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Research Article

DESIGN OF PRESSURE VESSEL USING ASME CODE, SECTION VIII, DIVISION 1

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ABSTRACT

High pressure rise is developed in the pressure vessel and pressure vessel has to withstand severe forces. So the selection of pressure vessel is most critical. That's why we can say that pressure vessel is the heart for storage of fluid. Pressure vessel must pass series of Hydrostatic tests. These tests examine the ability of the structure to withstand various pressures to see if protective zone around the operator station remains intact in an overturn. The structure is to be designed, fabricated, fitted and checked as per ASME standard. Plant safety and integrity are of fundamental concern in pressure vessel design and these of course depend on the adequacy of design codes. The performance of a pressure vessel under pressure can be determined by conducting a series of tests to the relevant ASME standard. Efforts are made in this paper to design the pressure vessel using ASME codes & standards to legalize the design

KEYWORDS: Steam Boilers, Pressure Vessel, ASME Codes & Standards.

I. INTRODUCTION

Vessels, tanks, and pipelines that carry, store, or receive fluids are called pressure vessels. A pressure vessel is defined as a container with a pressure differential between inside and outside. The inside pressure is usually higher than the outside, except for some isolated situations. The fluid inside the vessel may undergo a change in state as in the case of steam boilers, or may combine with other reagents as in the case of a chemical reactor. Pressure vessels often have a combination of high pressures together with high temperatures, and in some cases flammable fluids or highly radioactive materials. Because of such hazards it is imperative that the design be such that no leakage can occur. In addition these vessels have to be designed carefully to cope with the operating temperature and pressure. It should be borne in mind that the rupture of a pressure vessel has a potential to cause extensive physical injury and property damage. Plant safety and integrity are of fundamental concern in pressure vessel design and these of course depend on the adequacy of design codes. When discussing pressure vessels we must also consider tanks. Pressure vessels and tanks are significantly different in both design and construction: tanks, unlike pressure vessels, are limited to atmospheric pressure; and pressure vessels often have internals while most tanks do not (and those that do are limited to heating coils or mixers).

Pressure vessels are used in a number of industries; for example, the power generation industry for fossil and nuclear power, the petrochemical industry for storing and processing crude petroleum oil in tank farms as well as storing gasoline in service stations, and the chemical industry (in chemical reactors) to name but a few. Their use has expanded throughout the world. Pressure vessels and tanks are, in fact, essential to the chemical, petroleum, petrochemical and nuclear industries. It is in this class of equipment that the reactions, separations, and storage of raw materials occur. Generally speaking, pressurized equipment is required for a wide range of industrial plant for storage and manufacturing purposes. The size and geometric form of pressure vessels vary greatly from the large cylindrical vessels used for high-pressure gas storage to the small size used as hydraulic units for aircraft. Some are buried in the ground or deep in the ocean, but most are positioned on ground or supported in platforms.

A. History of ASME Codes for Pressure Vessel:

Pressure vessels store energy and as such, have inherent safety risks. Many states began to enact rule and regulations regarding the construction of steam boilers and pressure vessels following several catastrophic accidents that occurred at the turn of the twentieth century that resulted in large Loss of life. By 1911 it was apparent to manufacturers and users of boilers and pressure vessels that the lack of uniformity in these regulations between states made it difficult to construct vessels for Interstate commerce. A group of these interested parties appealed to the Council of the American Society of Mechanical Engineers to assist in the formulation of standard specifications for steam boilers and pressure vessels. (The American Society of Mechanical Engineers was organized in 1880. As an educational and technical society of mechanical engineers.) After years of development and Public comment, the first edition of the Code, ASME Rules of Construction of Stationary Boilers and for Allowable Working Pressures, was published in 1914 and formally adopted in the Spring of 1915. The first Code rules for pressure vessels, entitled Rules for the Construction of Unfired Pressure Vessels, followed in 1925. From this simple beginning the Code has now evolved into the present eleven Section document, with multiple subdivisions, parts, subsections, and Mandatory and non-mandatory appendices. Almost all pressure vessels used in the process industry In the United States are designed and constructed in accordance with Section VIII, Division 1.A pressure vessel is a closed container designed to hold gases or Liquids at a pressure different from the ambient pressure. The end Caps fitted to the cylindrical body is called heads. The legal definition of pressure vessel varies from country to Country, but often involves the maximum safe pressure (may need to be above half a bar) that the vessel is designed for and the pressure Volume product, particularly of the gaseous part (in some cases an Incompressible liquid portion can be excluded as it does not contribute to the potential energy stored in the vessel.) Pressure vessels are used in a variety of applications. These include the industry and the private sector. They appear in these sectors respectively as industrial compressed air receivers and domestic hot water storage tanks, other examples of pressure vessels are: diving cylinder, recompression chamber, distillation towers, autoclaves and many other vessels in mining or oil refineries and petrochemical plants, nuclear reactor vessel, habitat of a space ship, habitat of a Pressure vessels A basic history and today Submarine, pneumatic reservoir, hydraulic reservoir under pressure, rail vehicle airbrake reservoir, road vehicle, airbrake reservoir and Storage vessels for liquefied gasses as ammonia, chlorine, as ammonia, chlorine, Propane, butane and LPG. Steel Pressure Vessel In the industrial sector, pressure vessels are designed to operate safely at a specific pressure and temperature, technically referred to As the "Design Pressure" and "Design Temperature". A vessel that is inadequately designed to handle a high pressure constitutes a very significant safety hazard. Because of that, the design and Certification of pressure vessels is governed by design codes such as The ASME Boiler and Pressure Vessel Code in North America, the Pressure Equipment Directive of the EU (PED), Japanese Industrial Standard (JIS), CSA B51 in Canada, AS1210 in Australia and other international standards like Lloyd's, Germanischer Lloyd, Det NorskeVeritas, Stoomwezen etc.

Pressure vessels can theoretically be almost any shape, but shapes made of sections of spheres, cylinders and cones are usually employed. More complicated shapes have historically been much harder to analyze for safe operation and are usually far harder to construct. Theoretically a sphere would be the optimal shape of a pressure vessel. Unfortunately the sphere shape is difficult to manufacture, therefore more expensive, so most of the pressure vessels are cylindrical shape with 2:1 semi elliptical heads or end caps on each end. Smaller pressure vessels are arranged from a pipe and two covers. Disadvantage of these vessels is the fact that larger diameters make them relatively more expensive, so that for example the most economic shape of a 1,000 liters (35 cu ft), 250 bars (3,600 psi) pressure vessel might be a diameter of 914.4 millimeters (36 in) and a length of 1,701.8 millimeters (67 in) including the 2:1 semi elliptical domed end caps. Generally, almost any material with good tensile properties that is chemically stable in the chosen application can be employed. Many pressure vessels are made of steel. To manufacture a spherical pressure vessel, forged parts would have to be welded together. Some mechanical properties of steel are increased by forging, but welding can sometimes reduce these desirable properties. In case of welding, in order to make the pressure vessel meet international safety standards, carefully selected steel with a high impact resistance & corrosion resistant material should also be used. What we make in Newark today includes: Pressure Vessels, Expansion Vessels, Surge Tanks, Industrial silencers, Nuclear Industry, Oil industry, Process Tanks, Vacuum Vessels, Blow down Tanks, Diesel Engine Starter Bottles, Fabrication, ASME, u stamp pressure vessel, ISO9001, Dished ends, cylinder, ASME.

B. History Of Pressure Vessels:

Numerous boiler explosions took place through the late 1800s and early 1900s. This led to the enactment of the first code for construction of steam boilers by the Commonwealth of Massachusetts in 1907. This subsequently resulted in the development and publication of the ASME Boiler and Pressure Vessel Code in 1914, which sought to standardize the design, manufacturing, and inspection of boilers and pressure vessels. In 1921 the National Board of Boiler and Pressure Vessel Inspectors was organized to promote consistent inspection and testing. The publication of the section on locomotive boilers also appeared in 1921. The ASME and the ASTM(American Society for Testing and Materials) material specification merged in 1924. The first publication of Section VIII ''Unfired Pressure Vessels,'' appeared in 1925. This document was referred to as one of a theoretical factor of safety of 5. The petroleum industry did not consider it to be adequate for their purposes and also desired better utilization of available materials. The year 1928 saw the advent of welded pressure vessels. For higher pressures the welded shells were made thicker than 70 mm. These required nondestructive examination (NDE) before service. In 1934, a joint API–ASME Committee published the first edition of an unfired pressure vessel code specifically for the petroleum industry. In 1952 these two separate codes merged into a single code – the ASME Unfired Pressure Vessel Code, Section VIII. The ASME Pressure Vessel Code, Section VIII Division 2: ''Alternative Rules for Pressure Vessels,'' was published in 1968 and the original code became Section VIII Division 1: ''Pressure Vessels.'' A considerable boost was provided to the understanding of the basic behavior of pressure vessel components following the development of the nuclear power program in the U.S. and Europe in the late 1950s and early 1960s. Similar developments can be found in the British, French, German and Japanese codes, to name but a few. By 1960 the need for a code for pressure vessels for commercial nuclear plants became imperative. This resulted in publication of the 1963 Edition, Section III: ''Nuclear Pressure Vessels.'' This was a design by analysis code with a theoretical safety factor of 3. After the publication of Section III: ''Nuclear Pressure Vessels'' in 1963, it was necessary to modify Section VIII for general pressure vessels. ASME Code Section VIII Division 2: ''Alternate Rules for Pressure Vessels'' appeared as a result and provided a theoretical factor of safety of 3. In 1971, Section III: ''Nuclear Power Components'' were classified as (a) pumps, (b) valves, and (c) piping. The stress limits for emergency and faulted conditions were introduced. In addition, the addenda of 1971 added storage tanks. The addenda of summer 1972 introduced Appendix G on non ductile failure. The Appendix F On evaluation of faulted conditions was included in the addenda of winter 1972. The design of component supports and core support structures appeared in the addenda of winter 1973. ASME Section III Division 1 is devoted entirely to nuclear power components and also contains the rules for the design of nuclear pumps and valves. The recognition of concrete reactor and containment vessels led to the publication of the Section II Division 2 code in 1975. Three subsections (NB, NC and ND) of ASME Section III Division 1 cover the design and construction of equipment of Classes 1, 2, and 3, respectively. The most stringent is Class 1, which requires design by analysis. Class 2 permits design by analysis as well as the use of formulas. Class 3 prescribes design by formula, and is equivalent to Section VIII Division 1. The designer evaluates the safety function of each pressure vessel and applies the appropriate code class. Design of supports for Section III Division 1 vessels are not prescribed in the ASME Code. Section III has a subsection NF, which prescribes the design of supports for Class 1, 2, and 3 pressure vessels. The addenda of winter 1976 changed the nomenclature of design, normal, upset, testing and faulted conditions to level A, B, C and D service conditions. In the 1982 addenda, the fatigue curves were extended to 1011 cycles. In the 1996 addenda, the design rules for high-temperature service were incorporated. In 1976, Division 3 was published which contained rules on transport of irradiated materials. The need for uniform rules for in-service inspection of nuclear power plants led to the issuance of the 1970 edition of Section XI: ''Rules for In-service Inspection of Nuclear Plant Components.'' The organization of the ASME Boiler and Pressure Vessel Code is as follows:

1. Section I: Power Boilers
2. Section II: Material Specification:
   i. Ferrous Material Specifications – Part A
   ii. Non-ferrous Material Specifications – Part B
   iii. Specifications for Welding Rods, Electrodes, and Filler Metals –Part C
   iv. Properties – Part D
3. Section III Subsection NCA: General Requirements for Division 1 and Division 2
   i. Section III Division 1:
      a. Subsection NA: General Requirements
      b. Subsection NB: Class 1 Components
      c. Subsection NC: Class 2 Components
      d. Subsection ND: Class 3 Components
      e. Subsection NE: Class MC Components
      f. Subsection NF: Component Supports
      g. Subsection NG: Core Support Structures
      h. Appendices: Code Case N-47 Class 1: Components in Elevated Temperature Service
   ii. Section III, Division 2: Codes for Concrete Reactor Vessel and Containment
4. Section IV: Rules for Construction of Heating Boilers
5. Section V: Nondestructive Examinations
6. Section VI: Recommended Rules for the Care and Operation of Heating Boilers
7. Section VII: Recommended Guidelines for Care of Power Boilers
8. Section VIII
   i. Division 1: Pressure Vessels – Rules for Construction
   ii. Division 2: Pressure Vessels – Alternative Rules
9. Section IX: Welding and Brazing Qualifications
10. Section X: Fiberglass-Reinforced Plastic Pressure Vessels
11. Section XI: Rules for In-Service Inspection of Nuclear Power Plant Components

TABLE 1Design & Construction Codes for Pressure Vessels

Country	Code	Issuing authority
U.S.	ASMEBoiler&Pressure Vessel Code	ASME
U.K.	BS1515Fusion Welded Pressure Vessels BS5500Unfired Fusion WeldedPressureVessels	British Standard Institute
Germany	AD Merblatter	ArbeitsgemeinschaftDruckbehalter
Italy	ANCC	AssociazioneNationalePerIlControllo Peula Combustione
Netherlands	Regeis Voor Toestellen	Dienst voor het Stoomvezen
Sweden	Tryckkarlskommissionen	SwedishPressure Vessel Commission
Australia	AS1200:SAABoilerCode AS1210Unfired Pressure Vessels	Standards Association of Australia
Belgium	IBNConstructionCode for Pressure Vessels	Belgian Standards Institute
Japan	MITI Code	Ministry of International Trade and Industry
France	SNCTConstruction Code for Unfired Pressure	Syndicat National de la Chaudronnerie et de la Tuyauterie Industrielle

The rules for design, fabrication and inspection of pressure vessels are provided by codes that have been developed by industry and government in various countries. The design and construction codes all have established rules of safety governing design, fabrication and inspection of boilers, pressure vessels and nuclear components. These codes are intended to provide reasonable protection of life and property and also provide for margin for deterioration in service. Table 1 also includes the ASME Boiler and Pressure Vessel Code. Some of the significant features of the latest version of the ASME Code Section III are:

- Explicit consideration of thermal stress
- Recognition of fatigue as a possible mode of failure
- The use of plastic limit analysis
- Reliable prediction of ductile failure after some plastic action.

In addition there is a continuous attempt to understand all failure modes, and provide rational margins of safety against each type of failure. These margins are generally consistent with the consequence of the specific mode of failure. A word or two about the impact of technological advances in pressure vessel design should be mentioned. The last three decades have seen great strides made in the improvement of digital computations. In the 1960s the use of computers began to make an impact on design and analysis of Pressure vessels.

The rapid development of finite-element software has remarkably impacted the detailed design of pressure vessel components. These developments along with continuing increase in computing speed and storage capacity of the computer have really made the design process extremely quick and at the same time have led to very accurate design assessment. Initially in the early to mid-1970s, detailed finite-element analyses were generally performed for confirmatory analyses. Today these tasks are routinely accomplished in an interactive mode. The three dimensional finite-element analysis programs using solid elements are rapidly replacing plate, shell, and two-dimensional programs for routine structural design analysis of pressure vessels. In addition the concepts of computer-aided design (CAD) and computer-aided manufacturing (CAM) are being integrated.

In spite of some of the most rigorous, well-conceived safety rules and procedures ever put together, boiler and pressure vessel accidents continue to occur. In 1980, for example, the National Board of Boiler and Pressure Vessel Inspectors reported 1972 boiler and pressure vessel accidents, 108 injuries and 22 deaths.2 The pressure vessel explosions are of course rare nowadays and are often caused by incorrect operation or poorly monitored corrosion. Safety in boiler and pressure vessels can be achieved by:

- Proper design and construction
- Proper maintenance and inspection
- Proper operator performance and vessel operation.

The design and construction cures are dependent upon the formulation and adoption of good construction and installation codes and standards. Thus the ASME Pressure Vessel Code requires that all pressure vessels be designed for the most severe coincident pressure and temperature expected during the intended service. There can be no deviation from this requirement, even if the severe condition is short term and occurring only occasionally. Bush has presented statistics of pressure vessels and piping failures in the U.S., Germany and the UK.3 He has concluded that a 99 percent confidence upper boundary for the probability of disruptive failure to be less than 1 _ 10–5 per vessel year in the U.S. and Germany. According to his study, periodic inspection is believed to be a significant factor in enhancing pressure vessel reliability, and successful applications of ASME Boiler and Pressure Vessel Codes (Sections I and VIII) are responsible for the relatively low incidence of non critical failures early in life. Pierre and Bayle authored an international perspective of the design of pressure vessels in 1924.

II CLASSIFICATION OF PRESSURE VESSELS*:*

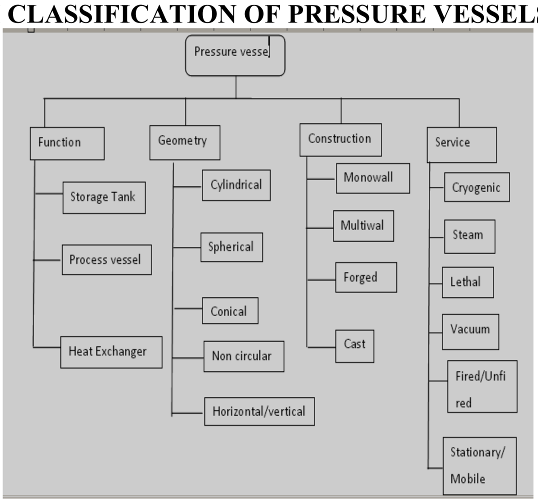

Figure 1 Classification of Pressure vessel

III DESIGN OF PRESSURE VESSEL AS PER ASME CODES:

General Description of Pressure Vessel Methodology.

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, the 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.

TABLE. 2 Design Data Table

1 Design drawing				
2 Specification				
3 Vessel (name)	Horizontal retention tank			
4 Equipment/ Item number				
5 Design Code & Addenda				
6 Design Pressure & Temperature	internal	78.46psi &150F	external	
7 Pressure & Temperature	75psi & 150F			
8 Vessel Diameter	96 INCH OD			
9 Volume	640cuft			
10 Design Liquid Level	47000 lbs			
11 Contents & Specific Gravity	1			
12 Service				
13 MAWP(Corrosion Temperature)	75 psi			
14 Map(N&C)				
15 Test Pressure	shop			
16 Heat Treatment				
17 Joint Efficiencies	Shell 1	Head 1		
18 Corrosion Allowance	Shell 0.0	Head 0.0	Nozzle 0.0	Boot -----
19 flange rating	Map Ambient	75 psig		
	MAWP	75 psig D.T		
	Hydro	98 psig		
	Ambient	D.T		
20. Materials	Allowable Stress			
Shell (SA-516 70)	20000			
Head(SA-516 70)	20000			
Nozzles(SA-106 B)	17100			
Flanges	-			
bolting	-			
21. weight	47000 lbs			
22. Remarks & Notes	-			

IV DESIGN METHODOLOGY OF PRESSURE VESSEL AS PER ASME CODES:

B. Design of Shell:

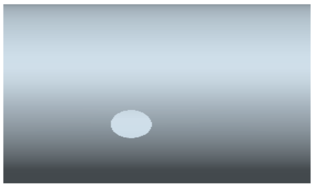

Figure 2 Pro-e model of shell

TABLE 3 Design 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, Suppart ASME Sec. II, Part D]
External Pressure 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 (T/T), L	120 INCH
Max. Design Temperature	150 F
Min. Design Metal Temperature, 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 Radius, R	47.89 INCH
Static Head = Vessel Diameter	96 INCH
Static Head Pressure (Water Head x Sp. Gr. 1.0)	2.216 PSIG
Internal Design Pressure At Bottom Of Vessel	257.898 PSIG
Max. Allowable Stress @ Design Temp.(150 oF), S	20000 PSIG
Max. Allowable Stress @ Test Temp.(55 oF), St	20000 PSIG
Hydrostatic Test Pressure, Ph = 1.3 x MAWP x (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 PSIG
Since P Does Not Exceed 0.385 SE, Use Thin Wall Equation: [1] Min. Wall Thickness For Longitudinal Joints, t1 = 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, t2 = PR / (2SE + 0.4P) [UG-27(c)(2)]	0.211 INCH
The Min. Wall Thickness Shall Be The Greater Of t1 or t2	0.222 INCH
By Adding Corrosion Allowance To Wall Thickness, t	0.222 INCH
Use Thickness Of Construction, t (Adopted Thickness)	0.313INCH
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

C. Design Of Head:

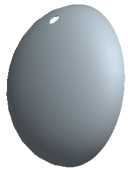

Figure 3 Pro-e model of shell

TABLE 4 Design of Head

Head Material, Carbon Steel ASME SA516 Grade 70	
Head Type [Seamless] Ellipsoidal 2:1	
Head Material Specifications [Table 1A, Support 1, ASME Sec. II, and Part D]	
External Pressure Chart No. CS-2	
Head ID	95.578INCH
Head OD [ASME B16.5-1996]	96 INCH
Head Outside Radius	48 INCH
Design Temperature	150 F
Operating Pressure	78.45 PSIG
Head Skirt Inside Diameter, D	95.375 INCH
Head Inside Radius, L (ri)	47.687 INCH
Max. Allowable Stress @ Design Temp. (150 oF), S	20000 PSIG
Max. Allowable Stress @ Test Temp. (55 oF), 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 0.66SE > P, Use Thin Wall Equation For Calculating The Min. Required Thickness Of Head, t: 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: t2 = P*Ro/ (2*S*E+0.8*P) [APPENDIX 1-1] (2)	0.23746 INCH
For Thin Wall Ellipsoidal 2:1 Head : Use Thickness Of Construction, t (Adopted Thickness)	0.313INCH

D. Design of Nozzle:

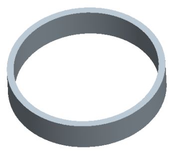

Figure 4 Pro-e model of Nozzle

TABLE 4 Design of Nozzle

M2. Nozzle Mark: N8 16" NPS, Sch.80, 300# WNRF (Manhole Located At Shell With Reinforcement)	
No. of nozzles, n	1
Nozzle Neck Thickness Calculation [UG-27(c) & Appendix 1-1]	
Nozzle Size, NPS	16 INCH
Nozzle Material	ASME SA 106 Grade 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 of), Sn	17100 PSIG
Max. Allowable Stress Of Nozzle Material @ Test Temp.(55 oF), 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)	0.0.35 INCH
By Adding Corrosion Allowance, t [UG-25]	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
With Thickness 0.75" [Table 2 Of ANSI B 36.10M-1985-(R-1994)]	

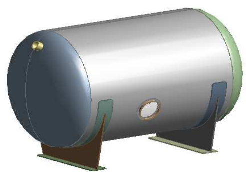

Figure 5 Pro-e model of Assembly of Horizontal Pressure Vessel.

V FUTURE SCOPE:

1. In future prototype model can be made and hydrostatic test can be performed smoothly with above design procedure.
2. Design of pressure vessel in PVELITE software can be accrue.
3. Further FEA analysis can be done to verify the above design procedure.

VI CONCLUSION:

The paper has led to numerous conclusions. However, major conclusions are as below:

The design of pressure vessel is initialized with the specification requirements in terms of standard technical specifications along with numerous requirements that lay hidden from the market.

The design of a pressure vessel is more of a selection procedure, selection of its components to be more precise rather designing each and every component.

Regarding storage of fluid for a pressure vessel system should be preferred due to its simplicity, better sensitivity, higher reliability, low maintenance, compactness for the same capacity.

The storage of fluid at high pressure in the pressure vessel is at the heart of its performance and is the first step towards the Design.

The pressure vessel components are merely selected, but the selection is very critical, a slight change in selection will lead to a different pressure vessel altogether from what is aimed to be designed.

It is observed that all the pressure vessel components are selected on basis of available ASME standards and the manufactures also follow the ASME standards while manufacturing the components. So that leaves the designer free from designing the components. 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.

The pressure vessel Selection Procedure after determining the inputs is a simplified process and can be automated to shorten the design cycle.

The following additional conclusions were made from the project study:

Selection of pressure vessel components should be according to standards rather than customizing the design.

- As abiding by the standards lead to:
- A universal approach
- Less time consumption.
- Easy replacement
- So less overall cost

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Note: *This Paper proposes some important tasks which are necessary in the modeling & Simulation procedure of Pressure Vessel. These tasks are modeling and virtual animation of the Pressure Vessel for checking the stresses at various locations on pressure vessel. Techniques of animation and checking Pressure Vessel are performed in Pro/ENGINEER Wildfire 3.0 & Ansys Workbench 12.0.*