DESIGN AND ANALYSIS OF PRESSURE VESSEL

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ABSTRACT

This technical paper presents design and analysis of pressure vessel. High pressure rise is developed in the pressure vessel and pressure vessel has to withstand severe forces. In the design of pressure vessel, safety is the primary consideration, due the potential impact of possible accident. There have a few main factors to design the safe pressure vessel. This writing is focusing on analyzing the safety parameter for allowable working pressure. Allowable working pressures are calculated by using Pressure Vessel Design Manual by Dennis Moss, third edition. The corruption of the vessel are probability occur at maximum pressure which is the element that only can sustain that pressure. Efforts are made in this paper to design the pressure vessel using ASME codes & standards to legalize the design.

Keyword: - *Design pressure, Design temperature, Vessel failure, Reinforcement design etc….*

1. INTRODUCTION

Tanks, vessel and pipelines that carry, store or receive fluids are called pressure vessel. 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. Pressure vessel often has a combination of high pressure together with high temperature and in some cases flammable fluids or highly radioactive material. Because of such hazards it is imperative that the design be such that no leakage can occur. In addition vessel has to be design carefully to cope with the operating temperature and pressure [1].

2. PROBLEM STATEMENT

Pressure vessel may have failed through corrosion fatigue because the wrong material was selected. The designer must be as familiar with categories and types of failure as with categories and types of stress and loadings [1].

● Material - Improper selection of material; defects in material.
● Design - Incorrect design data; inaccurate or incorrect de-sign methods; inadequate shop testing.
● Fabrication - Poor quality control; improper or insufficient fabrication procedures including welding.

3. METHODOLOGY

To design of pressure vessel the selection of Code are important as a reference guide to achieve the safety pressure vessel. The selections of ASME VIII div 2 are described. The standard of material use are explained. Beside of that, the design and analysis software to obtain the result are introduced.

3.1 Code Selection

ASME VIII (division 2) "Construction of Pressure vessel Codes" are selected. It is, however, emphasized that any standard selected for manufacture of the air receiver must be followed and complied with in entirety and the design must not be based on provisions from different standards [2].

3.2 Material Selection

Several of materials have been use in pressure vessel fabrication. The selection of material is based on the appropriateness of the design requirement. AU the materials used in the manufacture of the receivers shall comply with the requirements of the relevant design code and be identifiable with mill sheets. The selection of materials of the shell shall take into account the suitability of the materials with the maximum working pressure and fabrication process. For this kind of pressure vessel, the selection of material use is based on Appendix B:

Table - 1: Material assignment

Head	SA- 106 B
Shell	SA- 106 B
Nozzle -Relieve Valve	SA- 106 B
Pressure Gauge (PG)	SA- 106 B
Drain	SA- 106 B
Inlet	SA- 106 B
Outlet	SA- 106 B

According to ASTM standard this specification for pres-sure vessel is suitable for higher temperature services. The chemical and tensile requirement of Seamless Carbon steel pipe for high temperature service (SA-106 B) is as per table [3].

Table - 2: Material Composition

	Composition %, (Grade B)
Carbon, max	0.3
Manganese	0.29-1.06
Phosphorus, max	0.035
Sulfur, max	0.035
Silicon, min	0.10
Chrome, max	0.40
Copper, max	0.40
Molybdenum, max	0.15
Nickel, max	0.40
Vanadium. max	0.08

Table - 3: Material Properties

	Grade B
Tensile strength, min, psi (MPa)	60 000 (415)
Yield strength, min, psi (MPa)	35 000 (240)

3.3 Design Pressure

The pressure use in the design of a vessel is call design pressure. It is recommended to design a vessel and its parts for a higher pressure than the operating pressure. A design pressure higher than the operating pressure with 10 percent, whichever is the greater, will satisfy the requirement. The pressure of the fluid will also be considering. The maximum allowable working pressure (MAWP) for a vessel is the permissible pressure at the top of the vessel in its normal operating position at a specific temperature. This pressure is based on calculations for every element of the vessel using nominal thicknesses exclusive of corrosion allowance. It is the basis for establishing the set pressures of any pressure-relieving devices protecting the vessel. The design pressure may be substituted if the MAWP is not calculated. (UG22, ASME VIII.) [1].

3.4 Design Temperature

Design temperature is the temperature that will be maintained in the metal of the part of the vessel being considered for the specified operation of the vessel. For most vessels, it is the temperature that corresponds to the design pressure. However, there is a maximum design temperature and a minimum design temperature (MDMT) for any given vessel. The MDMT shall be the lowest temperature expected in service or the lowest allowable temperature as calculated for the individual parts. Design temperature for vessels un-der external pressure shall not exceed the maximum temperatures [1].

3.5 Corrosion Allowance

Corrosion occurring over the life of a vessel is catered for by a corrosion allowance, the design value of which depends upon the vessel duty and the corrosiveness of its content. A design criterion of corrosion allowance is 1 mm for air receiver in which condensation of air moisture is expected [1].

3.5 ASME Code, Section VIII, Division 1 vs. Division 2

ASME Code, Section VIII, Division 1 does not explicitly consider the effects of combined stress. Neither does it give detailed methods on how stresses are combined. ASME Code, Section VIII, Division 2, on the other hand, provides specific guidelines for stresses, how they are combined, and allowable stresses for categories of combined stresses. Division 2 is design by analysis whereas Division 1 is design by rules. Although stress analysis as utilized by Division 2 is beyond the scope of this text, the use of stress categories, definitions of stress, and allowable stresses is applicable.

Division 2 stress analysis considers all stresses in a triaxial state combined in accordance with the maximum shear stress theory. Division 1 and the procedures outlined in this book consider a biaxial state of stress combined in accordance with the maximum stress theory. Just as one would not design a nuclear reactor to the niles of Division 1, one would not design an air receiver by the techniques of Division 2. Each has its place and applications. The following discussion on categories of stress and allowables will utilize in- formation from Division 2, which can be applied in general to all vessels [1].

4. DESIGN

4.1 Shell Design

The minimum thickness or maximum allowable working pressure of cylindrical shells shall be the greater thickness or lesser pressure as given by (1) or (2) below.

Circumferential Stress (Longitudinal Joints)

When the thickness does not exceed one-half of the inside radius, or P does not exceed 0.385SE, the following formulas shall apply:

$$t = \frac{PR}{SE - 0.6P} \quad \text{or} \quad P = \frac{SEt}{R + 0.6t} \qquad (1)$$

Longitudinal Stress (Circumferential Joints)

When the thickness does not exceed one-half of the inside radius, or P does not exceed 1.25SE, the following formulas shall apply: [1]

$$t = \frac{PR}{2SE + 0.4P} \quad \text{or} \quad P = \frac{2SEt}{R - 0.4t} \qquad (2)$$

Table – 4: Design specifications for shell

NOTATION	SI		MKS	
P = internal pressure, psi	1740.4524	psi	12	Mpa
D = inside diameter, in.	59.05511811	in	1500	Mm
S = allowable or calculated stress, psi	20015.203	psi	138	Mpa

E =joint efficiency	1		1	
Corrosion Allowance	0.059055118	in	1.5	mm
FOS	3.5		3.5	
Tensile Stress	70053.2091	psi	483	Mpa
Yield Stress	50038.0065	psi	345	Mpa

4.2 Circumferential stress criterion

Checking for 0.385SE

S = 20015.203

E = 1

0.385SE = 7705.853001 > 1740.4524

$$t = \frac{PR}{SE - 0.6P}$$

t = 68.8073mm

4.3 Closure design

The required thickness at the thinnest point after forming of ellipsoidal, torispherical, hemispherical, conical, and toriconical heads under pressure on the concave side shall be computed by the appropriate formulas (UG-16). In addition, provision shall be made for any of the other loadings given in UG-22. The thickness of an unstayed ellipsoidal or torispherical head shall in no case be less than the required thickness of a seamless hemispherical head divided by the efficiency of the head-to-shell joint [3].

4.4 Ellipsoidal Heads design

The required thickness of a dished head of semi ellipsoidal form, in which half the minor axis equals one-fourth of the inside diameter of the head skirt, shall be determined by

t = 65.78947368mm [3]

$$t = \frac{PD}{2SE - 0.2P} \quad \text{or} \quad P = \frac{2SEt}{D + 0.2t} \qquad (1)$$

t = 65.78947368mm [3]

4.5 Nozzle and reinforcement

Openings in cylindrical or conical portions of vessels, or in formed heads, shall preferably be circular, elliptical, or obround. When the long dimension of an elliptical or obround opening exceeds twice the short dimensions, the reinforcement across the short dimensions shall be increased as necessary to provide against excessive distortion due to twisting moment.

The constraints for the nozzle design were flow rate & standard pipes availability. Due to the standard flow rates, the inlet and outlet diameter were taken as 100 and 80 mm respectively. [4]

Table – 5: Nozzle selection

Nozzle	1	2	3
	4" sch 40	3" sch 40	20" sch 40
ID , in	4.026	3.068	22.624
OD , in	4.5	3.5	24

4.6 Reinforcement Design

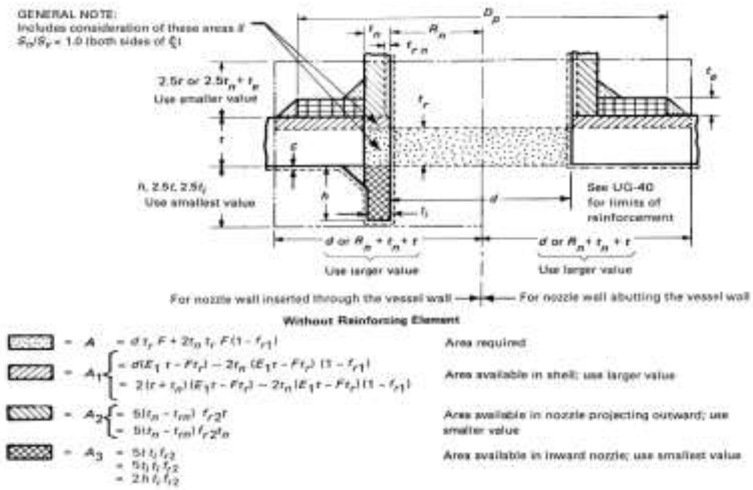

Fig -1: Reinforcement design

Table – 6: Check for reinforcement

Nozzle	1	2	3
	4" sch 40	3" sch 40	20" sch 40
A	7209.3582	5493.8676	40512.7968
A1	153.3906	116.8908	861.9744
A2	136.041460	109.354924	109.3549248
A3	0	0	0
A41	36.2379920	30.1005849	30.10058496
A43	0	0	0

Inlet nozzle 1
Available area = 325.67
Required area = 7209.3582
Available area < required area
Thus, reinforcement is required.
Inlet nozzle 2
Available area = 256.346
Required area = 5493.8676
Available area < required area
Thus, reinforcement is required.
Outlet nozzle 1
Available area = 1001.4299
Required area = 40512.7968
Available area < required area
Thus, reinforcement is required.

Table – 7: Reinforcement Design

Nozzle	1	2	3
	4" sch 40	3" sch 40	20" sch 40
A	7209.3582	5493.8676	40512.7968
A2	588.0214602	507.9949248	587.7229248

A42	2500	2500	3600
A5	4285	2555	36862.656
Available Area	7562.650052	5709.98631	41942.45391
Remark	OK	OK	OK

Available area of all the nozzles is greater than required area, the nozzles & reinforcement are safe in design [3].

4.7 Saddle supports

Table – 8: Saddle Dimensions

Vessel O.D.	Maximum Operating Weight	A	B	C	D	E	F	G	H	Bolt Diameter	#	Approximate Weight/Set
24	15,400	22	21	N.A.	0.5	7	4	0.25	15.2	1	122°	80
30	16,700	27	24			9	4		16.5		120°	100
36	15,700	33	27			12	6		18.8		125°	170
42	15,100	38	30			15			20.0		122°	200
48	25,330	44	33			18			22.3		127°	230
54	25,730	48	36			20			22.7		121°	270
60	38,000	54	39			23			25.0		124°	310
66	36,950	60	42			26			27.2		127°	350
72	50,700	64	45	10		28		0.375	27.6		122°	420
78	56,500	70	48	11	0.75	31	8		29.8		124°	710
84	57,525	74	51	12		33			30.2		121°	810
90	64,200	80	54	13		36			32.5		123°	880
96	65,400	86	57	14		39			34.7		125°	940
102	94,500	92	60	15		42	10	0.500	37.0	1⅛	126°	1,350
108	85,000	96	63	16		44			37.3		123°	1,430
114	164,000	102	66	17		47		0.625	39.6		125°	1,760
120	150,000	106	69	18		49			40.0		122°	1,800
132	127,500	118	75	20		55			44.5		125°	2,180
144	280,000	128	81	22		60			47.0		124°	2,500
156	266,000	140	87	24		66			51.8		126°	2,730

*Table is in inches and pounds and degrees.

Vessel outer diameter = 65 inch

Thus selecting support with vessel O.D. 66 inch which is next standard dimension available [1]

5. ASSEMBLY AND SIMULATION

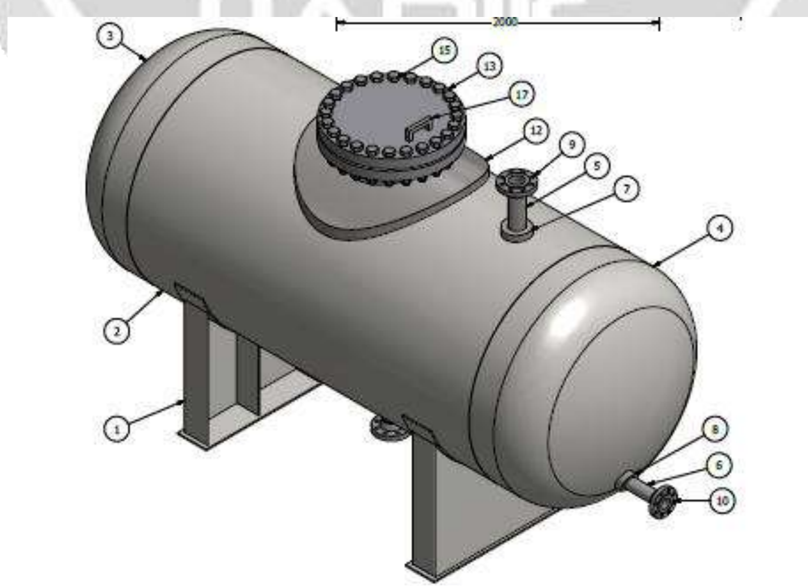

Fig -2: Pressure vessel assembly

Analysis is carried out to check various stresses and forces acting on vessel and magnitude of it at different points on same vessel. [5]

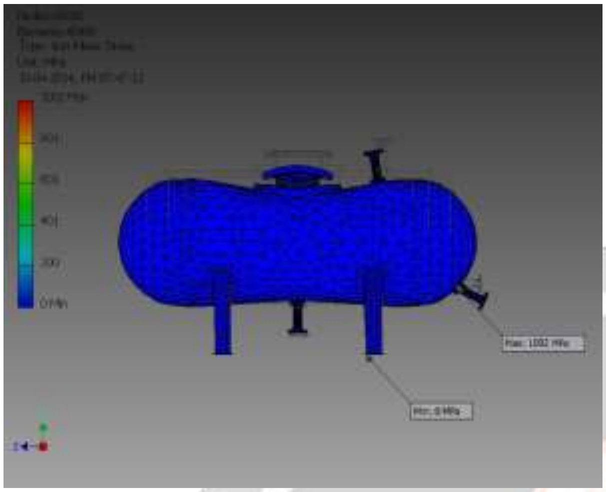

Fig -3: Von Misses stresses

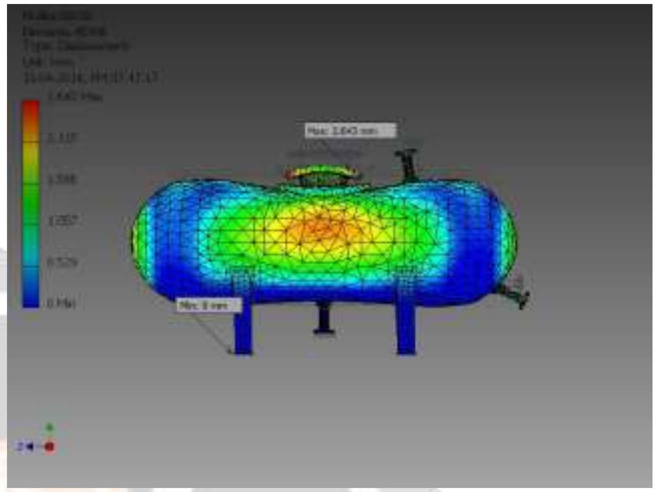

Fig -4: Displacement

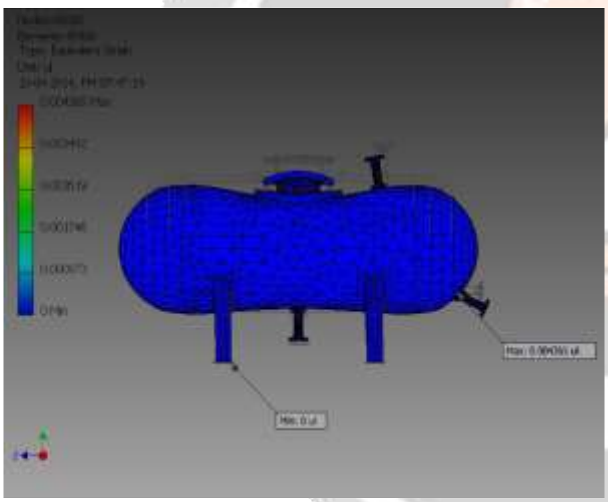

Fig -5: Equivalent Strain

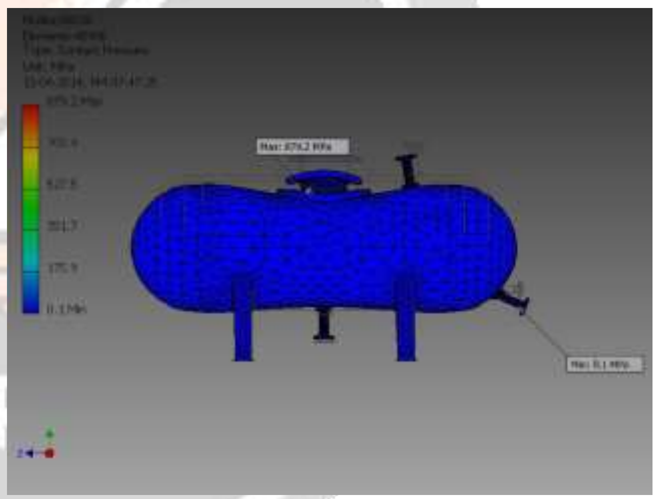

Fig -6: Contact pressure

5.1 Maximum Allowable Working Pressure (MAWP)

The MAWP for a vessel is the maximum permissible pres-sure at the top of the vessel in its normal operating position at a specific temperature, usually the design temperature. When calculated, the MAWP should be stamped on the nameplate.

The MAWP is the maximum pressure allowable in the “hot and corroded” condition. It is the least of the values calculated for the MAWP of any of the essential parts of the vessel, and adjusted for any difference in static head that may exist between the part considered and the top of the vessel. This pressure is based on calculations for every element of the vessel using nominal thicknesses exclusive of corrosion allowance. It is the basis for establishing the set pressures of any pressure-relieving devices protecting the vessel.

The design pressure may be substituted if the MAWP is not calculated. The MAWP for any vessel part is the maximum internal or external pressure, including any static head, together with the effect of any combination of loadings listed in UG-22 which are likely to occur, exclusive of corrosion allowance at the designated coincident operating temperature. The MAWP for the vessel will be governed by the MAWP of the weakest part. [1]

- *MAWP, corroded at Design Temperature P_w.*

Shell:

$$Pw = \frac{S_{DT}Et_{sc}}{R_c + 0.6t_{sc}} \text{ or } \frac{S_{DT}Et_{sc}}{R_o - 0.4t_{sc}}$$

2:1 S.E. Head:

$$Pw = \frac{2S_{DT}Et_{hc}}{D_c + 0.2t_{hc}} \text{ or } \frac{2S_{DT}Et_{hc}}{D_o - 1.8t_{hc}}$$

MAWP for shell = 1780.981678 psi
= 12.27943961 MPa
MAWP for head = 1880.363012 psi
= 12.96464996 MPa

5.2 Maximum Allowable Pressure (MAP)
The term MAP is often used. It refers to the maximum permissible pressure based on the weakest part in the new (uncorroded) and cold condition, and all other loadings are not taken into consideration [1].

- *MAP, new and cold, P_M.*

Shell:

$$P_M = \frac{S_aEt_{sn}}{R_n + 0.6t_{sn}} \text{ or } \frac{S_aEt_{sn}}{R_o - 0.4t_{sn}}$$

2:1 S.E. Head:

$$P_M = \frac{2S_aEt_{hn}}{D_n + 0.2t_{hn}} \text{ or } \frac{2S_aEt_{hn}}{D_o - 1.8t_{hn}}$$

- *Shop test pressure, P_S.*

$$P_s = 1.3P_M \text{ or } 1.3P_W\left[\frac{S_a}{S_{DT}}\right]$$

- *Field test pressure, P_F.*

$$P_F = 1.3P$$

MAP for shell = 1816.811129 psi
= 12.52647504 MPa
MAP for head = 1903.188837 psi
= 13.12202853 MPa
Shop Test Pressure, Ps = 2361.854467 psi
= 16.28441755 MPa
Field Test Pressure, Pf = 2262.58812 psi
= 15.6 MPa

6. CONCLUSIONS

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.
- 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.

7. REFERENCES

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BIOGRAPHIES

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