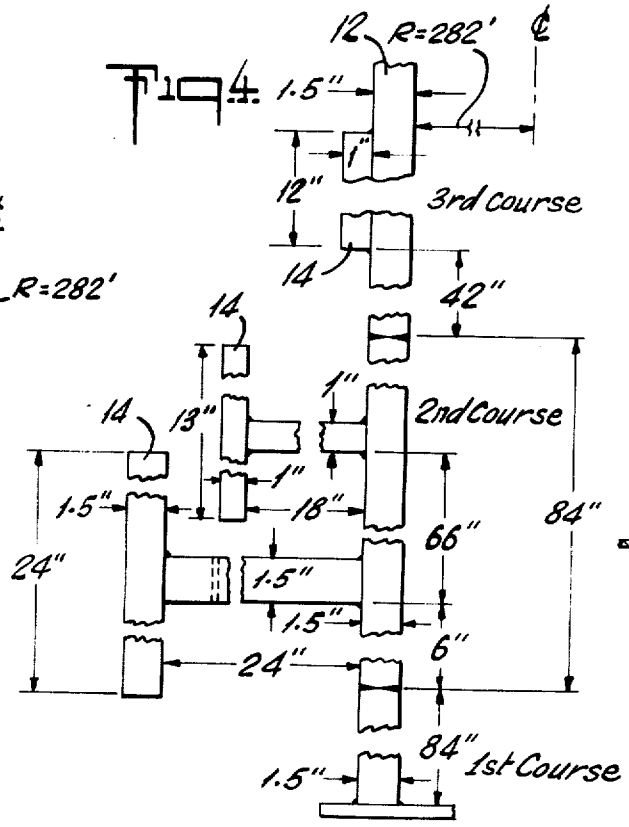


Fig. 4.



LARGE VOLUME TANKS

BACKGROUND OF THE INVENTION

This invention pertains principally to large tankage such as is commonly used in the petroleum industry. Since the size of crude oil tankers is continually increasing, it is highly desirable to have larger tanks to handle the shipments received from these exceptionally large vessels. Tanks have grown in average size over the years until today their size is effectively limited by the available technology, thus setting an upper limit to the size of the tanks. The cost per barrel of tankage is essentially the same beyond 400,000 bbls., making it appear that there is no advantage in making tanks any larger. While the cost per barrel for tankage may not indicate any advantage for larger tanks, there are sizable associated savings. As the cost of industrial real estate increases, the savings obtained from consolidating tankage becomes important. In addition, eliminating duplicate facilities which are associated with multiple tankage makes substantial savings possible by using fewer tanks. Thus, the advantages for making tanks larger remain even though the erected cost of the bare tankage does not change. Inasmuch as the total cost of tankage which is purchased by the petroleum industry is very large, anything that can be done to reduce the cost has a substantial impact on the cost of new facilities.

As has been stated heretofore, the size of tankage currently available is limited by present technology. The principal limitation is the ability to weld very thick shell plates in the field. Once a thickness of about 1-½ inches has been reached, it is necessary to stress relieve welds in the field. Since this is practically impossible, the size of tanks is limited to those having a thickness of no more than 1.5 inches, thereby leading to a practical capacity limit for large tankage. The present invention presents a technique by which this upper limit may be raised, making possible large tankage which has heretofore been impossible to construct.

Storage tanks have been described in the prior art, for example, see U.S. Pat. No. 3,471,053. Briefly summarizing, cylindrical storage tanks are built of successive rings of shell plates, commonly called courses, typically 6–10 ft. high. Each course has a uniform thickness. Inasmuch as the fluid pressure to be resisted by the tank walls decreases linearly with vertical height, it is possible to reduce the thickness of the courses. Thus, the bottom course has the greatest thickness, the next vertically adjacent course has a lesser thickness and so on until the thickness of the upper course is normally limited by other structural considerations rather than the hydrostatic pressure. Such designs do not make the maximum advantage of the metal thickness which is used since, while the pressure which must be resisted varies linearly, the thickness does not. In order to obtain the proper thickness at the points of highest stress in each course, it is necessary to apply more metal in that same course where less could serve. It is, of course, impractical under industrial conditions to fabricate plates of continuously varying thickness to make what would be, theoretically, a more nearly correct design. Such a design would also take advantage of the restraining effect of the tank bottom in order to reduce wall thickness. This latter effect has been used in the prior art reference previously mentioned.

Normally tanks in the petroleum industry are designed according to American Petroleum Institute (API) Standard 650, "Welded Steel Tanks for Oil Storage." Using this standard and typical tank steels, the practical upper limit for this design is about one million barrels, where the lower course reaches the practical limit of thickness, 1-½ inches. The erected cost of a barrel of tankage is almost constant for tanks above 400,000 barrels, while below this level the cost per barrel increases as the volume of the tank decreases. In the larger sizes, the additional costs for preparing, shipping, handling and field welding extremely heavy steel plate offsets the inherent cost advantage that would be expected for larger single tanks as opposed to multiple smaller ones. However, the cost savings in land and accessory equipment are such as to make the larger tanks desirable even though their erected cost per barrel is essentially the same.

The prior art has suggested several ways to build large tanks, although the problem itself is of relatively recent importance. The previously referred to U.S. Pat. No. 3,471,053 attempts to take advantage of the stiffening effect of the attachment of the tank walls to the bottom by reducing the thickness of the bottom course and placing a restraining band near the top of this course. This would allow tanks of up to about 1,500,000 bbls. to be built. Earlier patents such as U.S. Pat. No. 2,975,927 and U.S. Pat. No. 2,433,335 provided novel designs which attempted to avoid the limitations referred to, but which inherently present more difficult construction problems.

What has been needed, but not fully provided by the prior art, is a tank design which is no more expensive than the current designs, but which permits the breaking of the practical size barrier which has heretofore limited the size of tankage and building tanks of 3,000,000 bbls. capacity or more. The present invention applies an inherently simple technique which has been found to permit the use of thinner plate and in fact to reduce the cost of tankage by completely utulizing this tank metal. Thus, the cost per barrel may be reduced in spite of the fact that larger tanks are produced and a somewhat more complex structure is required.

SUMMARY OF THE INVENTION

Large storage tanks can be produced without exceeding the practical weld thickness limitation by reducing the course thickness and then positioning external bands around those courses so as to support them. It has been found that the inherent stiffness of thick plates for large tanks permits sufficiently wide spacing so that the number of bands required to support the tank courses is not excessive and the cost of overall construction may be reduced. Thus, a very close approach to a fully stressed design is made, which makes possible larger tanks than are now available. By proper selection of band thickness and placement in relation to the course materials, it is possible to use less steel by taking advantage of the improved material properties which are inherent with thin plate, to reduce the amount of welding required, and to improve weld quality. While the technique is primarily intended for use in large tanks such as are typical in the petroleum industry, alternatively it may be applied to chemical service where expensive materials are required to meet corrosion or contamination requirements, so that a minimum

of expensive material could be used while being supported structurally by external banding.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 illustrates a side view of a large storage tank utilizing the present invention.

FIG. 2 is a section taken substantially along line 2—2 of FIG. 1 illustrating the application of the present invention.

FIG. 3 illustrates a form of external band which may be applied.

FIG. 4 shows an alternative form of the external banding.

DESCRIPTION OF THE PREFERRED EMBODIMENT

FIG. 1 shows an elevation view of a typical tank 10 having support bands 14 according to the present invention. This tank contains seven courses 12, each approximately 8–10 feet in height. A typical diameter for this tank would be about 500 ft. The capacity of such a tank would be approximately 2,000,000 bbls. Inasmuch as the thickness of the bottom courses exceeds the API standards thickness limit of 1-½ inches, bands have been applied, one to the bottom course, two to the second course, and two to the third course in order to reduce the thickness of those courses and to stay within practical limitations of shell thickness. The band on the lower course acts in much the same way as that of the prior art reference, U.S. Pat. No. 3,471,053, in that it takes advantage of the restraint provided by the welding of the bottom plate to the bottom of the tank. By placing a band near the upper portion of the lower course, it is possible to restrain the lower course in cooperation with the restraint provided by the bottom weld.

As has been previously discussed, the normal designs produce an understressed structure since each course must be designed with relation to the maximum pressure applied to that course, whereas in fact this requires more metal than necessary at the upper portion of the course.

It has been found according to the present invention that by providing bands at regions of high stress, it is possible to use less metal than would be required with a uniform shell thickness since maximum advantage can be taken of the ability to locate additional support only where it is needed. Such an approach would not be practical were it not for the fact that the portion of each course which is unsupported does not deform to the extent which might be expected when the bands are rather widely spaced (typically about 4 ft.). The inherent stiffness of shell courses for large tanks permits the bands to be separated without bulging of the shell between. This effect permits the bands to be widely spaced, minimizing the amount of support material which is necessary. Such an approach is appropriate principally for the lower and thicker shell courses, where the bulge between the support bands can be restricted by the inherent stiffness of the shell. The advantages of providing support bands include the following:

1 by utilizing thinner base shell courses it is possible to take advantage of the better material properties inherent in the thinner plate, which means that higher allowable stresses may be used, reducing the shell thickness still further;

2 a closer approximation is made to the theoretical distribution of metal required to suit the nonuniform loading by the fluid contained within the tank and less metal is required to achieve a satisfactory design than is required when uniform shell courses are used; and

3 difficult deep penetration welding in the field is reduced due to the use of thinner shell plates, thus improving the quality of the welds and reducing their cost. In addition, the welding required for application of the external bands can be mainly done in the shop prior to shipping to the field. The amount of field welding is reduced and is much less difficult than that required for the shell welding.

FIG. 2 illustrates a section along line 2—2 of FIG. 1 illustrating both the stepdown design of the shell courses in the typical tank and also the external banding according to the invention. It will be observed that the support band is placed near the upper portion of the lower course in order to support that point and to coact with the restraint provided by the bottom which is normally at the point of highest pressure. With larger sizes, the bands may begin above the bottom course. On courses above the lowest, it is desirable to place support bands so as to maximize the amount of unsupported shell between them as influenced by the support of the adjacent courses. Typically this will require two bands in each of the courses above the bottom one inasmuch as they are not affected by the restraint provided by the bottom of the tank. Utilizing these principles, calculations may be made by conventional analysis techniques in order to optimize the location and thickness and dimensions of these supporting bands.

EXAMPLE

A large tank having a 2.5 million barrel capacity is given as an example of the application of the invention. FIG. 3 shows a tank designed according to API Standard 650. FIG. 4 shows the same tank designed according to this invention. In both figures the tank has a radius of 282 feet and an overall height of 56 feet consisting of 8 courses, each 7 feet in height. It will be seen in FIG. 3 that the lower four courses exceed the maximum allowable thickness under the API Standard of 1.5 inches. The lower course thicknesses must be reduced. FIG. 4 illustrates the lower three courses on which supporting bands have been used in order to make possible the thinning of the tank wall to 1-½ inches. Two novel aspects are shown in this figure; first, the lowest course does not require a supporting band in this example, and, second, the supporting bands are not flat bands as shown in FIG. 2, but are T-shaped members which extend outwardly from the tank wall. Either type of support may be used, as well as other shapes, selected for economy and other practical design considerations. It should be noted that while the support bands are shown here as welded directly to the tank wall, which is the most likely method of application for this type of band, it is also within the scope of the invention that the support bands rest upon clips welded independently to the tank wall, which would be particularly useful for purposes of supporting the flat bands of FIG. 2.

The positioning and sizing of the support bands was made according to the following description of the general calculational approach:

The lower courses are assumed to be 1.5 in. thick, or some other acceptable thickness. The first stiffener is

placed near the circumferential joint between the first two courses generally in the region where the stress in the assumed structure would most exceed the allowable stress. If the courses have the same allowable stress, this is the region in which the radial deflection of the tank wall is greatest.

Since the tank bottom prevents the bottom of the wall from displacing radially, and imposes its plastic hinge moment onto the shell wall, the deflection of a point on the lower courses of the tank is:

$$\frac{w}{p_o R^2/ET_o} = \frac{T_o}{T_1}\left\{1 - \frac{BZ}{BL} - e^{-BZ}(\cos BZ + \hat{M}_p \sin BZ)\right\} + \frac{Z}{L}$$

Where:

Z = Distance up tank wall from tank bottom
w = Radial deflection outward
p_0= Radial pressure outward at the bottom
R = Tank radius
E = Young's Modulus
T_o= Calculated thickness of unstiffened bottom course
T_1= Actual thickness of bottom course
L = Combined height of courses requiring reinforcement
v = Poisson's ratio
$B = \{3(1-v^2)\}^{1/4}/\sqrt{RT_1}$
M_p= (plastic hinge moment of tank bottom plate) × $2B^2/p_o$

If the courses have different allowable stresses, as permitted in API Standard 650, and the point of maximum stress is found to be in one of the courses of higher allowable stress, the stiffener could be enlarged and positioned nearer the course of lower allowable stress to further reduce its radial deformation.

The number (n) of stiffeners needed in addition to the lowest stiffener depends on the height (h_c) at which the actual and calculated thicknesses are equal, the size of the tank as indicated by the parameter B, and the position (h_b) of the lowest stiffener. The limits on n of

$$.7(h_c - h_b)B < n < (h_c - h_b)B$$

give the practical range of the additional number of stiffeners needed. It has been found, further, that equal spacing of the stiffeners between the positions h_c and h_b yields close to the optimum stiffener distribution.

The cross-sectional area (A) of a stiffener reinforcing the tank wall from height h_1 up to height h_u is approximately

$$A = \int_{h_1}^{h_u} (T_c - T_a)\,dh$$

Where: h = distance up tank wall

$$T_c = \frac{(\text{pressure at } h) \times (\text{tank radius})}{(\text{allowable stress at } h)}$$

T_a = actual wall thickness at h

This area is typically only 50 percent to 75 percent of the area which would be required as additional course thickness according to the prior art. Thus, significant metal savings are made possible by this invention.

The beneficial influence of the tank bottom on reinforcing the bottom of the tank wall has been found to reduce the area required in the lowest stiffener by about one-third. The area (A_1) of the lowest stiffener is therefore about

$$A_1 = \frac{2}{3}\int_0^{h^*} (T_c - T_a)\,dh$$

where h^* is a point about midway between the lowest two stiffeners.

The above procedure gives a first approximation of the optimum spacing and sizing of the stiffeners of this invention. The next step in the design of a tank using this invention should be the stress analysis of the design resulting from this first approximation. Many suitable shell analysis computer programs are currently available. Slight modifications in the design based upon such analysis should yield a very close approximation of the optimum design using this invention.

This procedure was followed in designing the example tank. The stresses found in this analysis are within the allowable limits of API Standard 650.

The foregoing description is of the preferred embodiments. Changes may be made without departing from the spirit of the invention as defined by the claims which follow.

What is claimed is:

1. A cylindrical storage tank of the type having a circular bottom plate attached at its circumference to a cylindrical wall extending perpendicularly upward from the plane of the bottom and comprising a series of ring courses of uniform thickness attached at their edges to adjacent ones of said courses, wherein the improvement comprises:

a. a support band secured about the lowest of said courses at a point where the radial deflection is greatest as defined by the equation,

$w/p_oR^2/ET_o = T_o/T_1\ \{1 - BZ/BL - e^{-BZ}(\cos BZ + \hat{M}_p \sin BZ)\} + Z/L$

where:

Z = Distance up tank wall from tank bottom
w = Radial deflection outward
p_o= Radial pressure outward at the bottom
R = Tank radius
E = Young's Modulus
T_o= Calculated thickness of unstiffened bottom course
T_1= Actual thickness of bottom course
L = Combined height of courses requiring reinforcement
v = Poisson's ratio
$B = \{3(1-v^2)\}^{1/4}/\sqrt{RT_1}$
M_p= (plastic hinge moment of tank bottom plate × $2B^2/po$

b. at least one additional support band secured about a ring course above said lowest course and thinner than said lowest course, the number of said bands being between 0.7 $(h_c - h_b)B$ and $(h_c - h_b)B$,

where:

h_c= height of the tank where the actual and calculated thickness are equal
h_b= height of the band of (a).

2. The tank of claim 1 wherein the area of the band of (a) is determined by the equation:

$$A = \frac{2}{3}\int_0^{h^*} (T_c - T_a)\ dh$$

where:

h = height from tank bottom

h* = height midway between lowest two stiffeners

T_c = (pressure at h) x (tank radius)/allowable stress at h

T_a = actual wall thickness at h.

3. The tank of claim 1 wherein the area of the bands of (b) are determined by the equation:

$$A = \int_{h_1}^{h_u} (T_c - T_a)\ dh$$

where:

h_u = maximum height defining area supported by band h_1 = minimum height defining area supported by band

4. A tank of claim 1 wherein the greatest radial deflection of (a) is small enough so that the allowable stress is not exceeded and support bands are required only above the lowest course.

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