Review on Stresses in Cylindrical Pressure Vessel and its Design as per ASME Code

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***Abstract*— High pressure is developed in pressure vessel so pressure vessel has to withstand several forces developed due to internal pressure So selection of pressure vessel is most critical. For safety purpose the pressure vessel has to be designed according to ASME standards. In general the cylindrical shell is made of a uniform thickness which is determined by the maximum circumferential stress due to the internal pressure. Since the longitudinal stress is only one-half of this circumferential stress, these vessels have available abeam strength which makes the two-saddle support system ideal for a wide range of proportions. The structure is to be designed fabricated and checked as per ASME. By knowing these stresses, it is possible to determine which pressure vessel is designed for internal pressure alone, and to design structurally adequate and economical stiffening for vessel which require it. The section VIII, division 1 and division 2 are used in design. Division 1 correspond to 'design by rule and Division 2 correspond to 'design by Analysis'**

In this paper, the horizontal pressure vessel supported on saddles is designed according to the guidelines given in Div 1 and Div 2.

Efforts are made in this paper to understand the various stresses developed in pressure vessel and design the pressure vessel using ASME codes & standards to legalize the design

***Keywords*— Pressure vessel, Steam Boilers, ASME Code Introduction**

I. INTRODUCTION

A pressure vessel is defined as container with pressure differential between inside and outside, except for some isolated situations. The fluid inside the pressure vessel may undergo state of change like in case of boilers. Pressure vessel have combination of high pressure together with high temperature and may be with flammable radioactive material because of these hazards it is important to design the pressure vessel such that no leakage can take place as well as the pressure vessel is to be designed carefully to cope with high pressure and temperature Plant safety and integrity are of fundamental concern in pressure vessel design and these depend on adequacy of design codes. In general the cylindrical shell is made of a uniform thickness which is determined by the maximum circumferential stress due to the internal pressure. Since the longitudinal stress is only one-half of this circumferential stress, these vessels have available abeam strength which makes the two-saddle support system ideal for a wide range of proportions. The structure is to be designed fabricated and checked as per ASME. Pressure vessels are used in no of industries like power generation industry for fossil and nuclear power generation, In petrochemical industry for storage of petroleum oil in tank as well as for storage of gasoline in service stations and in the chemical industry. The size and geometric form of pressure vessel vary from large cylindrical vessel for high pressure application to small size used as hydraulic unit of aircraft.

The pressure vessels are of different types such as

• Spherical (e.g. LPG storage tanks)
• Cylindrical (e.g. liquid storage tanks)
• Cylindrical shells with hemispherical ends (e.g. distillation columns)

A. .Classification of pressure vessels

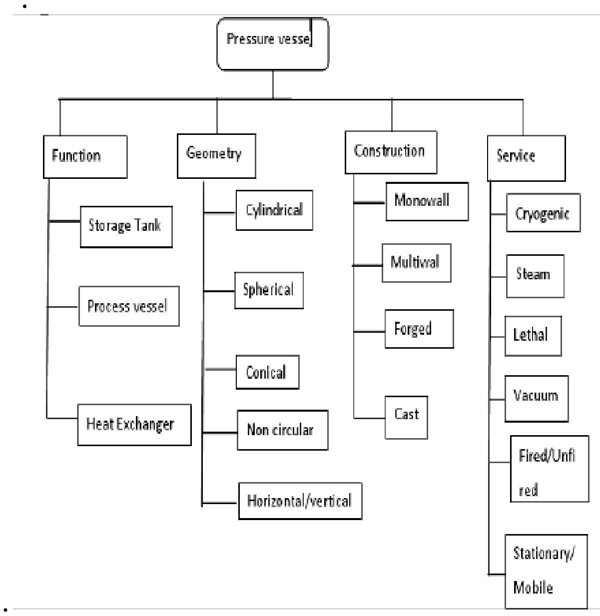
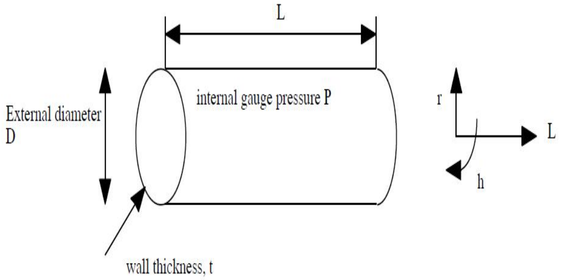


Fig.1 Classification of pressure vessels

B. Stresses in Cylinders and Spheres

1) For cylindrical pressure vessel

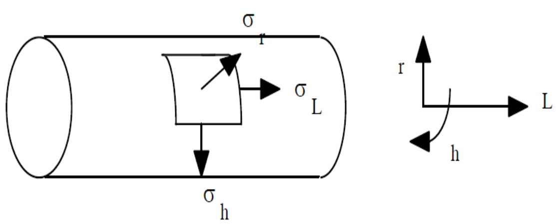


hydrostatic pressure causes stresses in three dimensions.

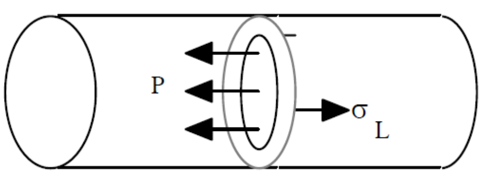
Longitudinal stress (axial) σ_L

Radial stress σ_r

Hoop stress σ_h



The longitudinal stress σ_L-



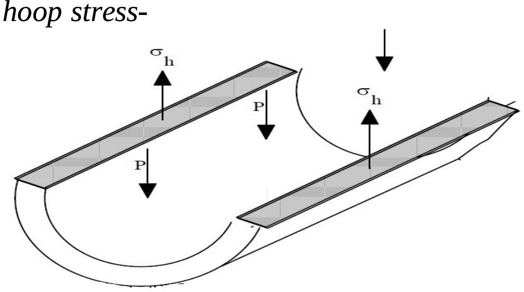
Force equilibrium

$$\frac{\pi D^2}{4} = \pi D t \sigma L$$

If P>0 , then σL is tensile

$$\sigma L = \frac{PD}{4t}$$

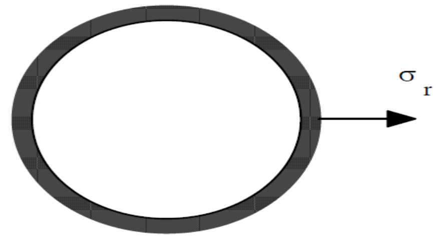
The hoop stress-



Force balance, $DLP = 2\sigma h L t$

$$\sigma h = \frac{PD}{2t}$$

Radial Stress σr -



σr varies from P on inner surface to 0 on the outer face

$\sigma r = 0(P)$

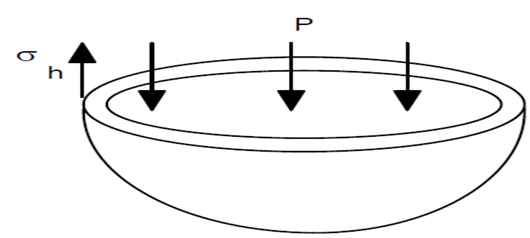
$\sigma h, \sigma L \approx P(\frac{D}{2t})$

Thin walled, so D$\gg t$

So $\sigma h, \sigma L \gg \sigma r$

So neglect σr

2) *For spherical pressure vessel*



$$P\frac{\pi D^2}{4} = \sigma h \pi D t$$

$$\sigma h = \frac{PD}{4t}$$

II DESIGN OF PRESSURE VESSEL AS PER ASME CODE

A. General Description of Pressure Vessel

A.UG-1 Scope:

The requirements of part UG are applicable to all pressure vessels and vessel parts and shall be used in conjunction with the specific requirements in subsections B and C and the

Mandatory Appendices that pertain to the method of fabrication and the material used.

B. UG-4 General Materials:

When specifications, grades, classes, and types are referenced, and material specification in Section-2, part A or Part B is a dual-unit specification (e.g., SA-516/SA-516M), the design values and rules shall be applicable to either the U.S. Customary version of the material specification or the SI unit version of the material specification. For e.g. when SA-516M Grade 485 is used in construction, the design values listed for its equivalent, SA-516 Grade 70, in either the U.S. Customary of metric section-2, Part D (as appropriate) shall be used.

C. UG-27 (C) Cylindrical Shells:

The minimum thickness for maximum Allowable working pressure of one-half cylindrical shells shall be the greater thickness of lesser pressure as given by

(1) Circumferential stress (Longitudinal joints); When the thickness does not exceed one-half of the inside radius, or p does not exceed 1.25SE.

(2) Longitudinal stress (Circumferential joints)

When the thickness does not exceed one-half of the inside radius, or P does not exceed 1.25SE.

D. UG-99 (b):

Except as otherwise permitted in (a) above and 274, vessels designed for internal pressure shall be subjected to a hydrostatic test pressure which at every point in the vessel is at least equal to 1.3 times the maximum allowable working pressure to be marked on the vessel multiplied by the lowest ratio (for the material of which the vessel is constant) of the stress value S for the test temperature on the vessel to the test stress value S for the design temperature (see UG-21). All loadings that may excite during this test shall be given consideration.

E.UG-32 (F) Ellipsoidal Heads:

The required thickness of a dished head of semi ellipsoidal form, in which half the minor axis (inside depth of the head minus the skirt) equals one-half of the inside diameter of the head skirt. An acceptable Approximation of 2:1 ellipsoidal head is one with a knuckle radius 0.17D and a spherical radius of 0.90D.

NOTE: for ellipsoidal heads with Ts/L<0.002, the rules of 1-4(f) shall also be met.

F. UG-32 (F) Hemispherical Heads:

When the thickness of a hemispherical head does not exceed 0.356L or P does not exceed 0.665SE.

G. UG 40 Limits Of Reinforcement:

As per type (b) reinforcement The limits of reinforcement, measured parallel to the vessel wall, shall be at a distance, on each side of the axis of the opening, equal to the greater of the following:

(1) The diameter d of the finished opening.

(2) The radius Rn of the finished opening plus the vessel wall thickness t, plus the nozzle wall thickness tn.

H. UG-45 Nozzle Neck Thickness:

As per type UG-45(a): the minimum wall thickness of a nozzle neck or the other connection (including access openings and opening for inspection) shall not be less than the thickness computed from the Applicable loadings in UG-22 plus the thickness added for allowable for correction and threading, as Applicable (see UG-31 C 2), on the connection.

UG-45(b): Additionally, the minimum thickness of a nozzle neck of other connection (except for access opening and openings for inspection only) shall not be less than the smaller of the nozzle wall thickness as determined by the applicable rule in(b)(1) or (b)(3) below, and the wall thickness as determined by (b)(4) below.

UG-45(b)(1): for vessels under internal pressure only, he thickness (plus correction allowance) required for pressure (assuming E=1.0) for shell or head at the location where the nozzle neck or other connection attaches to the vessel but in no case less than the minimum thickness specified for the material in UG-16(b)

UG-45(B)(2): For vessels under external pressure only, the thickness (plus correction allowance) obtained by using the external design pressure as an equivalent internal design pressure (assuming E=1.0) in the formula for the shell or head at the location where the nozzle neck of other connection attaches to the vessel but in no case less the minimum thickness specified for the material in UG-16(b);

UG-45(b)(3): for vessels designed for both internal and external pressure, the greater of the thickness Determined by (b)(1) or (b)(2) above

UG-45 (b)(4): the minimum thickness of standard wall pipe plus the thickness added for correction Allowance on the connection; for nozzles larger than the largest pipe size included in ASME B36, 10M, the wall thickness of that largest size plus the thickness added for correction allowance on the connection.

I. UG-16(b) General Design:

As per (b) of UG-16(b) Minimum Thickness of pressure Retaining Components: The minimum thickness of shells and heads used in compressed air service, steam service, and water service, made from material listed in table UCS-23, shall be 3/32 in (2.5 mm) exclusive of any correction allowance.

J. UG-22 Loadings:

As per type(c) Superimposed static reactions from weight of attached equipment, such as motors, machinery, other vessels, piping, linings, and insulations:

(1) Internal (see Appendix D);

(2) Vessel supports, such as lugs, rings, skirts, saddles, and legs (see Appendix G).

UW-(c) (2): Separate reinforcement elements may be added to the outside surface of the shell wall, the inside surface of the shell wall, or to both surfaces of the shell wall. When this is done, the nozzle and reinforced is no longer considered a nozzle with integral reinforcement and the F factor in UG-37(a) shall be F=1.0 figure UW-16.1

sketches (a-1), (a-2), and (a-3) depict various applications of reinforcement element added to sketch (a).

Any of these applications of reinforcement elements may be used with necks of the types shown in fig. UW-16.1 sketches (b), (c), (d), and (e) or any other integral reinforcement types listed in (1) above. The reinforcement plates shall be attached by welds at the outer edge of the plate, and at the nozzle neck periphery or inner edge of the plate if no nozzle neck is adjusted to the plate

III DESIGN DATA TABLE FOR PRESSURE VESSEL

1. Design drawing				
2.Specifications				
3. Vessel (name)	Horizontal retention tank			
4.Equipemt / Item number				
5. Design code and addenda				
6. Design pressure and temperature	Internal	78.46 psi and 150F	external	
7.pressure and temperature	75 psi and 150F			
8. vessel diameter	96 INCH OD			
9.Volume	640 cuft			
10.Design liquid level	47000 lbs			
11.contents and specific gravity	1			
12.Service				
13. Ma WP (Corrosion temperature)	75 psi			
14.Map (N&c)				
15. Test pressure	Shop			
16. Heat treatment				
17. Joint efficiencies	Shell1		Head 1	
18. Corrosion allowance	Shell 0.0	Head 0.01	Nozzle 0.0	Boot 0.0
19.Flange rating	Map ambient	75psig		
	MA WP	75psig D.T		
	Hydro	98psig		
	Ambient	D.T		
20. Materials	Allowable stress			
Shell (SA-516 Gr 70)	20000			
Head (SA-516 Gr 70)	20000			
Nozzles (SA-106 B)	17100			
Flanges	-			
Bolting	-			
21. Weight	47000lbs			

IV DESIGN METHADOLOGY OF PRESSURE VESSEL AS PER ASME CODES

A. Design of Shell

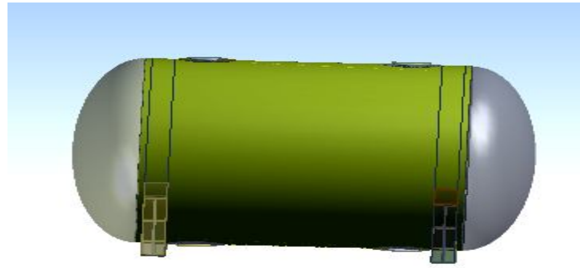


Fig 2. Catia model of Shell

Cylindrical shell thickness under internal pressure [UG-27(c)]	
Shell material, killed Carbon Steel, ASME SA516 Grade 70	
HYDROSTATIC Tested	
Shell Material Specifications	[Table 1A, Support ASME Sec II, Part D]
External Presuure Chart No.	CS-2
Vessel Inside Diameter	95.78 INCH
Shell Inside Diameter, D	95.78 INCH
Shell Inside Radius, Ri	47.89 INCH
Shell Length From Tangent To Tangent, L	120 INCH
Max. Design Temperature	150 F
Min. Design Metal Temperature, P (MDMT)	-20 F
Max.Operating Temperature	150 F
Max. Operating Pressure	75 PSIG
Max. Internal Design Pressure, P (MAWP)	78.46 PSIG
External Design Pressure (Full Vacuum)	Not Applicable
Shell Inside Diameter, D	95.78 INCH
Shell Inside Diameter, R	47.89 INCH
Static Head- Vessel Diameter	96 INCH
Static Head Pressure (Water Head * Sp.Gravity 1)	2.216898 INCH
Internal Design Pressure At Bottom Of Vessel	257.898 INCH
Max. Allowable Stress @	20000

Design Temp (150 0F) S	PSIG
Max. Allowable Stress @ Test Temp (55 0 F) St	20000 PSIG
Hydrostatic Test Pressure, Ph- 1.3×MAWP× (St/S) [UG-99(b)]	98 PSIG
Corrosion Allowance, C [Ug-25]	0 INCH
Joint Efficiency, E [Table UW-12]	1
[Spot Radiography],[TABLE UCS-57]	15%
Value Of 0.385*S*E [UG-27(c) (1)	6545 INCH
Since P Doesn't exceed 0.385 SE, Use Thin Wall Equation: [1]Min.Wall Thickness For Longitudinal Joints, t 1= PR / (SE-0.6P) [UG-27(c) (1)]	0.222 INCH
Value Of 1.25 SE [UG-27 (c) (1)]	75 PSIG
Since P Does Not Exceed 1.25 SE , Use Thin Wall Equation : [2]Min.Wall Thickness For Circumferential joints, t 2= PR / (2SE+0.4P) [UG-27(c) (2)]	0.211 INCH
The Min Wall Thickness Shall Be The Greater Of t 1 or t 2	0.222 INCH
By Adding Corrosion Allowance To Wall Thickness, t	0.222 INCH
Use Thickness Of Construction, t (Adopted Thickness)	0.313 INCH
Corroded Thickness= Adopted Thk = Corrosion allowance	0.313 INCH
Ladders and Platforms	Not applicable
Hot/Cold Insulation	Not applicable
Post Weld Heat Treatment, PWHT	Not applicable

B. Design Of Head

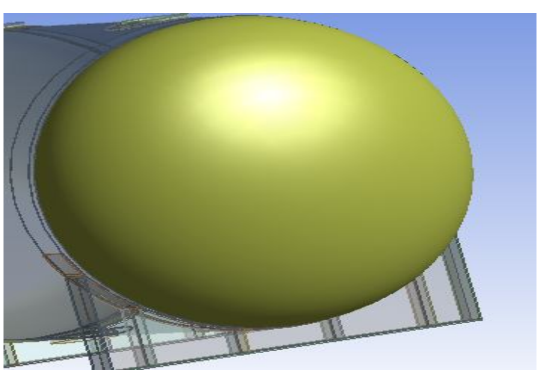


Fig.3 Catia model of elliptical head

Head Material, Carbon steel ASME SA516 Grade 70	
Head Type[Seamless] Ellipsoidal 2:1	
Head Material Specification [Table 1A, Support 1,ASME Sec II,and Part D]	
External Pressure Chart No.CS-2	
Head ID	95.578 INCH
Head OD[ASME B16.5-1996]	96 INCH
Head Outside Radius	48 INCH
Design Temperature	150 F
Operating Pressure	78.45 INCH
Head Skirt Inside Diameter, D	95.375 INCH
Head Inside Radius, L(ri)	47.687 INCH
Max.Allowable Stress @ Design Temp.(150 0F), S	20000 PSIG
Max.Allowable Stress @ Test Temp.(55 0F) St	20000 PSIG
Corrosion Allowance, C[UG-25]	0 INCH
Joint Efficiency, E (Seamless & Full Radiography) [TABLE UW-12]	0.85
Outside Diameter Of Head, Do	96 INCH
Outside Radius Of Head, Ro	48 INCH
Value Of 0.66*S*E	11305 PSIG
Since The Value Of 0.66E> P, Use Thin Wall Equation For Calculating The Min Required Thickness Of Head, t1=P*D/(2*S*E-0.2*P)[UG-32(d) (1)]	0.211 INCH
Compare To Thickness Of Seamless Spherical Shell Ps=0.665 S*E	11305 PSIG
Since P< Ps, Calculate Thickness For Thin Wall Spherical Shell t 2=P*Ro / (2*S*E+0.8*P) [APPENDIX 1-1] (2)	0.23746 INCH
For Thin Walled Ellipsoidal 2:1 Head: Use Thickness Of Construction, t (Adopted Thickness)	0.313 INCH

C. Design of Nozzle

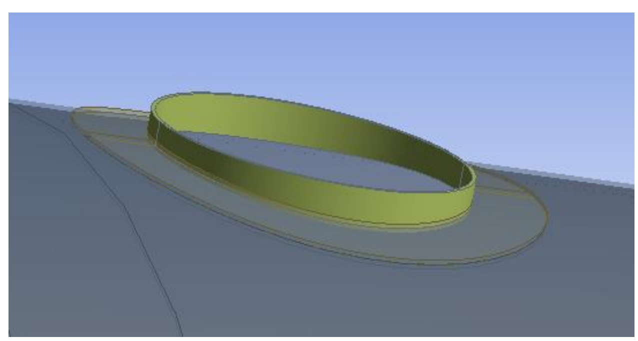


Fig 4. Catia model of Nozzle

M2, Nozzle Mark: N8 16" NPS, Sch 80, 300# WNRF (Manhole Located Shell With Reinforcement)	
No of Nozzles, n	
Nozzle Neck Thickness Calculation[UG-27(c)& Appendix1	
Nozzle Size NPS	16 INCH
Nozzle Material	ASME SA106 Grd B
Design Pressure, P	78.46 PSIG
For Nominated Design Pressure & Temperature, Flange rating. 300 [ANSI/ASME B16.5-1996]	
Max Allowable Stress Of Nozzle Material @ Design Temp (150 0 F) Sn	17100 PSIG
Max Allowable Stress Of Nozzle Material @ Test Temp (55 0 F) Snt	17100 PSIG
[Table 1A, Subpart 1, ASME Sec II Part D]	
Outside Radius Of Nozzle, Ron	16 INCH
Joint Efficiency Of Nozzle, En (Seamless Pipe)	1
Nozzle Corrosion Allowance, Can	0 INCH
Nozzle Thickness Calculation: Longitudinal Stress, t = P*Ron / (Sn*En-0.6*P)	0.35 INCH
By Adding Corrosion Allowance 12.5 % To The Thickness Of Nozzle, 1	0 INCH
By adding Pipe Tolerance 12.5 % To The Thickness Of Nozzle, t	0.46813 INCH
Use Nozzle 16" NPS With Selected Neck Sch.80	0.75 INCH

D. Catia Model

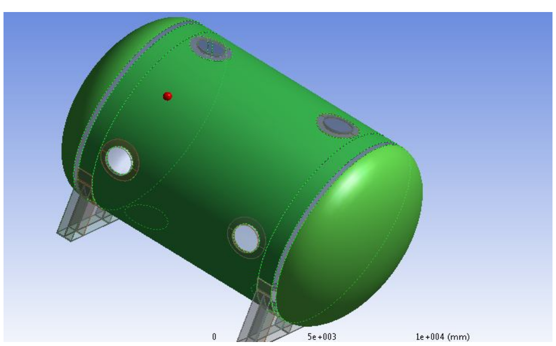


Fig5 CATIA Model of pressure vessel

IV. FUTURE SCOPE

1. Prototype model can be made and hydrostatic test can be performed smoothly with above design procedure.
2. Further FEA analysis can be done to verify the above design procedure

V. CONCLUSION

The design of pressure vessel is more of a selection procedure, selection of its component rather than designing each and every component, For storage of fluid the pressure vessel is mostly used because of its simplicity, high reliability, lower maintenance and compactness. The main parameter towards the design of pressure vessel is its high pressure fluid storage.The selection of pressure vessel component is very critical, slight change in selection will lead tp different pressure vessel altogether from what is aimed to be designed.

It is observed that all the manufactures of pressure vessels follow the ASME design codes for designing of pressure vessel so that leaves the designer free from designing the component. This aspect of design greatly reduces the development time for a new pressure vessel.

It also allows the designer the freedom to play with multiple prototypes for the pressure vessel before finalizing the decision. Selection of pressure vessel should be according to standards rather than customizing the design additional conclusion were made from project study are Low overall cost, Less time consumption, Universal Approach, Easy replacement

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