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Horizontal Pressure Vessel Designing by Implementation of ASME Codes, Section VIII, Division1.

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ABSTRACT—In this paper, a suitable design of a Horizontal Pressure Vessel is achieved using the ASME codes Section VIII – Division 1. All the vessel components (Heads – Shell – Nozzles – Manhole – Saddles) are designed and calculated under the ASME codes with the help of both ANSYS and Solid works software. The designed HPV is used to holding 10m^3 of pressurized Propane gas. The pressure vessel has been drawn, modeled, simulated and analyzed and compared by both ASME codes and ANSYS software. The whole pressure vessel which is cylindrical in shape with its total components has been tested and validated against this computer program. The meshing of the desired and important components is carried out properly. Theoretical validation has been achieved on the total components and gave results within limits, also the results demonstrated that the highest stresses are experienced by the manhole followed by the shell, whereas the four nozzles, two saddles, and both heads are subjected to the lowest stresses. The results showed that the total deformation increases the further the component is from the saddle

Keywords: **Horizontal Pressure Vessels, ASME codes, Stresses, Deformation, Meshing and ANSYS.**

I. Introduction

The leakproof containers are known by the name on Pressure Vessels. They are playing the main role in the industry previously, now, and in the future. They may be of any shape and range from beverage bottles to the sophisticated ones encountered in engineering construction [1]. The pressure vessels are tanks (containers) containing inside pressure that differs from atmospheric pressure. These vessels are used for the confining fluids such as: (liquids – gases – vapors) inside them under pressure may higher or lower than ambient pressure. The pressure vessels are differ in terms of capacity, heat and pressure, some of these vessels may contain more the half million barrels of crude oil, the temperatures sometimes more than 200^{o}C specially in asphalt tanks and high viscosity products, the temperatures may slope down to 14^{o}C especially in tanks store hydrocarbons materials such as propane, butane and others, so it is necessary to understand the types of reservoirs and their components as well as the appropriate storage methods [2]. The ever-increasing of using pressure vessels for storing fluids, especially in the oil industry led to dealing with excess thinking during designing them, the reason that is these vessels may contain more the 15 psi inside, this can be dangerous and may lead to deadly accidents. So according to these hazards and dangers, while designing the vessels it is very important, they have to designed carefully to overcome the under usual conditions such as: pressures, stresses, temperatures, and earthquakes. The Design of pressure vessels thus has to be done in accordance with specific codes which give formulas, rules and specifications for satisfactory and safe construction of the main vessel components [3]. The carried-out studies have confirmed that all tested stresses values on the pressure vessel are more than the allowable ranges of internal pressure, and this permits the aimed vessel at the safe side to be manufactured.

II. Pressure Vessels

The pressure vessel is a closed leakproof container designed to hold liquids, vapors, gases, or any other objects at a pressure substantially lower or higher than the ambient pressure. Figure 1, illustrates

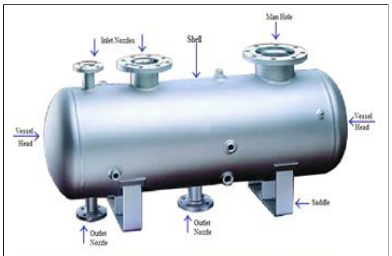


Figure 1. Horizontal Pressure Vessel

the general shape of horizontal pressure vessel. Based on ASME Code Section VIII, we can define pressure vessels are containers for the containment of pressure, either internal or external. This pressure may be obtained from an external source or by the application of heat from a direct or indirect source, or any combination thereof [4]. 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 [5]. During designing pressure vessels, the safety factor should be taken into consideration carefully. The safety factors are generally applied to the pressure vessel materials so that significant assurances exist that the component can safely perform in the operating environment [6], moreover, the design processes also should consist of the main parameters to guarantee the optimum safety possible. Some of these important parameters are high Pressure – high temperature – minimum design temperature for brittle fracture and corrosion allowance. Most pressure vessels are made of Carbon Steel (CS). Mostly, all pressure vessel components are welded together refers to CS properties.

The geometrical shape of pressure vessels can take the form of large cylindrical vessels (horizontal or vertical) – large spherical vessels – typical vessels, which are used for high-pressure gas storage, while small-sized pressure vessels are used as a hydraulic vessel in aircraft.

1.1 Pressure Vessels Types

The pressure vessels can be classified into many categories, Figure 2, below shows that these categories and their criteria include function, geometry, construction, and services. Each classification is shown in the hierarchy below [7].

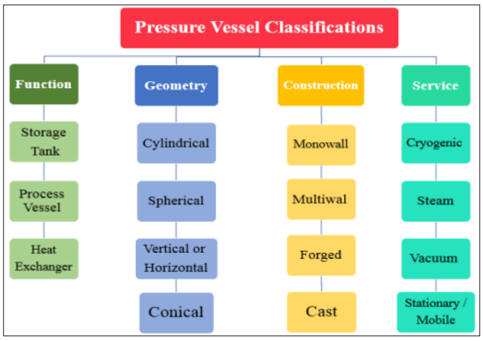


Figure (2). Pressure vessels types and their classification

1.2 Pressure Vessels Inspection

After a pressure vessel transmits from its initial manufacturing and testing to its end application, it may undergo alterations or repairs. Whenever something like this happens, the pressure vessel needs to be inspected again to see whether it still meets all requirements [8]. There are many types of tests that can be used to ensure the safety of the new pressure vessel and the installed equipment, and reliability. These tests are used to eliminate or to the limitation of hazards that may occur if over-pressurized vessels explode. Two primary types of tests that are performed on pressure vessels include hydrostatic and pneumatic tests. The key difference between these two types is that hydrostatic testing uses water as the test medium, and pneumatic testing uses a non-flammable, non-toxic gas like air or nitrogen [8]. The first mentioned test is preferred according to its safer method during testing and less energy is released if any fracture occurs.

Beyond these basic inspections, there are five non-destructive tests (NDT) or non-destructive examination (NDE), they are: (1) Visual Test. (2) Liquid Penetrant Test. (3) Magnetic Particle Test. (4) Radiographic Test and (5) Ultrasonic Testing. These tests are playing important role to ensure safety and reliability of the designed pressure vessel.

III. Pressure Vessel Design

The design of pressure vessels should be achieved with maximum care, the reason that those work under enormous pressure, and any damage or rupture that occurs on the vessel may lead to seriously harmful to the workers and the firm. As a rule, the ASME Sec VIII Division1 is used to govern the design of pressure vessels. The horizontal pressure vessel should be designed according to the minimum requirements of design without any flaws in the vessel parts. The selected ranges used in designing horizontal vessel is (0.1MPa to 20MPa). The concerned vessel under this study is composed of (Shell – Heads – Nozzles – Manhole – Saddles).

2.1 Pressure Vessel Specifications

The specifications concept is used to describe an attempt to capture and incorporate the best applications with regard to the design. The concerned pressure vessel in this study is used to contain Liquid Propane Gas (LPG), this vessel has two elliptical heads, two fixed saddles, inner shell diameter (D_i) mm and total shell length is (L_s). This vessel is designed to hold about 10m3 of LPG at design pressure of (P_d) MPa. Table 1, displays the studied pressure vessel specifications, meanwhile Table 2, shows the mechanical properties of Carbon steel

Table 1. Pressure Vessel Specifications

Parameter	Value
Shell inner Diameter (D_i)	1600mm
Shell length (L_s)	4500mm
Design pressure (P_d)	1.61MPa
Vessel material	SA516 Gr 70

Table 2. Mechanical Properties of Gr70 Carbon Steel

Parameter	Value
Density (ρ)	3.24kg/m
Yield strength ($\boldsymbol{\sigma_s}$)	260MPa
Tensile strength ($\boldsymbol{\sigma_t}$)	410 – 540 MPa
Elastic modulus (E)	145 – 205MPa
Hardness	55 max.
Elongation	24%

2.2 Shell design

To get the optimum shell design is achieved by using the ASME codes. The main consideration in designing the shell is the shell thickness which depends on two main aspects the design pressure and corrosion allowance. The corrosion allowance can be given directly by ASME standards equals to 3mm. The shell thickness is determined by the total pressure inside the vessel which is the sum of the design pressure and static pressure, this can be formulated as following:

$$P_t = P_d + P_H$$
$$P_H = \rho g H$$

Where:

t = Wall thickness
D_o = Outside diameter
D_i = Inside diameter
SF = Straight flange height
DH = Dished height
TH_i = Total height inside
CR = Crown radius
KR = Kunckle radius
P_t= Total pressure.
P_d= Design pressure
P_H= Static head pressure
ρ= Contained fluid's density, for LPG is $495\frac{kg}{m^3}$
g= gravitational acceleration
H= Fluid's height

The obtained total pressure inside the vessel by using those equations is 1.63MPa. The formula in ASME section VIII, Division 1, paragraph UG-27, used for calculating the wall thickness and design pressure of thin wall pressure vessels are:

a) Longitudinal Stress (circumferential welds) when, p < 1.25SE

$$t = \frac{PR}{2SE + 0.4P}$$
$$P_a = \frac{2SEt_f}{R - 0.4t_f}$$

b) Circumferential Stress (Longitudinal welds) when, p < 0.385SE

$$t = \frac{PR}{SE - 0.6P}$$
$$P_b = \frac{SEt_f}{R + 0.6t_f}$$

Where:

t= Minimum design wall thickness (mm)
P= Design pressure (psi)
D= Outside diameter (mm)
R= Internal radius (mm)
E= Welding factor
C= Corrosion Allowance (mm)
S= Maximum allowance stress according to ASME

Using previous equations results in a longitudinal maximum pressure of 425.28MPa and a circumferential maximum pressure of 208.96MPa, which are both much higher than the total pressure. Table 3, presents all shell design parameters. It clear that the chosen shell thickness is safe and will be adopted in this study.

Table 3. Shell Design Parameters.

Parameter	Value
Total pressure (P_t)	1.63MPa
Safety factor (sf)	4
Joint efficiency (E)	1
Maximum allowable stress ($\boldsymbol{s}$)	120.66MPa
Corrosion allowance (c.a.)	3mm
Final shell thickness (t_f)	14mm
Longitudinal maximum pressure (Pa)	425.28MPa
Circumferential maximum pressure (Pb)	208.96MPa

2.3 Heads Design

For the pressure vessel heads design all conditions are the same with shell such as design pressure, design temperature, same material and same wall thickness which approximately equals the shell thickness. Figure 3, shows detailed vessel head.

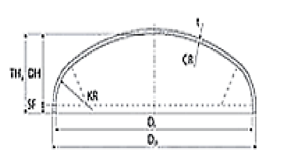


Figure3. Head Design Parameters

Where:

The pressure vessel used in this study has elliptical heads. The head's inner crown height (h_i) and its thickness are calculated according the following equation:

Parameter	**Value**
Final manhole external diameter ($\boldsymbol{D_{o,m}}$)	609.6mm
Final manhole thickness ($\boldsymbol{t_{n,m}}$)	9.53mm
Ratio of allowable stress ($\boldsymbol{f_r}$)	<1
No. bolt holes	15

$$h_i = \frac{D_i}{4}$$

This results in an inside crown height of 0.4m. As for the head's minimum thickness ($t_{h,min}$), next equation is used to obtain the thickness:

$$t_{h,min} = \frac{P_t \times D_i \times k}{2S \times E_L - 0.2P_t}$$

Where E_L = the head longitude efficiency, which is set to 1 in this study.

K = factor which is calculated using the following equation:

$$k = 0.167\left(2 + \left(\frac{D_i}{2h_i}\right)^2\right)$$

To calculate the maximum pressure (P_{max}) allowed by this thickness, the following equation is used:

$$P_{max} = \frac{2S \times E_l \times t_h}{k \times D_i + 0.2t_h}$$

This results in a maximum allowable pressure of 1.65MPa, which is higher than the total pressure acting on the head.

Table 4: Head Design Parameters

2.4 Manhole Design

The main aim of manhole for letting personnel in and out the vessel to easily for inspection, repair and cleaning. According to ASME standards pressure vessels diameters exceeding 500mm should have a manhole without providing the minimum standard size. It's clear that the manhole region is weak, the amount of this weakness depending on hole diameter. The manhole design depends on two main considerations are: (1) primary membrane stresses in the vessel must be within limits set by the allowable tensile stress and (2) peak stresses should be kept within acceptable limits to ensure satisfactory fatigue life [9]. For manhole the required thickness on the vessel at the longitudinal and circumferential sections are given in the following equation:

$$t_{a,m} = \frac{P \times R_{im}}{2S \times E + 0.4 \times P_t} + c.a.$$

$$t_{b,m} = \frac{P \times R_{im}}{S \times E - 0.6 \times P_t} + c.a.$$

$$R_{im} = \frac{D_{om}}{2} - t_{nm} + c.a$$

Where:

$t_{a,m}$ = Minimum manhole thickness
$t_{b,m}$ = Circumferential sections
R_{im} = Internal radius of manhole.

The obtained minimum thickness by using the previous equation about 5mm and 7mm respectively, these values considered less the nominal thickness according the study, then it can be considered reliable and trusted, the manhole parameters are presented in Table 5, the required minimum area (A_m) obtained by the following equation:

$$A_m = D_{im}t_{a,m}f_r + 2t_{a,m}(t_{n,m} - c.a.)(1 - f_r)$$

The manhole internal diameter calculated by the equation:

$$D_{i,m} = D_{o,m} - 2(t_{n,m} - c.a.)$$

f_r = the ratio of allowable stress, which in this study is < 1.

The obtained minimum area of the manhole flange cover is 8351.56mm^2, then it can be considered reliable design and trusted

Table 5: Manhole Design Parameters

Parameter	**Value**
Inside crown height (h_i)	400mm
Head longitude efficiency (E_L)	1

	Outer diameter (D_o) (mm)	**Thickness (t_f) (mm)**	**Minimum area (A) (mm^2)**
Charge inlet	60.33mm	3.91mm	819.1mm^2
Outlet	88.9mm	5.49mm	1175mm^2
Drain	60.33mm	3.91mm	819.1mm^2
Pressure transmitter	60.33mm	3.91mm	819.1mm^2

2.5 Nozzles Design

The nozzles are cylindrical components that pierce the shell body, they usually end with flanges for fit connection with vessel. Furthermore, these nozzles should be supported to prevent any leakage or failure. The studied vessel contained of four nozzles, Figure 4, clarifies the nozzle design parameters, and the below Table 6, displays all calculated design parameters of the concerned nozzles.

Table 6: Nozzle Design Parameters

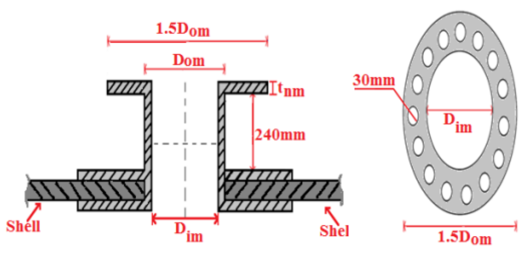


Figure 4. Nozzle Design Parameters

2.6 Saddle

The main aim of the saddle is to supply a steady base to the horizontal pressure vessel, at the same time they support the weight load over the vessel to prevent the local stress in the shell. Saddles contain base plate, web ribs and wear plate. The bottom portion of the saddle is normally welded or bolded to the ground. A typical shape of saddle support is shown on Figure 5, meanwhile the figure 6, presents the design parameters of the saddle. According to the ASME codes the saddles are designed with minimum contact angle between the shell and the saddle. Table 7, presents all dimensions needed for this study.

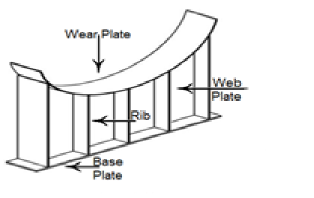


Figure 5. Horizontal Saddle Support

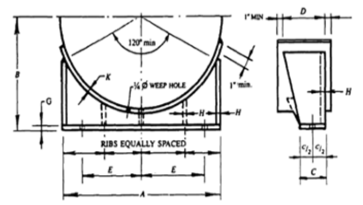


Figure 6. Saddle Design Parameters

Table7: Values of Saddle Dimensions for the Current Pressure Vessel

Dimension	Value
A	1460.44mm
B	1066.76mm
C	152.40mm
D	279.40mm
E	558.79mm
G	19.05mm
H	9.53mm
K	9.53mm

The following equations are used to estimate the weight of the vessel and contained fluid respectively:

$$m_p = V_p \rho_p$$
$$m_f = 0.8 V_f \rho_f$$

Where:

m_p & m_f = Mass of pressure vessel and contained fluid.
V_p & V_f= Volume of the vessel and contained fluid.
ρ_p & ρ_f= Density of the vessel and contained fluid.

By using the mentioned equations and given data, the obtained results are:

The horizontal force caused by the weight can expressed by the equation:

$$F = K_{11} Q$$

Where:

F= Horizontal force
K_{11}= Constant depends on the contact angle = 0.204
Q= Load on the saddle = $Q = g m_{saddle}$
H= Web plate thickness
σ_{saddle}= Stress = $\sigma_{saddle} = \frac{3F}{HR_i}$

IV. ANSYS Static Structural

The static structural analysis for the pressure vessel was entirely achieved by the ANSYS software to analyze subjected stresses and deformation caused by the internal pressure and weight of fluid in the vessel.

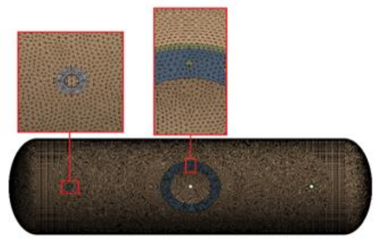


Figure 7. View of Meshed Vessel

The structural analysis simulation was done using a three-dimensional static finite element method using an ANSYS package known as ANSYS Static Structural. The mathematical model, which includes the boundary conditions, equations used to calculate total deformation and the equivalent stress, and the numerical analysis method achieved. During this study, a meshing achieved with 1176522 nodes and 596164 elements was used on the total body, Figure 7, is used to show the meshing operation on the total body. The obtained results show that the cell aspect ratio was very low with a value of not exceeding 1.16 for the majority of the elements for 99.7% of the cells.

The boundary conditions set in this study include a constant pressure induced by the contained fluid, the total weight and fixed support at the bottom part of the saddle. Figure 8, presents the boundary conditions values on the total vessel body. The pressure is set to 1.63MPa, which is the total pressure the

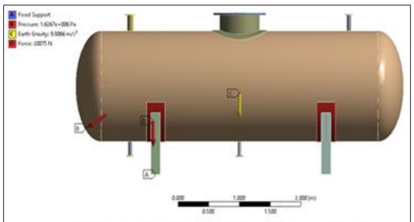


Figure 8. Boundary Conditions Values on Shell

vessel is subjected to, and is applied to the inner walls of the vessel's shell, head, nozzles and manhole.

Results also show that the highest equivalent stress occurred in the contact area between the manhole and the shell, while the lowest stresses occurred at the majority of the saddle. The highest equivalent stress experienced by the pressure vessel is 36.18% lower than the maximum tensile strength of the material, which is 483MPa.

The results of the equivalent stress distribution show that the highest stresses occur at the contact point between the part of the saddle that is in contact with the shell and the support of the saddle. Figure 9, presents these stresses values, the highest stress on the saddle reached was 172MPa, which is 64.17% less than the ultimate strength of the material.

Figure 9. Stresses Distributed on Vessel and Saddle

Results on the manhole show that the highest equivalent stress experienced is 244Mpa, as shown in the Figure10, which is 49.48% lower than the tensile strength of the material. The results also show that there is no observable pattern to the distribution of the stresses across the manhole.

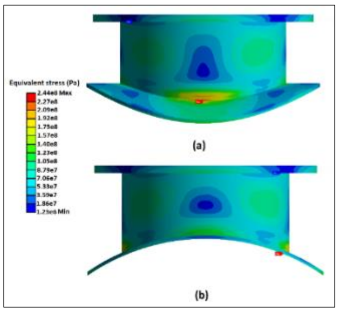


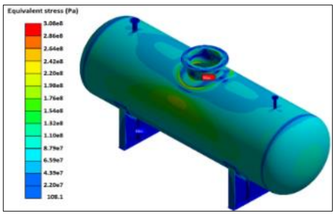
Figure 10. Stress Distribution on Manhole.
(a) Front View of the Manhole and (b) the Isometric View

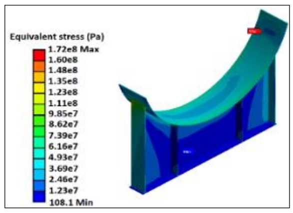
V. Conclusion

Every year ASME organizes an annual meeting concerned with pressure vessel technology, usually more than 500 papers are presented and many of them are published by ASME in special booklets. During this paper a horizontal pressure vessel has been studied and designed with the help of ASME standard section VIII division 1, all the designs were achieved according to these standards. The stresses and deformation distribution on the modeled, meshed and simulated pressure vessel have been accomplished also with help of the ANSYS Static Structural package and gave the desired results. The obtained stresses were less than the utmost strength of the material, these stresses were located between the shell and the manhole which is the highest one, meanwhile, the lowest stresses were located at the saddle, meantime the highest deformation was located at the manhole and the region which far from the saddle, at the same time the saddles do not subject to any deformation.

References

Capacity of the tank =	$10.38m^3$
Mass of the fluid =	5115.86kg
Vessel volume =	$0.43m^3$
Total tank mass =	3347.08kg
Total vessel mass with fluid =	8462.94kg





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