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Design and Analysis of Storage Tank

Visal B¹, Sibin B²

P.G. Student, Department of Mechanical Engineering, SNGCE, Kadayiruppu, Kerala, India¹

Assistant Professor, Department of Mechanical Engineering, SNGCE, Kadayiruppu, Kerala, India²

ABSTRACT: Storage tanks are containers that hold liquids, compressed gases (gas tank) or mediums used for the short or long-term storage. Most storage tanks are designed according to American Petroleum Institute API-650 specification. These tanks can have different sizes, ranging from 2 to 60 m diameter or more. Here the primary consideration was given to the safety of tank under various circumstances. The framework of this project was to design a storage tank with the following specifications and its analyse;

Height = 12m, diameter = 8m, density of oil = 860 kg/m³.

A systematic investigation have done in terms of various design procedures for storage tank and here the design was done in Variable Design Point Method.

The Finite Element Analysis of storage tank was done using ANSYS software. The sloshing effect on storage tank was mainly considered in this project. The static and wind load analysis has also been done using ANSYS.

KEYWORDS: API-650, Finite Element Analysis, ANSYS, Sloshing, Wind load.

I. INTRODUCTION

Storage tanks are containers that hold liquids, compressed gases (gas tank) or mediums used for the short- or long-term storage of heat or cold. The term can be used for reservoirs (artificial lakes and ponds), and for manufactured containers.

Storage tanks operate under no (or very little) pressure, distinguishing them from pressure vessels. Storage tanks are often cylindrical in shape, perpendicular to the ground with flat bottoms, and a fixed or floating roof. There are usually many environmental regulations applied to the design and operation of storage tanks, often depending on the nature of the fluid contained within. Above ground storage tanks (AST) differ from underground storage tanks (UST) in the kinds of regulations that are applied.

There are three methods allowed by API 650 to determine the required plate thickness of the shell. The first method is the one-foot-method (1FM) which is based on the "membrane theory". The required shell plate thickness for each shell course is calculated using the circumferential stress at a point 0.3 m (1-ft) above the lower horizontal weld seam of the shell course due to hydrostatic pressure of the stored liquid. The reasoning behind this assumption is that the tank bottom plates provide restraint to reduce circumferential stress due to hydrostatic pressure at the bottom 0.3 m (1-ft) of the lowest shell course. Similarly, a shell course other than the lowest shell course, has generally thicker shell plates below. The plate below provides some restraint at the lower portion of the shell course in consideration. The 1FM is used successfully for the majority of the tanks. However, the designs based on the 1FM may become conservative and cost prohibitive for larger diameter tanks. Therefore, API 650 limits the applicability of this method to tanks up to 61 m (200-ft) in diameter.

The second method to calculate the required shell plate thickness is the variable-design-point method (VDPM) that is also based on the "membrane theory". The VDPM was proposed by Zick and McGrath in 1968 and later adopted by API 650 as a refined method to calculate the required shell plate thickness especially for tanks more than 61 m (200-ft) in diameter. The VDPM takes into consideration the restraint provided by the tank bottom plates to the first shell course and the restraint provided by each lower shell course to the upper shell course. The VDPM uses a variable distance instead of fixed distance of 0.3 m (1-ft), as used in 1FM, above the circumferential seam for each shell course to calculate the maximum stress due to hydrostatic pressure. The variable distance in VDPM is a function of the shell

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plate thickness above and below the seam. Most of the time designs based on VDPM are more economical compared with those based on the IFM. However, for some tank geometries the VDPM may become un-conservative and the tank shell thicknesses designed in accordance with VDPM may be overstressed. Buzek showed that the restraint provided by the tank bottom on the tank shell produces circumferential stresses of sinusoidal nature varying with the distance from the tank bottom. For certain tank diameter and height proportions, this sinusoidal varying restraining stress may add to the stress due to the hydrostatic circumferential stress and the design based on VDPM may become un-conservative. Therefore, API 650 limits the applicability of the VDPM for the tanks with L/H ratio less than 1000/6 in SI units (refer to the nomenclature for the definition of these terms). For the storage tanks where the L/H ratio is more than 1000/6, tank shell thickness should be determined using linear analysis.

II. RELATED WORK

1. During the last two decades of the 1800s, wooden barrels were replaced by riveted steel tanks to store petroleum as well as liquid chemical products. Because steel tanks give more strength than wooden barrels.
2. In 1964 at Niigata resulted in the loss of containment of several tanks due to damage sustained during the earthquake, which added to the ensuing inferno and continued to burn for 13 days. This incident highlighted several problems including that of floating roofs becoming dislodged and jamming, with the resulting fire being attributed to sparks from the damaged roof being shaken violently.
3. As the use of petroleum products increased, hydrocarbon producers and companies that install tanks soon created their own associations. In 1919, the American Petroleum Institute (API) was formed.
4. During the early days of steel tank production, riveting was the most common method of joining steel. In the 1920s and 1930s, many steel fabricators turned to arc welding, which led to higher quality tanks.
5. Petroleum Equipment Institute (PEI) was founded in 1951. Then API and PEI have developed several important storage tank system standards and guidelines that are widely respected in many countries.
6. In 1956, STI issued its Midwest 56 Standard, a widely-recognized method to designing the size and location of opening along the top of an underground storage tank.
7. In Naples, Italy 21st December 1985, reported an accident during a filling operation, fuel overflowed through the roof of a floating roof tank for almost an hour and a half. The spill was followed by a vapour cloud, which rapidly formed and ignited, the source of the ignition being a pumping station. The explosion resulted in the injury of five personnel, and the destruction of the facility.
8. The 1990s gave way to a completely new and unexpected trend – an extraordinary demand for aboveground storage tanks. Because of the negative connotations regarding expensive UST clean-ups.
9. In Mississippi, 5th June 2006, three contractors were killed and one was seriously injured in an explosion and fire at an oilfield. The contractors were stood on a gantry situated above four oil production tanks, preparing to weld piping, when it is assumed that a welding tool ignited flammable vapours from one of the tanks.

III. DESIGN PROCEDURE

STORAGE TANK DETAILS	
Tank capacity	88000 L
Tank diameter	79 m
Tank height	18 m
Height of each shell course	2 m
Specific gravity of product	1
Minimum thickness required as per safety norms	10 mm
Corrosion allowance	3 mm

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Tank material	SA 537 CLASS-1
Design stress, S_d	220 MPa
Hydrostatic stress, S_t	236 MPa

DESIGN OF SHELL PLATES

Check the ratio, $L/H \leq 1000/6$

We can use VDPM method only if this ratio is satisfied unless we must use linear analysis.

Bottom shell course

(a) Design condition

Bottom shell course thickness

$$t_{1d} = \left[1.06 - \frac{0.0696D}{H} \sqrt{\frac{HG}{S_d}} \right] \left[\frac{4.9HDG}{S_d} \right] + CA = 33.82$$

(b) Hydrostatic condition

$$t_{1t} = \left[1.06 - \frac{0.0696D}{H} \sqrt{\frac{H}{S_t}} \right] \left[\frac{4.9HD}{S_t} \right] = 28.8 \text{ mm}$$

$t_{1d} > t_{1t}$; so select t_{1d} as the shell thickness

ie $t_{1d} = 33.8 \approx 34 \text{ mm}$

Check the ratio

$$\frac{L}{H} \leq \frac{1000}{6}$$

$$61.48 \leq 166.67$$

Thus the ratio satisfies, so we can use VDPM method.

Divide the total height 18m into 9 equal steps. These steps are called shell course. So each shell course having 2m height.

Second shell course

(a) Design condition

$H = 16 \text{ m}$

$h_1 = 2 \text{ m}$

$r = 39.5 \text{ m}$

$t_{1d} = 34 \text{ mm}$

$CA = 3 \text{ mm}$

$t_1 = 31 \text{ mm}$

$G = 1$

Check the ratio,

$$\frac{h_1}{\sqrt{rt_1}}$$

where, h_1 = Height of the bottom shell course in mm

r = Tank radius in mm

t_1 = Calculated corroded thickness of bottom shell course in mm

if the value of the ratio is less than 1.375 $t_2 = t_1$

if the value of the ratio is greater than or equal to 2.625; $t_2 = t_{2a}$

if the value of the ratio is greater than 1.375 but less than 2.625

$$t_2 = t_{2a} + (t_1 - t_{2a}) \left[2.1 - \frac{h_1}{1.25\sqrt{rt_1}} \right]$$

where, t_2 = minimum design thickness of the second shell course

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t_{2a} = corroded thickness of the second shell course in mm

Checking

$$\frac{h_1}{\sqrt{rt_1}} = 1.725$$

$$1.375 < 1.725 < 2.625$$

So we applying equation

$$t_2 = t_{2a} + (t_1 - t_{2a}) \left[2.1 - \frac{h_1}{1.25\sqrt{rt_1}} \right]$$

To find t_{2a}

$$t_u = \frac{4.9D(H - 0.3)G}{S_d} = 27.62 \text{ mm}$$

$$K = \frac{t_L}{t_u} = 1.122$$

$$C = \frac{\sqrt{K}(K - 1)}{1 + K^{1.5}} = .059$$

$$x_1 = 0.61\sqrt{rt_u} + 320CH = 302.75 \text{ mm}$$

$$x_2 = 1000CH = 944 \text{ mm}$$

$$x_3 = 1.22 \times \sqrt{rt_u} = 40.3 \text{ mm}$$

$$t_{2a} = \frac{4.9D \left(H - \frac{x}{1000} \right) G}{S_d} = 28.08$$

Trials to find t_{2a}

t_u mm	K	C	x_1 mm	x_2 mm	x_3 mm	x mm	t_{2a} mm
27.62	1.122	0.059	302.75	944	40.3	40.3	28.08
28.08	1.10	0.048	266.075	768	40.63	40.63	28.08

Thickness of second shell course,

$$t_{2d} = t_{2a} + (t_1 - t_{2a}) \left[2.1 - \frac{h_1}{1.25\sqrt{rt_1}} \right] + CA = 31.98 \approx 32$$

(b) Hydrostatic condition

$$t_u = \frac{4.9D(16 - 0.3)}{S_t} = 25.75 \text{ mm}$$

$$K = \frac{t_L}{t_u} = 1.16$$

$$C = \frac{\sqrt{K}(K - 1)}{1 + K^{1.5}} = 0.077$$

$$x_1 = 413.69 \text{ mm}, x_2 = 1232 \text{ mm}, x_3 = 38.91 \text{ mm}$$

$$t_{2a} = \frac{4.9D \left(H - \frac{x}{1000} \right) G}{S_t} = 26.18$$

Trials to find t_{2a}

t_u mm	K	C	x_1 mm	x_2 mm	x_3 mm	x mm	t_{2a} mm
25.75	1.16	0.077	413.69	1232	38.91	38.91	26.18
26.18	1.146	0.07	378.02	1120	39.23	39.23	26.18

$$t_{2t} = t_{2a} + (t_1 - t_{2a}) \left[2.1 - \frac{h_1}{1.25\sqrt{rt_1}} \right] = 26.18 \approx 28$$

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$$t_{2d} > t_{2t}$$

So t_{2d} is taken as second shell course thickness.

∴ Thickness of second shell course = 32 mm

Third shell course

The procedure for second shell course is repeating to get the thickness of all other shell courses. The table given below shows the thickness for all other shell courses.

SHELL COURSES	THICKNESS (mm)
Third shell course	30
Fourth shell course	28
Fifth shell course	28
Sixth shell course	26
Seventh shell course	24
Eighth shell course	24
Ninth shell course	22

DESIGN OF BOTTOM PLATES

As per API 650 code bottom plates have a minimum thickness of 6 mm. So we take bottom plate thickness as 9 mm with corrosion allowance. The standard plate available in markets is about 10 mm. So the bottom plate thickness is taken as 10 mm.

DESIGN OF ANNULAR PLATE

$$\text{Product stress} = \frac{t_d \times S_d}{\text{as constructed } t \text{ exclusive of CA}}$$

$$= \frac{30.8 \times 220}{31} = 218.5 \text{ MPa}$$

$$\text{Hydrostatic stress} = \frac{31}{t_t \times S_t}$$

$$\text{as constructed } t$$

$$= \frac{28.8 \times 236}{34} = 199.9 \text{ MPa}$$

Plate Thickness ^a of First Shell Course (mm)	Stress ^b in First Shell Course (MPa)			
	≤ 190	≤ 210	≤ 220	≤ 250
$t \leq 19$	6	6	7	9
$19 < t \leq 25$	6	7	10	11
$25 < t \leq 32$	6	9	12	14
$32 < t \leq 40$	8	11	14	17
$40 < t \leq 45$	9	13	16	19

Product stress > Hydrostatic stress. So take product stress. So from above table we get annular plate thickness as 14 mm.

So annular plate thickness = 17 mm (with CA)

Standard plate available in 18 mm. So the thickness of annular plate is taken as 18 mm.

IV. ANALYSIS

STATIC ANALYSIS

Hydrostatic pressure, $P = \rho gH$

Where, ρ = Density of water

g = acceleration due to gravity

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H = height of tank from each shell course

SHELL COURSE	PRESSURE N/m ²
First shell course	176580
Second shell course	156960
Third shell course	137340
Fourth shell course	117720
Fifth shell course	98100
Sixth shell course	78480
Seventh shell course	58860
Eighth shell course	39240
Ninth shell course	19620

WIND LOAD ANALYSIS

Design wind speed, $V_z = V_b \times K_1 \times K_2 \times K_3$

Where, V_b = basic wind speed

K_1 = probability factor (risk coefficient)

K_2 = terrain, height and structure size factor

K_3 = topography factor

$V_z = 39 \times 1.06 \times 0.67 \times 1 = 27.69 \text{ m/s}$

Drag force, $F = \frac{1}{2} \times \rho V_z^2 S C_d$

Where, ρ = Density of air

V_z = Design wind speed

S = Cross sectional area

C_d = Coefficient of drag

$$F_1 = \frac{1}{2} \times 1.225 \times 27.69^2 \times 180 \times 0.82 = 69316.78 \text{ N}$$

$$F_2 = \frac{1}{2} \times 1.225 \times 27.69^2 \times 81 \times 0.82 = 31192.55 \text{ N}$$

IV. RESULTS AND OBSERVATIONS

STATIC ANALYSIS

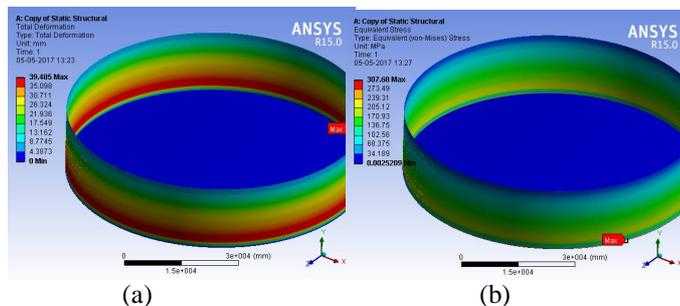


Fig 1: Static analysis result (a) Total deformation (b) Equivalent stress

The fig 1(a) shows the total deformation on tank due to hydrostatic pressure. The red region in the bottom has the maximum deformation and is about 39 mm. the blue region has the minimum deformation and is about 0 mm.

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minimum deformation is at the fixed supported face. Fig 1(b) shows the stress distribution on tank. Its maximum value is about 307 MPa. the stress is almost uniformly distributed over the body.

WIND LOAD ANALYSIS

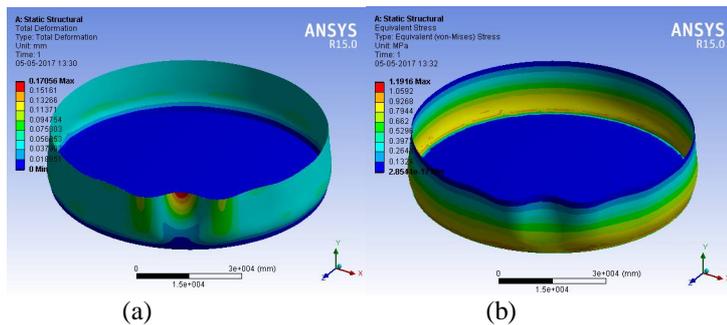


Fig 2: Wind load analysis result (a) Total deformation (b) Equivalent stress

The fig 2(a) shows the total deformation of the tank under wind load. The deformation is maximum at the top of front face of the tank. Front face is the face where the wind load is acting perpendicular direction. The minimum deformation is at the fixed supported face. And the maximum value is about 0.17 mm. Fig 2(b) shows the stress distribution on tank due to wind load. The maximum stress value is about 1.1MPa.

SLOSHING ANALYSIS

The fig 3(a) shows the pressure distribution during sloshing. The maximum pressure is at the bottom of tank. Then the result of sloshing is coupled with transient structural and analysed for deformation. Fig 3(b) shows the total deformation due to sloshing. Deformation is maximum at the bottom of tank, and is shown in red colour. Minimum value is shown in blue colour. The maximum deformation is about 24 mm. fig 3(c) shows the stress distribution its maximum value is shown in red. The blue strips in vertical direction is due to the stiffeners used in it. The analysis without stiffener has a higher deformation value. So that the stiffeners are added to the tank.

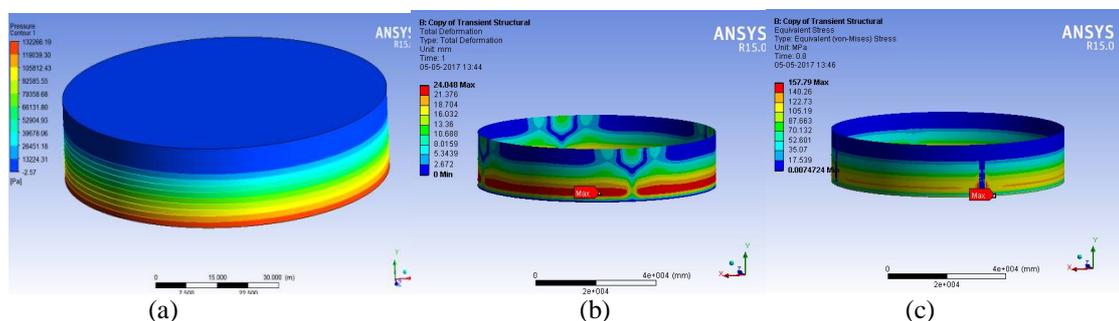


Fig 3: (a) Pressure distribution on sloshing analysis (b) Total deformation on transient structural analysis (c) Equivalent stress on transient structural analysis

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OBSERVATIONS

The table given below, shows the deformation and stress value during various analysis carried out in this work.

ANALYSIS	TOTAL DEFORMATION mm	VON-MISES STRESS MPa
STATIC ANALYSIS	39.485	307.68
WIND LOAD ANALYSIS	0.17	1.1916
SLOSHING ANALYSIS	24.048	157.79

V. CONCLUSION

STATIC ANALYSIS

The result shown in previous section, the maximum deformation occurred at the bottom region. This is due to the linear increase in hydrostatic pressure from top to bottom. Here the deformation has a bit higher value. To reduce the deformation some stiffeners can be added to the tank in various shapes and this can be taken as a future study.

WIND LOAD ANALYSIS

For the wind load maximum deformation was occurred at the top shell course of storage tank. The thickness of upper shell course has the minimum thickness value, so that the deformation get the higher value at the top of tank.

SLOSHING ANALYSIS

For the sloshing analysis the maximum deformation was occurred at the bottom of the storage tank. This is due to the pressure is maximum at the bottom of the tank. The deformation calculated from the transient analysis was not exceeded the critical limit, so that the storage tank is safe under the analysed conditions.

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