Jan. 12, 1960 H. C. BOARDMAN 2,920,784

LIQUID STORAGE VESSEL

Original Filed March 22, 1952

4 Sheets-Sheet 1

Fig. 1.

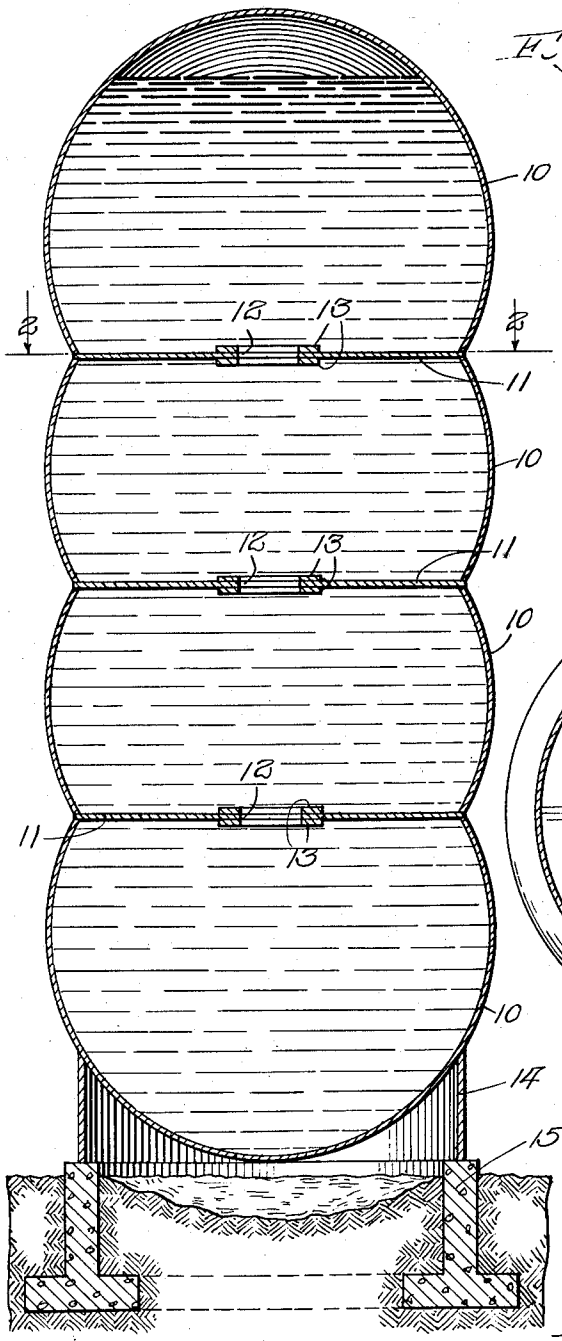
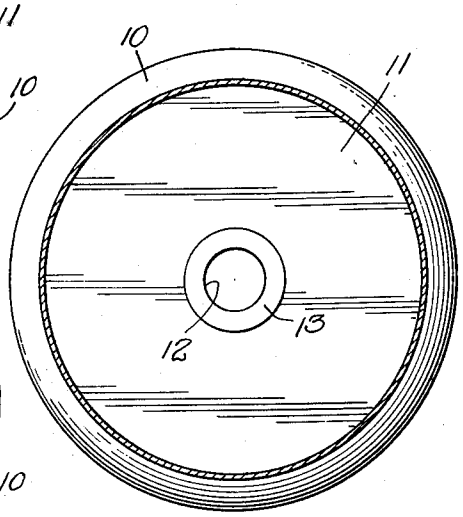



Fig. 2.



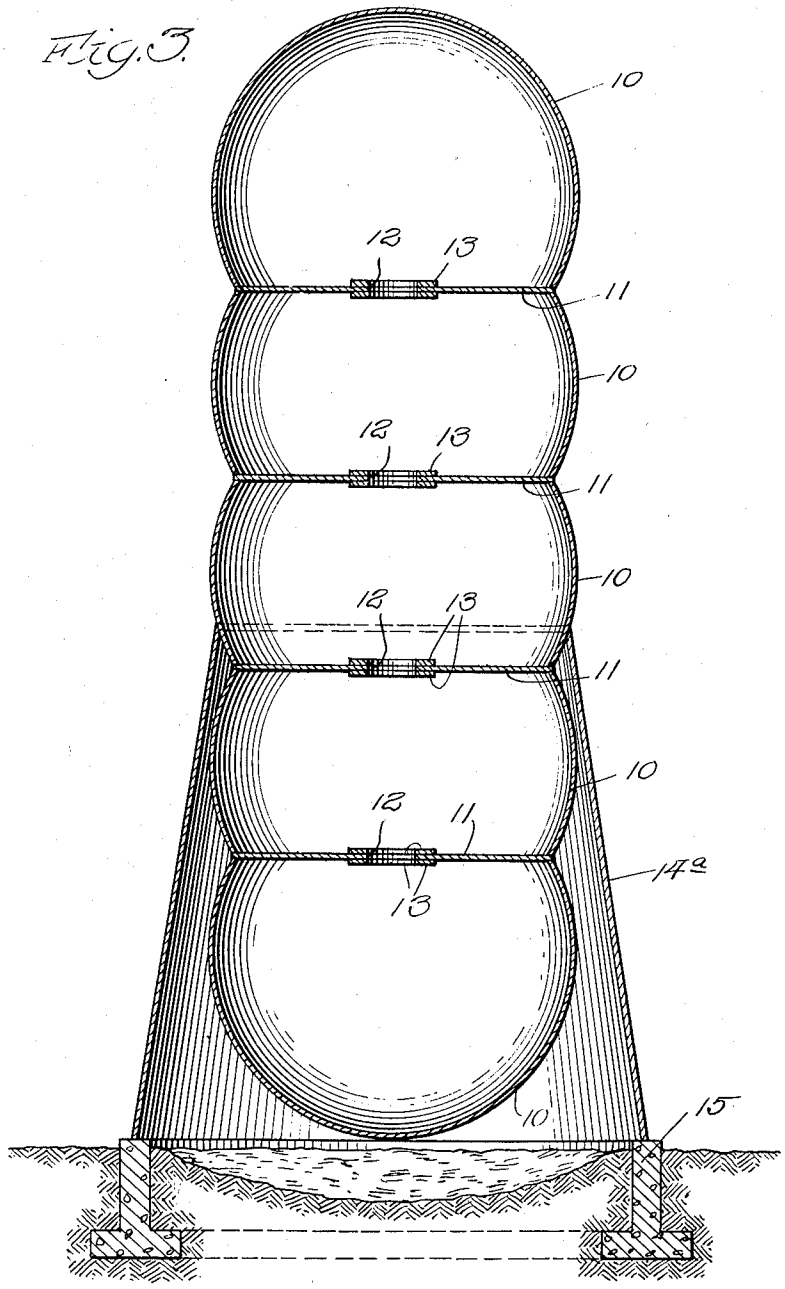

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LIQUID STORAGE VESSEL

Original Filed March 22, 1952 4 Sheets-Sheet 2

Fig. 3.




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LIQUID STORAGE VESSEL

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Fig. 4.

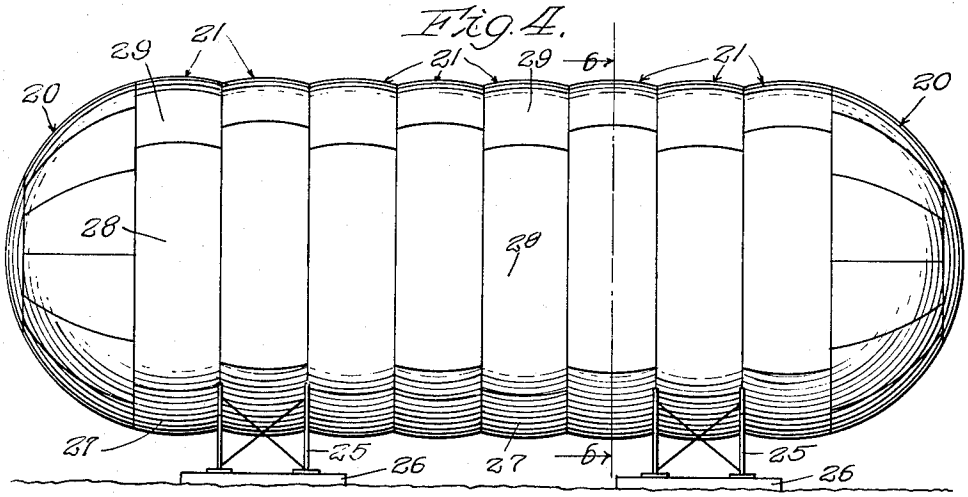



Fig. 5.

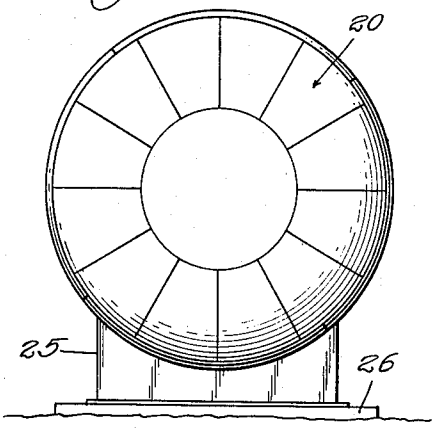
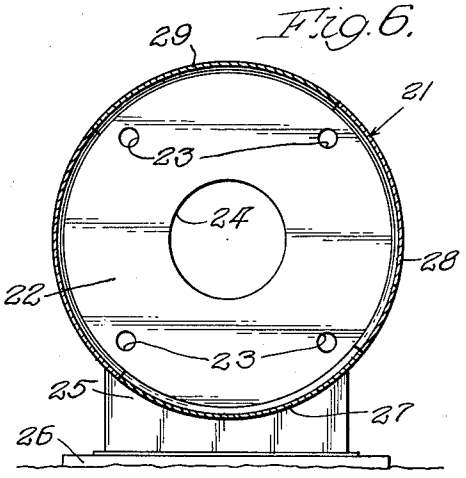



Fig. 6.



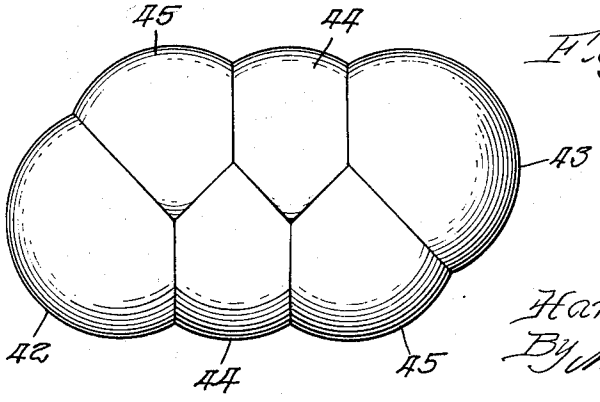




Fig. 11.

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Original Filed March 22, 1952

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Fig. 7.

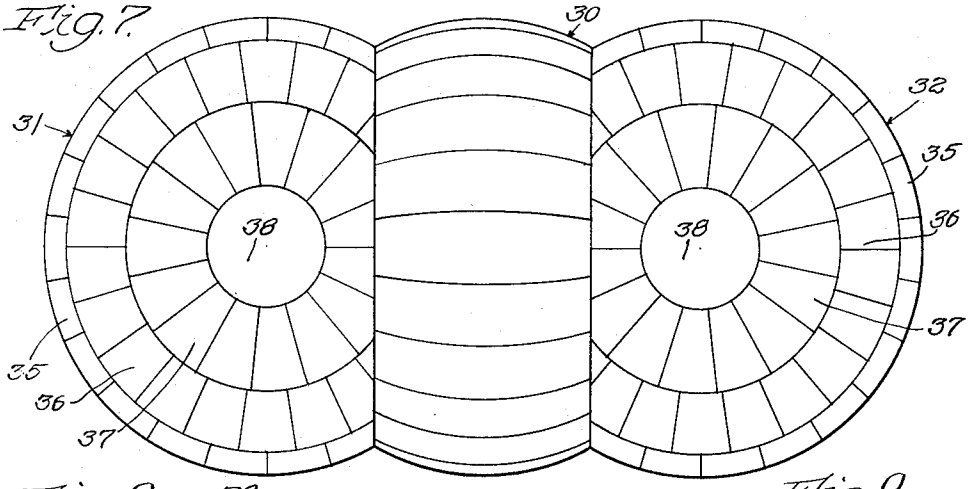



Fig. 8.

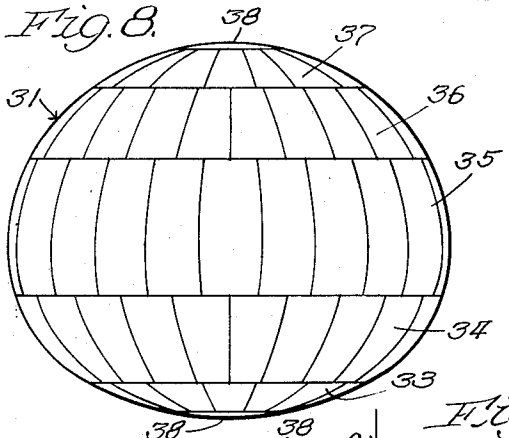



Fig. 9.

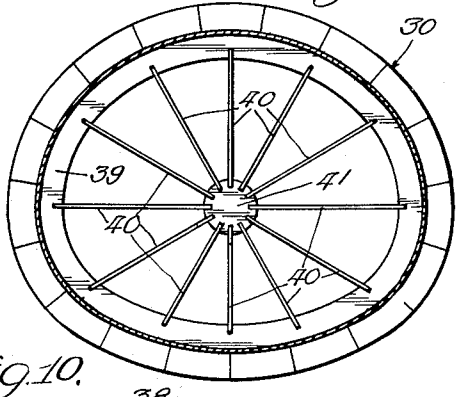
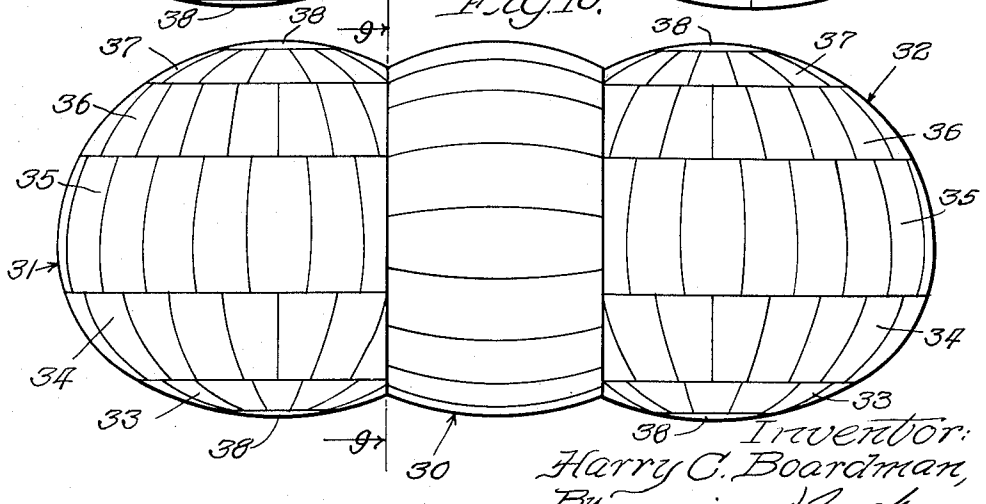



Fig. 10.




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United States Patent Office 2,920,784 Patented Jan. 12, 1960



2,920,784

# LIQUID STORAGE VESSEL

Harry C. Boardman, Chicago, Ill., assignor to Chicago Bridge & Iron Company, a corporation of Illinois

Continuation of application Serial No. 278,037, March 22, 1952. This application December 1, 1955, Serial No. 550,407

1 Claim. (Cl. 220—1)

This invention relates to storage vessels for the storage of liquids.

This application is a continuation of my copending application, Serial No. 278,037, filed March 22, 1952, now abandoned, which is a continuation-in-part of my copending application, Serial No. 608,884, filed August 4, 1945, now issued as Patent No. 2,672,254.

Vessels of large size are widely used for the storage of various fluids including gases and liquids. When liquids are to be stored, the vessels must be strong enough to sustain the weight of the liquid as well as the vapor pressure of this liquid. In some instances the gas pressure may be controlling over the pressure caused by the liquid weight. As the size of the vessels increase, the thickness of the steel used in their construction becomes quite large. This may create the difficult problem of field stress relieve after welding plates having thicknesses over one and one quarter inches.

The storage vessel of the present invention overcomes the necessity of using plates having troublesome thicknesses by utilizing segments of spherical or spheroidal form joined together at intersections of the segments and provided with diaphragms secured to the segments in the planes of intersection. The segments are formed of courses of plates dished to the proper curvature and the plates have a thickness less than required for a single sphere or spheroid of the same capacity adapted for the same storage. The vessel segments should be aligned horizontally and supported in a manner consistent with the intended use of the vessel.

Specific embodiments of the invention will be described with reference to the accompanying drawings in which:

Figure 1 is a vertical section taken through a vertically aligned storage vessel constructed according to the method of this invention;

Figure 2 is a horizontal sectional view taken substantially along line 2—2 in Figure 1;

Figure 3 is a vertical sectional view taken through the center of a vessel substantially like that shown in Figure 1 showing a different type of support;

Figure 4 is a side elevational view of a horizontally aligned vessel of the invention having sphere segments;

Figure 5 is an end view of the vessel shown in Figure 4;

Figure 6 is a vertical sectional view taken substantially along line 6—6 in Figure 4;

Figure 7 is a top plan view of a vessel having spheroidal segments;

Figure 8 is an end view of the vessel shown in Figure 7;

Figure 9 is a sectional view through the vessel of Figure 7 taken substantially along line 9—9 in Figure 10;

Figure 10 is a side elevational view of the vessel of Figure 7; and

Figure 11 is a top plan of a much larger vessel on a reduced scale showing spheroidal segments joined together in a plan somewhat different from that of Figures 7–10.



In the storage vessel of this invention, the thickness of the metal plates used is kept to a minimum to reduce cost, welding, and other problems. In accomplishing this in a vertically aligned sphere segment vessel with diaphragms positioned at the intersection of each pair of adjacent sphere segments, the vessel has sides that are thinner than the sides of a single sphere of the same capacity adapted for the same storage pressure. This permits the building of a larger vessel for a given thickness of metal plates.

In order to make the most economical use of metal, the storage vessels of this invention are constructed according to certain mathematical formulae that I have found are applicable to this type of construction.

In the embodiments shown in Figures 1–3, the vessel comprises intersecting sphere segments **10** arranged along a vertical axis with diaphragms **11** having circular cutout portions **12** in their centers with reinforcements around the edges of the cutout portions. These reinforcements may consist of circular flat plates **13** arranged one on top and one on the bottom of the section of diaphragm surrounding each hole. These plates are welded or otherwise attached to the diaphragms. The entire vessel is supported by a cylindrical support **14** or frusto conical support **14***a* resting on a foundation **15**.

Although the vessel is shown constructed of sphere segments having equal radii of curvature, this is not required, as the radii may differ one from the other. The intersecting sphere segments, each of which has a constant radius of curvature, may be replaced by members having different radii of curvature in different planes.

In order to make the most economical use of construction material, I have developed mathematical formulae for the thickness of each segment and each diaphragm; this makes possible the fabrication of a storage vessel of a given capacity with the smallest amount of construction material possible. The development of these mathematical formulae will be presented in the discussion following.

The basic equation for curved surfaces under load:

$$\frac{T_1}{R_1}+\frac{T_2}{R_2}=P \tag{1}$$

expresses equilibrium of radially inward and outward forces when $T_1$ equals latitude stress in pounds per inch of meridian arc, $T_2$ equals meridian stress in pounds per inch of latitude arc, $R_1$ equals radius of curvature in inches at right angles to a meridian plane, $R_2$ equals radius of curvature in inches in a meridian plane, and P equals combined gas and liquid pressure in pounds per square inch. If the radius of curvature of the surface is constant for each individual shell the $R_1$ equals $R_2$ equals R, and Equation 1 becomes

$$T_1+T_2=PR \tag{1a}$$

and

$$T_1=PR-T_2=R\left(P-\frac{T_2}{R}\right) \tag{2}$$

$$T_2=PR-T_1=R\left(P-\frac{T_1}{R}\right) \tag{3}$$

The point of greatest stress in a spherical shell above the support is just above its bottom diaphragm. The weight of liquid above the diaphragm per square inch of diaphragm equals

$$\frac{W}{\pi h^2}$$

where W equals the total weight of liquid above the diaphragm and $h$ equals the radius of the diaphragm in

inches. This unit weight of liquid is designated herein as L.

In all the equations set out herein the diaphragm area is taken as the overall area covered by the diaphragm and bounded by its circumference. No allowance is made for any cutout areas of the type shown in the drawings, as these will ordinarily be reinforced at their margins.

If all above the diaphragm is considered a free body, the equilibrium equation of vertical forces at the diaphragm will be

$$2\pi hT_2\left(\frac{h}{R}\right)=P\pi h^2-W \tag{4}$$

when $P\pi h^2$ equals the total pressure at the diaphragm, Equation 4 becomes

$$T_2=\tfrac{1}{2}R\left(P-\frac{W}{\pi h^2}\right) \tag{5}$$

$$T_2=\tfrac{1}{2}R(P-L) \tag{6}$$

The horizontal component of $T_2$ is equal to T, the diaphragm stress in pounds per inch of the periphery of the diaphragm. Then

$$T=\tfrac{1}{2}a(P-L)+\tfrac{1}{2}b(P-L)=\tfrac{1}{2}V(P-LQ \tag{7}$$

where $a$ equals distance from the diaphragm in inches to the center of curvature of its top spherical shell, $b$ equals distance from the diaphragm to the center of curvature of its bottom shell, and the total distance between centers of curvature $(a+b)$ equals V. Substituting in Equation 2 the value of $T_2$ from Equation 6 there is obtained the expression

$$T_1=R\left[P-\frac{\tfrac{1}{2}R(P-L)}{R}\right] \tag{8}$$

or,

$$T_1=\tfrac{1}{2}R(P+L) \tag{9}$$

When the spherical shells and diaphragms are below the support, then the weight of liquid below the diaphragm becomes the important factor. Here

$$L_1=\frac{W_1}{\pi h^2}$$

where $W_1$ is the weight of the liquid below the diaphragm and $h$ is the radius of the diaphragm. The equations for stresses below the support become

$$T=\tfrac{1}{2}V(P+L_1) \tag{10}$$

$$T_1=\tfrac{1}{2}R(P-L_1) \tag{11}$$

$$T_2=\tfrac{1}{2}R(P+L_1) \tag{12}$$

For all points above the shell to which the support is attached the greatest stress in the spherical shell will be just above the diaphragm located at the bottom of the shell. For all spherical shells below the shell to which support is attached the greatest stress will be just below the diaphragm located at the top of the shell.

For each shell above the shell to which the support is attached the latitude stress $(T_1)$ is always greater than the meridian stress $(T_2)$. For each shell below the shell to which the support is attached the meridian stress $(T_2)$ is greater than the latitude stress $(T_1)$. This can be seen by inspection of the Equations 6 and 9 and of Equations 11 and 12. Therefore above the shell to which the support is attached, the latitude stress $(T_1)$ governs the design, while below the shell to which the support is attached the meridian stress $(T_2)$ governs the design.

The analysis of stresses given herein for a vessel constructed according to my methods shows that the stresses vary over the entire vessel. As a practical matter a large storage vessel of vertical shells is seldom made with sides varying in thickness from one level to another on the vessel. Therefore in constructing this improved storage vessel each shell is made of uniform thickness,



and each diaphragm is made of uniform thickness, but the shells vary in thickness one from the other, and the diaphragms vary in thickness one from the other. In designing spherical shells to be used above the shell to which the support is attached the latitude stress $(T_1)$ at a level just above the diaphragm at the bottom of the shell is used as the design basis. For shells to be used below the shell to which the support is attached the meridian stress $(T_2)$ at a level just below the diaphragm at the top of the shell is used as the design basis. For the diaphragms the stress (T) at the periphery of the diaphragm is used as the design basis.

The thickness of the shells and diaphragms may be found by dividing the design stress by the product of the allowable working stress and the joint efficiency. The allowable working stress is determined by the type of construction material used, as well as other factors well understood by one skilled in the art. The joint efficiency depends upon the type of joint used in building the vessel. This efficiency factor has been assigned to the different types of joints, such as welded joints, riveted joints, and the like. This, too, is well understood by the man skilled in the art.

From the foregoing it is evident that for shells and diaphragms above the shell to which the support is attached the thicknesses are determined by dividing Equations 9 and 7, respectively, by the product of the allowable working stress and the joint efficiency. For shells and diaphragms below the shell to which the support is attached, the thicknesses are found by dividing Equations 12 and 10 respectively by the product of allowable working stress and joint efficiency. The thickness equation for shells above the shell to which the support is attached becomes

$$A=\frac{R(P+L)}{2SE} \tag{13}$$

and for diaphragms above the shell to which the support is attached becomes

$$B=\frac{V(P-L)}{2SE} \tag{14}$$

which should be compared with

$$B=\frac{Vp}{2SE} \tag{15}$$

where $p$ is the gas pressure alone, and the larger value of B used in the design. In the ordinary normal storage vessel Equation 15 will always be greater than Equation 14. The thickness for shells below the shell to which the support is attached becomes

$$A_1=\frac{R(P+L_1)}{2SE} \tag{16}$$

and for diaphragms below the shell to which the support is attached becomes

$$B_1=\frac{V(P-L_1)}{2SE} \tag{17}$$

where A and $A_1$ equal shell thickness above and below the shell to which the support is attached, respectively, and B and $B_1$ equal diaphragm thickness above and below the shell to which the support is attached respectively. S equals the allowable working stress, and E equals the joint efficiency.

When the support is on an intermediate shell, as shown in Figure 3, Equation 9 should be applied to both the level of the upper diaphragm and the level of the support; and Equation 12 should be applied to both the level of the lower diaphragm and the level of the support. At the level of the support W is the weight of liquid above the support, and $W_1$ the weight of liquid below the support, and $h$ is the horizontal radius of the shell in the plane of the support. The greatest stress given by these four equations is the design stress for this shell.

When the support is on the bottom shell, as shown in

Figure 1, Equation 9 should be applied to both the level of the upper diaphragm and the level of the support; Equation 12 should be applied to the level of the support, and to the lowest point on the bottom where $L_1$ equals zero. The greatest stress given by these four equations is the design stress for this shell.

In the normal storage vessel constructed according to the method of this invention, the shells are progressively thinner the farther they are above and below the support; the diaphragms above the support are of equal thickness, and the diaphragms below the support are progressively thinner the farther they are below the support. When the storage vessel is constructed in its preferred form with the support on the lowermost shell, with the radius of each shell the same and with the distances between centers of adjacent shells the same, the diaphragms will all be the same thickness and this thickness will be determined by means of Equation 15.

Each individual shell of a series of vertical shells and each diaphragm is preferably of uniform thickness. It is also preferred that each shell have a constant radius of curvature, although the radii of curvature of all the shells will not necessarily be the same. Preferably the storage vessel is constructed with the support connected to the vessel at a circle determined by a horizontal plane intersecting the vessel. In order to provide economy of material in constructing the support, it may conveniently be located at a horizontal circle on the bottom portion of the lowermost shell. This means that the support is joined to the lowermost shell at or below the equator of this shell.

It will be noted from a consideration of Equations 13 and 16 that the allowable working stress (S) and joint efficiency (E) are both constant, as in the factor L for the critical cross-section. This results in a simplified formula of which it can be said that, for any course of plates forming a segment of a vessel, the quotient of the controlling stress per inch of cross-sectional arc and the thickness is a constant. This is true for the sphere segments and also for the spheroidal segments. In the latter, the shape of the vessel is much more adapted to the storage of liquid wherein the pressure causing the controlling stress is due to the weight of the liquid more than to the gas pressure. Ideally, the spheroids, which are generally of the shape shown and described in detail in Horton Patent No. 1,778,944, issued October 21, 1930, are constructed with a plate thickness uniform throughout the vessel. This ideal condition would be fully realized if it were possible to keep the spheroid full of the normal liquid under the normal vapor pressure; in fact, because of partial liquid loadings and variable vapor pressure, the lower portion of the shell is somewhat thicker than the upper portion. Nevertheless, the radii of curvature are so varied from top to bottom of the vessel that, at the critical level in each course of plates forming a segment of the shell, the quotient of the controlling stress and the thickness is a constant.

A form of the invention particularly adapted for storage of liquid and gas at a substantially high pressure is illustrated in Figures 4–6. Herein the segments are spherical in form and the vessel has a spherical head **20** at each end of a plurality of spherical segments **21** intermediate the heads **20**. A diaphragm **22** is positioned at the plane of intersection of each of the spherical segments and is provided with a number of pressure equalizing openings **23** and **24** to permit communication between the interiors of the segments. Ordinarily the diaphragm openings are reinforced so that the diaphragms may be designed for the total pressure over the whole diaphragm area without consideration of localized stresses due to the openings. The vessel may be supported on foundation plates **25** resting on prepared foundations **26**. Ordinarily the foundation plates are connected to the vessel at the intersection of adjacent spherical segments so that the thrust is taken upon the diaphragm as well as the shell.

The plate make-up of the spherical segments is such as to accommodate the weight of the liquid and thus a greater pressure or stress at the bottom of the vessel than at the top. It will be noted that each spherical segment has a bottom plate **27**, an intermediate plate **28** and a top plate **29** which are aligned substantially horizontally with similar plates of alternate spherical segments. The particular vessel shown in Figures 4–6 has a capacity of 10,000 barrels and a working pressure of 125 pounds per square inch. At the designed capacity and storage, the bottom and center plates **27** and **28** may be constructed with a thickness of approximately 1.10 inches and the top plate **29** can have a thickness of approximately 1.06 inches. It is preferred however to make the bottom plates 1.10 inches thick with a vertical distance of 4½ feet from the bottom, the center plates 1.09 inches thick, and the top plate 1.06 inches thick with a vertical distance of 6 feet from the top. The end spherical heads are constructed in orange peel fashion with the bottom plates of thicker material than the top plates. The thickness corresponds with the thickness of the plates in the spherical segments **21**.

The vessel illustrated in Figures 7–10 is more readily adapted for the storage of liquid and gas wherein the liquid creates the controlling stress on the vessel plates. Three spheroidal segments are shown joined together to form a vessel. Each segment has a radius in a vertical plane less than the radius of the same area in a plane normal to the vertical plane where both planes pass through a line normal to the point in the area in question. The vessel may be constructed with a plurality of intermediate sections **30** and with end sections **31** and **32** but for purposes of illustration only a three segment vessel is shown. The vertical and end views of Figures 10 and 8 respectively show that the plates of the end sections **31** and **32** are arranged in horizontal courses **33–37** with caps **38** on the top and bottom. The intermediate section **30** is also formed with plates arranged in horizontal courses. All of these plates are dished with a proper curvature to conform to the spheroidal shape of the segments and the courses preferably vary in thickness relatively the same as described in connection with the vessel of Figures 4–6.

Thus it will be seen that, while the thicknesses of the various courses may vary, the thickness of each course is constant. Therefore, under predetermined conditions of load, even though the unit stress varies from the top of any course to the bottom of that course because of the uniform thickness of the course, the unit stress at any given level in the course is a constant. Stated slightly differently, the maximum unit stress in each course is a constant.

Diaphragms are placed in the vessel at the intersection of the segments. The diaphragm may consist of a plate with openings therein for providing communication between the segments as described in connection with the vessel of Figures 4–6 or in the alternative may be constructed as shown in Figure 9 wherein an annular plate **39** is secured as by welding to the intersection of segments **30** and **31**. This plate extends inwardly and forms a support for a plurality of diametrically extending tie rods **40** which are connected at their inner ends to a ring **41**.

The particular vessel chosen for illustration in Figures 7–10 has a capacity of approximately 47,300 barrels and is intended to store liquid at a working pressure of 20 pounds per square inch. To give a clear picture of a size of such a vessel, it may suffice to say that the height of the vessel is approximately 50 feet, its width a little over 66 feet and its length slightly in excess of 133 feet. Ordinarily the vessel will be supported on a prepared foundation resting directly on the ground and conforming in shape to the lower portion of the vessel.

Considerably larger capacity vessels can be constructed by adding additional spheroidal segments such as is illustrated in Figure 11. Here, the end sections 42 and 43 form closures for one horizontally aligned pair of segments, which are designated 44, and another horizontally aligned pair of segments, designated 45. Additional spheroidal segments 44 and 45 may be added to extend the length of the vessel without increasing the thickness of the plates necessary in any one segment. In this manner a vessel having a virtually unlimited capacity for a given thickness of plate, may be easily and readily constructed.

The segmental vessels of Figures 1 and 2 may be constructed with each segment formed of plates having a single thickness in the smaller sizes and the segments mounted horizontally as illustrated in Figures 4, 5, and 6. Such vessels can be used for storage at considerably higher pressure for the given thickness of plate than for a single spherical vessel. Working pressures up to 1000 pounds per square inch may be accommodated in such vessels.

The foregoing detailed description has been given for clearness of understanding only, and no unnecessary limitations should be understood therefrom, for some modifications will be obvious to those skilled in the art.

I claim:

A liquid storage vessel having a shell comprising a plurality of horizontally aligned spheroidal segments joined along vertical planes to form a vessel, each segment being formed of courses of spheroidally curved plates joined together to form said vessel, the plate thickness of each spheroidal segment being less than that required for a spheroidal shell of the same capacity adapted for liquid storage, the plates of each course being of constant thickness and the thickness of the several courses being varied whereby under predetermined conditions of load the maximum unit stresses in all courses are equal, and diaphragms located at the vertical intersections of the spheroidal segments, each of said diaphragms comprising a plate annulus secured to the intersection of the joined segments and a plurality of tie rods extending diametrically of the plate annulus and secured at their ends to an annulus, said diaphragms being secured to the segments at their planes of intersection and said diaphragms having openings permitting intercommunication between said segments.

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UNITED STATES PATENT OFFICE

# CERTIFICATE OF CORRECTION

Patent No. 2,920,784 January 12, 1960

Harry C. Boardman

It is hereby certified that error appears in the printed specification of the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 3, line 25, equation (7) should appear as shown below instead of as in the patent:

$$T=\tfrac{1}{2}a(P-L)+\tfrac{1}{2}b(P-L)=\tfrac{1}{2}V(P-L)$$

Signed and sealed this 1st day of November 1960.

(SEAL)
Attest:

KARL H. AXLINE
Attesting Officer

ROBERT C. WATSON
Commissioner of Patents