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Design and Analysis of Pressure Vessel Using PV Elite Software

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ABSTRACT

The pressure vessel is a closed vessel used for storing fluids at higher pressures different from the ambient pressure. As the pressure vessels are stored under high pressure, there is a potential impact for an accident, so pressure vessel has to be design accurately and safely. A horizontal pressure vessel is designed with reference to the ASME Sec.VIII Division1 Codes and standard. AutoCAD is used for drafting the scaled construction drawing of pressure vessel. These construction drawings are used as reference for the remaining process. As the mathematical calculation of pressure vessel becomes tedious, graphical based software PV Elite was used for analysis on shell, head, nozzle and saddle supports.

Keywords *- Pressure vessel, ASME Section VIII division 1, AutoCAD, PV Elite software.*

1. INTRODUCTION

Pressure Vessel is a container for either internal or external pressures in the vessel. The internal or external pressures obtained in the vessel are through by an external source or because of applying heat through a direct or indirect source or by any combinational means. Pressure vessels may differ greatly, but their name specifies that these vessels are under pressure with an addition of chemicals or temperature. Storing harmful chemicals in pressure vessels is an important segment in every chemical industry because of its safety. For any pressure vessel design, a primary consideration is its safety. As the pressure vessels are used for storing, there is a potential impact chance for an accident. ASME codes of unfired pressure vessel rules and standards are used to construct this pressure vessel. Beside of this, local codes and standards are to be used during the construction of the pressure vessel. Based on company point of view, the design should achieve with adequate standards, safety and at low cost.

1.1 Objective

The main purpose of this project is to design a safe pressure vessel based on standards and codes. The objectives are:

- Selecting a pressure vessel and designing it according to ASME codes and standards.
- To create a drafting design of pressure vessel using Auto CAD software.
- To analyses the safety parameters of pressure vessel using PV Elite software.

2. METHODOLOGY

Many standards of engineering have possessed information regarding the design and construction for a pressure vessel. The emphasis has been made through the selection of standards to manufacturing a pressure vessel has to obey and compile within the entirety and no other different standards are not to be used to design.

2.1 Selection of Codes

For pressure vessel designing, the code selection is very important for achieving safe pressure vessel condition under a reference guide. Pressure vessels are usually designed according to the ASME sec VIII codes. Div-1 is about the pressure vessel rules for construction; Division2 is about the alternative rules for pressure vessel.

2.2 Material Specifications

Based on the design requirements the appropriate materials are selected. The materials used for the manufacturing of this pressure vessel have to satisfy the requirements of the specified design codes and its details are:

Components	Material Used
Shell	SA516 Gr60
Dished End	SA516 Gr60
Nozzle	SA106 GrB

Table 1: Part Material.

The chemical and mechanical composition requirement of shell and dished end heads is as per table 2 and 3.

Composition	C	Si	Cu	Ni	Mn	Mo	Cr	Nb	P	S	Al	Ti	V
Percentage	0.18	0.4	0.3	0.3	0.95/1.5	0.08	0.3	0.01	0.015	0.008	0.02	0.03	0.02

Table 2: Chemical Composition ASME SA516 Gr60.

PROPERTIES	VALUE
Tensile Strength N/mm^2	415 - 580
Yield Stress / min N/mm^2	265

Table 3: Mechanical Values of ASME SA516 Gr 60.

The chemical and mechanical composition requirement of Nozzle is as per table 4 and 5.

Composition	C	Si	Cu	Ni	Mn	Mo	Cr	P	S	V
Percentage	0.18	0.10	0.40	0.40	0.29/1.06	0.15	0.40	0.035	0.035	0.08

Table 4: Chemical Composition of SA 106 Gr B.

Properties	Value
Tensile Strength N/mm^2	415
Yield Stress / min N/mm^2	240

Table 5: Mechanical Values of SA 106 Gr B.

3. RESULTS AND DISCUSSION

A pressure vessel horizontally placed on saddle supports was designed according to the design data input. SRAAC industry was planning to design a pressure vessel, as in need of a pressure vessel for storing chemical. As chemical storing pressure vessels are potentially hazardous equipment so designing and analysis should be done according to an approved code. The designing of a pressure vessel based on ASME code standards according to the company requirement. The Auto CAD view of pressure vessel is shown in fig 1. The design and analysis with ASME codes using PV Elite software has been carried out. The design of pressure vessel is shown in fig 2. The analysis results are shown below.

3.1 Geometry Modeling

Design Code: ASME SEC VIII DIV.1
Capacity: $50m^3$
Design Pressure: 12.5 kg/cm^2g
Design Temperature: 60^0C
Working Pressure: 10 Kg/cm^2g
Working Temperature: 43^0C
Corrosion Allowance: 3mm
Joint Efficiency: 1.0
Operating Weight: 57794kgs
Total Empty Weight: 17350kgs
Weight Full of Water: 66050kgs
Density: 875 kg/cm^3
Wind Velocity: 140.4 Km/hr (IS: 875, Part – 3)
Radiography: 100 %
Hydro Test Pressure: 16.25 Kg/cm^2g

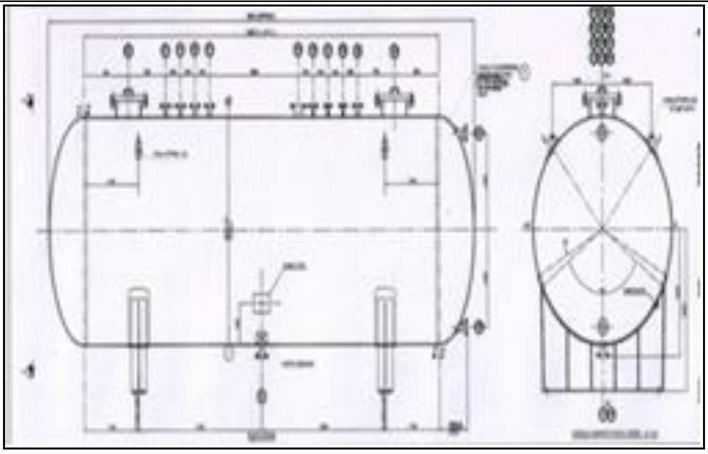

Fig 1: Pressure Vessel.

3.2 Internal Pressure Calculation

According to ASME Section VIII, Division 1 2010 EDITION, 2011a ADDENDA

Ellipsoidal head from 10 to 20, SA516 Gr 60, UCS66 CurveD at 55⁰C

Thickness of internal Pressures (tr) = $\frac{(P*D*K)}{(2*SE-0.2P)}$ Appendix1-4 (c)

$$=\frac{(2*1202.20*1*20.00)}{(1*2800.00+0.2*20.00)}$$

= 17.5615mm

For given thickness, max. Allowable working pressures, corrode (MAWP)

$$=\frac{(2*S*E*t)}{(K*D+0.2t)}$$

Appendix1-4 (c)

$$=\frac{(2*1202.20*1*20.00)}{(1*2800.00+.02*20.00)}$$

= 17.150kgf/cm^2

Max. Allowable pressures at New and Cold (MAPNC)

$$=\frac{(2*S*E*t)}{(K*D+0.2t)}$$

$$=\frac{(2*1202.20*1*20.00)}{(1*2800.00+.02*20.00)}$$

= 17.150kgf/cm^2

For given pressure and thickness, actual stresses corroded (Sact)

$$=\frac{(P*(K*D+0.2t))}{(2*E*t)}$$

$$=\frac{(12.50*(0.997*2806.00+0.2*17.00))}{(2*1*17.00)}$$

= 1029.936 kgf/cm^2

Required thickness of straight flanges,

$$=\frac{(P*R)}{(S*E-0.6P)}+C$$

Per UG-27 (c)(1)

$$=\frac{(12.5*1403)}{(1202.20*1.00-0.6*12.50)}+3$$

=17.679mm

Cylindrical shell from 20 to 30 SA516 Gr 60, UCS66 CurveD at 55^0C,

Thickness of internal pressure (tr),

$$= \frac{(P * D)}{(S * E - 0.6P)} \quad \text{Per UG27 (c)(1)}$$

$$= \frac{(12.5 * 1403)}{(1202.20 * 1.00 - 0.6 * 12.50)}$$

= 17.6788mm

For given thickness, max. Allowable working pressure, corrode (MAWP),

$$= \frac{(S * E * t)}{(R + 0.6t)} \quad \text{Per UG27 (c)(1)}$$

$$= \frac{(1202.25 * 1 * 18.00)}{(1400.00 + 0.6 * 18.00)}$$

=15.339kgf/cm^2

For given thickness and pressure, actual stress, corroded (Sact)

$$= \frac{\left(P * (R + 0.6t)\right)}{(E * t)}$$

$$= \frac{\left(12.50 * (1403.00 + 0.6 * 15.00)\right)}{(1.00 * 15.00)}$$

= 1176.667 kgf/cm^2

Ellipsoidal Head from 30 to 40 SA516 Gr 60, UCS66 Curve D at 55^0C,

Thickness of internal pressure (tr)

$$= \frac{(P * D * K)}{(2 * SE - 0.2P)} \quad \text{Appendix1-4(c)}$$

$$= \frac{(12.50 * 2806.00 * 0.97)}{(2 * 1202.25 * 1.00 - 0.2 * 12.50)}$$

= 17.5609mm

For given thickness, Max. Allowable working pressure, corrode (MAWP)

$$= \frac{(2 * S * E * t)}{(K * D + 0.2t)} \quad \text{per appendix1-4(c)}$$

$$= \frac{(2 * 1202.25 * 1 * 17.00)}{(0.997 * 2806 + 0.2 * 17.00)}$$

=14.591kgf/cm^2

Max. Allowable pressure for New and Cold (MAPNC),

$$= \frac{(2 * S * E * t)}{(K * D + 0.2t)} \quad \text{per appendix1-4(c)}$$

$$= \frac{(2 * 1202.20 * 1 * 17.00)}{(0.997 * 2806 + 0.2 * 17.00)}$$

=17.150kgf/cm^2

For given thickness, actual stress, corrode (Sact),

$$= \frac{\left(P * (K * D + 0.2t)\right)}{(2 * E * t)}$$

$$= \frac{\left(12.50 * (0.997 * 2806.00 + 0.2 * 17.00)\right)}{(2 * 1 * 17.00)}$$

= 1029.936 kgf/cm^2

Required thickness of straight flanges,

$$= \frac{(P * R)}{(S * E - 0.6P)} + C \quad \text{Per UG 27 (c)(1)}$$

$$= \frac{(12.5 * 1403)}{(1202.20 * 1.00 - 0.6 * 12.50)} + 3$$

= 17.679 mm

3.3 External Pressure Calculation

Ellipsoidal head from 10 to 20 Ext. Chart CS2 at 0^0C

Module of elasticity in chart CS2 at 0^0C = 0.204E+07 kgf/cm^2

Max. Allowable external pressure Results (MAEP),

Tca	OD	D / t	Factor A	B
17.00	2840.00	167.06	0.0008314	809.83

EMAP = B/ (KO*(D/t)) = 809.8262/ (0.90*167.0588) = 5.3862 kgf/cm^2

Cylindrical shell from 20 to 30 Ext. Chart CS2 at 0^0C

Module of Elasticity in chart CS2 at 00C = 0.204E+07 kgf/cm^2

Max. Allowable external pressure Results (MAEP),

Tca	OD	SLEN	D /t	L / D	Factor A	B
15.00	2836.00	7766.67	189.07	2.7386	0.0001801	183.58

EMAP = (4*B) / (3*(D/t)) = (4*183.5779)/ (3*189.0667) = 1.2946 kgf/cm^2

Max. Stiffened length Results (Slen),

Tca	OD	SLEN	D / t	L / D	Factor A	B
15.00	2836.00	7766.67	189.07	2.7386	0.0001801	183.58

EMAP = (4*B)/ (3*(D/t)) = (4*183.5779)/ (3*189.0667) = 1.2946 kgf/cm^2

Ellipsoidal head from 10 to 20 Ext. Chart CS2 at 0^0C

Module of Elasticity in chart CS2 at 0^0C = 0.204E+07 kgf/cm^2

Max. Allowable external pressure Results (MAEP),

Tca	OD	D / t	FACTOR A	B
17.00	2840	167.06	0.0008314	809.83

EMAP = B / (Ko*(D / t)) = 809.8262/ (0.90 *167.0588) = 5.3862 kgf/cm^2

3.4 Saddle Reaction Results Due To Wind or Seismic

Force at saddle reaction due to wind Ft (Fwt)

= Ftr * ((Ft / Num of saddles) + Z Force Load) * (B/E)

= 3*(556 / 2+0) *(1700.0001/2470)

= 574.0 kgf

Force at saddle reaction due to wind Fl or friction (Fw1)

=Max (F1, Friction Load, Sum of X Forces)* (B/Ls)

= Max (195.15, 0, 0) *(1700.0001 /4000.0002)

=82.9kgf

Force at saddle reaction due to earthquake Fl or friction (Fs1)

= Max (F1, Friction Force, Sum of X Forces) *(B/Ls)

=Max (0.17, 0, 0) *(1700.0001/4000.0002)

= 0.1 kgf

Force at saddle reaction due to earthquake Ft (Fst)

= Ftr * (Ft + Z Force Load) *(B/E)

= 3.00*(0+0) *1700.0001/2470.00

= 0.0 kgf

Results of load combination for Q of wind or seismic (Q)

= Saddle Loads + Max (Fw1, Fwt, Fs1, Fst)

= 6757+ Max (82, 573, 0, 0)

= 7331.1 kgf

3.5 Nozzle Calculation

ASME Sec.VIII Div.1 Codes: 2010, 2011a, UG37 to UG45

Reinforcement Computing, Description: Nozzle

Actual inside diameter for calculations 42.850 mm

Actual thickness for calculation 8.732 mm

Cylindrical shell required thickness, Tr (internal pressure)

$$= \frac{(P * R)}{(S * E - 0.6P)}$$

$$= \frac{(12.50 * 1403.00)}{(1202.00 * 1.00 - 0.6 * 12.50)}$$

=14.6788 mm

Nozzle wall required thickness, Trn (internal pressure)

$$= \frac{(P * R_o)}{(S * E - 0.6P)}$$

$$= \frac{(12.50 * 24.42)}{(1202.00 * 1.00 - 0.6 * 12.50)}$$

= 0.2555 mm

UG45 min. thickness for nozzle neck requirement: Internal pressures

Internal or external wall thickness, tra = 3.4882 mm

Thickness for wall Per UG16b, tr16b = 4.5000 mm

Shell or head thickness for wall internal pressures, trb1 = 17.6788 mm

Thickness for wall, trb1 = Max (trb1, tr16b) = 17.6788 mm

Thickness for wall, trb2 = Max (trb2, tr16b) = 4.5000 mm

Thickness for wall Per UG4, tb3 = 8.258 mm

Thickness of nozzle candidate determined (tb)

= min (tb3, max (tb1, tb2))

=min (8.258, max (17.679, 4.5))

=8.258 mm

Nozzle necks min. wall thickness (tUG45)

= max (ta, tb) = max (3.4882, 8.2578) = 8.2578 mm

Thickness of available nozzle neck = 0.875 * 13.487 = 11.801 mm

Minimum Design Metal Temperature (MDMT) For Nozzle Junction Calculation

Governing Thickness, tg = 7.645, tr =0.256, c=3.000mm, E* =1.00,

Stress ratio;

$$= \frac{(tr * E)}{(tg - c)}$$

$$= \frac{0.256 * 1.00}{7.645 - 3}$$

=0.055

Temperature reduction = 78^{0}C

MDMT between Nozzle neck and flange welds	Curve: B
Minimum metal temperature without impact per UCS66	- 29^{0}C
Minimum metal temperature at reqd. Thk. (UCS66.1)	-104^{0}C

MDMT between nozzle and shell or head welds of the nozzle (UCS66(a)1(b))	Curve B
Minimum metal temperature without impact per UCS66	-29^{0}C
Minimum metal temperature at reqd. Thk. (UCS66.1)	-104^{0}C
At all sub joints for this junction, governing MDMT	-104^{0}C
Temperature reduction in ANSI Flange MDMT per UCS66.1	
Unadjusted MDMT of ANSI B 16.5/ 47 flange per UCS66c	-290C
Temp. reduction in Flange MDMT per UCS66(b)1(b)	-1040C
Temp. reduction in Flange MDMT per UCS66(b)1(c)	-1040C

Stress reduction Ratio Per UCS66 (b)1(b)

$$= \frac{\text{Design Pressure}}{\text{Ambient Rating}}$$

$$= \frac{12.50}{52.11}$$

= 0.24

<u>SUMMARY OF NOZZLE PRESSURE/ STRESS RESULTS</u>

Allowed Local Primary Membrane Stress, Sallow = 1803.37 kgf/cm²

Stresses of primary membrane for local, PL = 1169.17 kgf/cm²

Max. Allowable pressures working, Pmax = 12.85 kgf/cm²

Calculation For Weld Sizes, Description: Nozzle
Nozzle or shell weld intermediate calculations Tmin = 5.7376 mm
Result Per UW16.1

	Thickness required	Actual Thickness
Nozzle Welds	4.0163 = 0.7* tmin	6.7342 =0.7*Wo mm

Nozzle N1 Max. Allowable pressure at this location
Converge Maximum Allowable pressures for operating case 12.772 kgf/cm^2

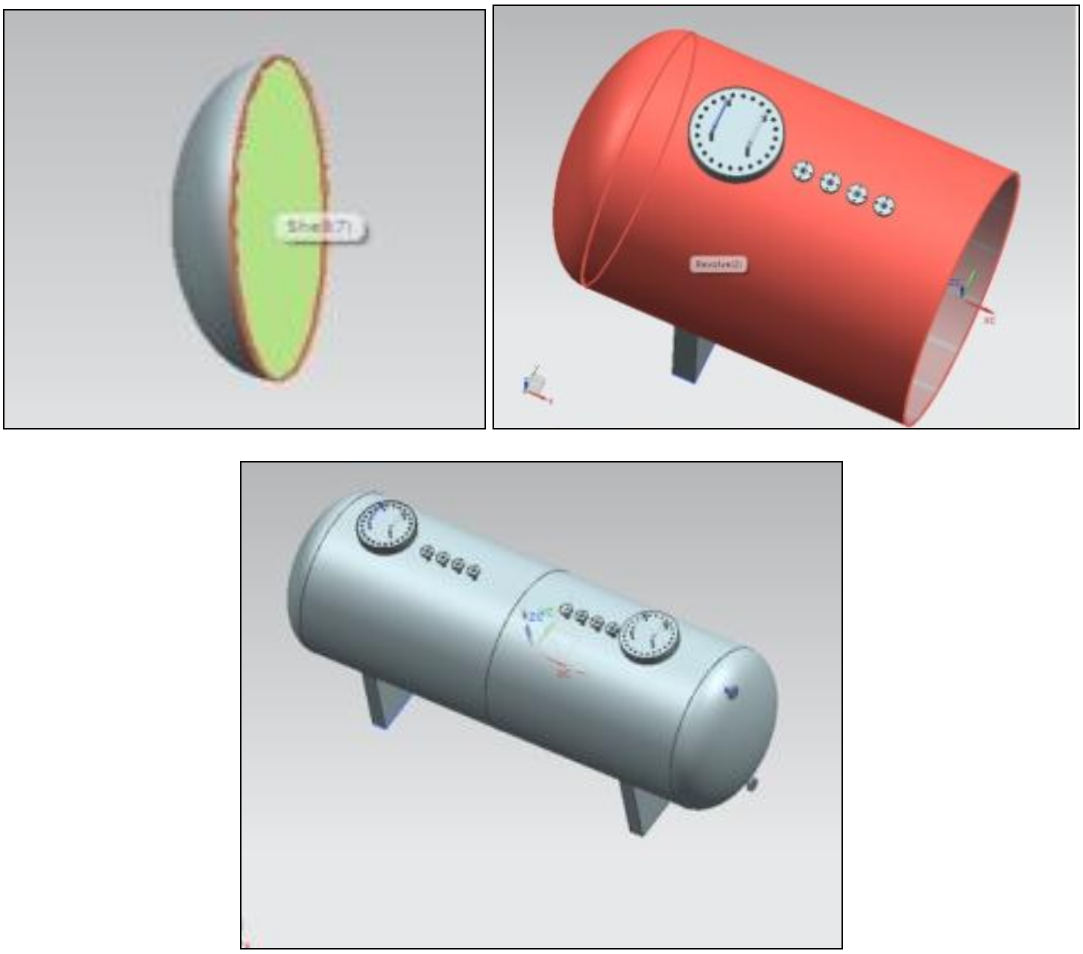

Fig 2: Pressure Vessel in PV Elite Software.

4. CONCLUSION

The designing of safe pressure vessel and its quality testing according to company requirements and ASME Code standards is been successfully completed. First for every pressure vessel safety is the primary aspect and it is acquired by following the design rules and procedures. PV Elite software is fast and produces accurate analysis within less time. Analysis of pressure vessel is much easier in PV Elite software. PV Elite performs calculations as the data is typed in and the results are easy to read and understand.

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