Design of Pressure vessel for the Hydrocarbon release from the Vent Header Lines

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Abstract – *Flaring is the controlled burning of natural gas in the course of routine oil and gas production operations. This burning occurs at the end of flare stock or boom. Therefore, the paper deals with the design of pressure vessel for the collection of hydrocarbon release from the vent header lines of pilot plant. The components of a horizontal vessel like shell, head, and supports have been designed as per the ASME code section VIII Div. 1. It deals with suitable material selection and manual design of the vessel components along with validation from ANSYS. In order to avoid the release of hazardous chemicals or liquids, a pressure vessel is designed using section VIII, Div. 1(Ed 2015). Hydrostatic and radiographic tests have been performed on the vessel to determine the ability of the vessel to withstand various pressures.*

***Key Words*: ASME codes & standards, shell, head, saddle, Tests, APV**

1.INTRODUCTION

Vessels and tanks that carry, store, or receive fluids under pressure are called pressure vessels. The inside pressure is usually higher than the outside pressure except for some isolated conditions. Pressure vessels often combine high pressures together with high temperatures and in some cases flammable fluids. Pressure vessel V300 has been designed for a pilot plant. It is to be designed for collecting hydrocarbon liquids emerging from vent header lines. Hence the material selected must be given either corrosion allowance or the material must be resistant to corrosion caused by hydrocarbons, whichever is safe and feasible. Hence, the economical design of a pressure vessel and its components has been made in the subsequent sections.

1.1 Literature Review

The book consists of design procedure for the different components of the pressure vessel including the information about the materials to be used for construction as per the application. [1]

This section of the ASME code deals with the rules for the construction of the pressure vessel from the material procurement to the manufacturing and testing [2]

This section deals with the material specifications for the construction of pressure vessel and the part D consists of the properties of the materials. [3]

Stiffening rings are the popular way of controlling stress in certain locations and they prove to be cost effective. The paper deals with quantifying the role of stiffening rings. [4]

1.2 Introduction to ASME code

The organization of the ASME pressure vessel code is as follows:

Section II: Material Specification:

- Ferrous Material Specifications – Part A
- Non-ferrous Material Specifications – Part B
- Specifications for Welding Rods, Electrodes, and Filler Metals –Part C
- Properties – Part D

Section VIII:

- Division 1: Pressure Vessels – Rules for construction.
- Division 2: Pressure Vessels – Alternative Rules.

1.3 Guidelines from ASME

- UG-1 Scope
- UG-4 General Materials
- UG-27 (C) Cylindrical Shells
- UG-32 F Ellipsoidal Heads
- UG 40 Limits Of Reinforcement
- UG-45 Nozzle Neck Thickness

1.4 Nomenclature

APV	:	American Pressure Vessel
ASME	:	American Society of Mechanical Engineers
UG	:	Unfired Vessels General guidelines
V300	:	Liquid collection tank for collecting hydrocarbon release from vent header lines

2. PROBLEM DEFINITION

To design and manufacture the pressure vessel for hydrocarbon toxic release from the vent header lines for the design pressure and temperature of **10 kg/cm²** and **70⁰C** respectively with selection of a suitable material which will give safety and is most economical and to design the pressure vessel to avoid environmental contamination from a pilot plant including the safety features. It also aims at designing shell, head, supports, openings and reinforcements.

3. MATERIAL SELECTION FOR V300

Several of materials have been use in pressure vessel fabrication. The selection of material is based on the design requirement. All the materials used in the manufacture of the receivers shall comply with the requirements of the relevant design code.
The material **SA240GR316** for manufacturing of V300 has been selected. Grade 316 is immune from sensitization (grain boundary carbide precipitation). SA240 GR316 is a molybdenum-bearing austenitic stainless steel which is more resistant to general corrosion and pitting/ crevice corrosion than the conventional chromium-nickel austenitic stainless steels such as Type 304. A case study of the chemical composition would reveal that Type 316 is considerably more resistant than any of the other chromium-nickel types to solutions of Sulphuric acid. At temperatures as high as 120°F (49°C), Types 316 and 317 are resistant to concentrations of this acid up to 5 percent. At temperatures under 100°F (38°C), this alloy has excellent resistance to higher concentrations.
Where condensation of sulphur-bearing gases occurs, this alloy is much more resistant than other types of stainless steels. In such applications, however, the acid concentration has a marked influence on the rate of attack and should be carefully determined. The molybdenum-bearing Type 316 stainless steel also provides resistance to a wide variety of other environments.
The material properties have been studied from Section II of the code, Part D. [3]

4. INPUT PARAMETERS

- Max. operating temperature = 50^0C
- Max. Operating pressure = $3.5Kg/cm^2$
- Design temperature = 70^0C
- Design pressure = 10Kg
- Relief set pressure = Atm.
- Material = SA240GR316
- Orientation = Horizontal
- Shell diameter = 0.75m
- Length of shell = 2.5m
- Nominal volume = $1.5m^3$

4.1 Nozzles and Connections

Liquid Inlet	=	2 inches
Make-Up	=	3 inches
Spare	=	2 inches
Liquid Inlet	=	2 inches
Vent	=	2 inches
Liquid Inlet	=	2 inches
Liquid Indicator (high)	=	1 inch
Liquid Indicator (low)	=	1 inch
Spare	=	1 inch

The Design pressure is taken as equal to maximum operating pressure including static head + 10%. Normal design pressure = 1.1Pmax

5. DESIGN OF SHELL

The Shell is the cylindrical component or the main body of the vessel which encloses the contents of the vessel. It is designed as per the ASME code section VIII, UG-27 for the shells subjected to internal pressure and UG-28 for the shells subjected to external pressure.[2]

Cylindrical Shell Design Information

Design Pressure:	1.0000 MPa	Design Temperature:	70 °C
Static Head:	0.0000 MPa	Long. Joint Efficiency:	85 %
Shell Material:	SA-240 Gr. 316	Factor B Chart:	HA-2
		Material Stress (hot):	138.00 MPa
Shell Length:	2440.00 mm	Material Stress (cold):	138.00 MPa
		Compressive Stress:	71.23 MPa
Corrosion Allowance:	0.00 mm	Actual Circumferential Stress:	111.00 MPa
External Corrosion Allowance:	0.00 mm	Actual Longitudinal Stress:	54.91 MPa
Inside Diameter (new):	750.00 mm	Extreme Fiber Elongation:	0.53 %
Inside Diameter (corroded):	750.00 mm	Specific Gravity:	1.00
Shell Surface Area:	5.81 sq. m	Weight of Fluid:	1077.96 kg
Shell Estimated Volume:	1.08 cu. m	Total Flooded Shell Weight:	1263.61 kg
Circ. Joint Efficiency:	85 %	Shell Weight:	185.65 kg

Minimum Design Metal Temperature Data

Minimum Design Metal Temperature: -5 °C
Other Exemption

External Pressure Data

Design Pressure (Pa):	0.1000 MPa	Design Temperature:	70 °C
Dimension L:	2440.00 mm		
Ext. Minimum t:	3.82 mm	Ext. Nominal t:	4.00 mm
Minimum L/Do:	3.2190	Nominal L/Do:	3.2190
Minimum Do/t:	198.4290	Nominal Do/t:	189.5000
Minimum Factor A:	0.0001583	Nominal Factor A:	0.0001731
Minimum Factor B:	14.91 MPa	Nominal Factor B:	16.31 MPa

Design Thickness Calculations

Longitudinal Stress Calculations per Paragraph UG-27(c)(2)

$$t = \frac{PR}{2SE + 0.4P} = \frac{1.0000 * 375.00}{2 * 138.00 * 0.85 + 0.4 * 1.0000} = 1.60 + 0.00_{(corrosion)} + 0.00_{(ext.\ corrosion)} = \text{minimum of } \mathbf{1.60}\ mm$$

Circumferential Stress Calculations per UG-27(c)(1)

$$t = \frac{PR}{SE - 0.6P} = \frac{1.0000 * 375.00}{138.00 * 0.85 - 0.6 * 1.0000} = 3.21 + 0.00_{(corrosion)} + 0.00_{(ext.\ corrosion)} = \text{minimum of } \mathbf{3.21}\ mm$$

Maximum External Pressure Calculation per Paragraph UG-28

$$Pa\ (\text{using } nominal\ t) = \frac{4B}{3(D_0 / t)} = \frac{4 * 16.31}{3 * (758.00 / 4.00)} = \text{maximum external pressure of } \mathbf{0.1148}\ MPa$$

External loads do not control design.

Nominal Shell Thickness Selected = **4.00** mm

Fig. 1 APV calculations for design of Shell

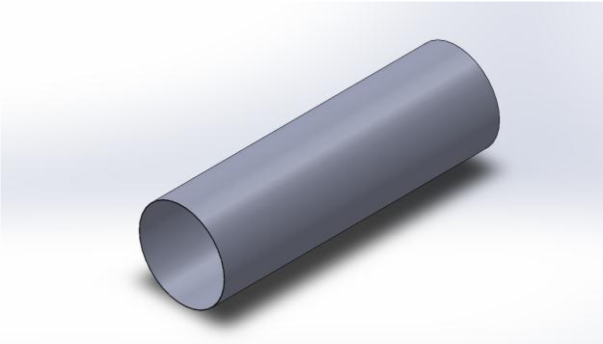

Fig. 2 Model of Shell on Solidworks

6. DESIGN OF ELLIPSOIDAL HEAD

The head is the part of the vessel that encloses the vessel shells from both sides.
Ellipsoidal Heads(2:1) : pressures over 1.5 N/mm²
An acceptable approximation of a 2:1 ellipsoidal head is one with a knuckle radius of 0.17D and a spherical radius of 0.90D[2]

Ellipsoidal Head Design Information

Design Pressure:	0.1000 MPa	Design Temperature:	70 °C
Static Head:	0.0000 MPa	Joint Efficiency:	85 %
Head Material:	SA-240 Gr. 316	Factor B Chart:	HA-2
		Material Stress (hot):	138.00 MPa
Corrosion Allowance:	0.00 mm	Material Stress (cold):	138.00 MPa
External Corrosion Allowance:	0.00 mm	Actual Head Stress:	17.66 MPa
Head Location:	Right	Straight Flange :	0.00 mm
Inside Diameter (new) :	750.00 mm	Head Depth (h) :	187.50 mm
Inside Diameter (corroded) :	750.00 mm	Thin Out :	1.50 mm
$K = \frac{1}{6}[2 + (D/2h)^2]$:	1.00		
Extreme Fiber Elongation:	2.34 %		
Head Surface Area:	0.62 sq. m	Specific Gravity:	1.00
Head Estimated Volume:	0.06 cu. m	Weight of Fluid:	55.22 kg
Head Weight:	19.90 kg	Total Flooded Head Weight:	75.13 kg

Minimum Design Metal Temperature Data

Minimum Design Metal Temperature: -5 °C
Other Exemption

External Pressure Data

Design Pressure (Pa):	0.1000 MPa	Design Temperature:	70 °C
Ext. Minimum t:	3.47 mm	Ext. Nominal t:	4.00 mm
Minimum t - Ca - ext. Ca - Thin Out:	1.97 mm	Nominal t - Ca - ext. Ca. - Thin Out:	2.50 mm
Minimum Factor A:	0.0003650	Nominal Factor A:	0.0004632
Minimum Factor B:	34.43 MPa	Nominal Factor B:	43.71 MPa

Design Thickness Calculations

Design Thickness Calculations per Appendix 1-4(c)

$$t = \frac{PDK}{2SE - 0.2P} = \frac{0.1000 * 750.00 * 1.00}{2 * 138.00 * 0.85 - 0.2 * 0.1000}$$

= Greater of (0.32(Calc.), 1.50(Min. t)) + $0.00_{(corrosion)} + 0.00_{(ext.\ corrosion)} + 1.50_{(thin\ out)}$ = minimum of **3.00** mm

Maximum External Pressure Calculation per Paragraph UG-33

$$\textbf{Pa}\ (\text{using } nominal\,t) = \frac{B}{\frac{Ro}{t}} = \frac{43.71}{\frac{674.62}{2.50}}$$ = maximum external pressure of **0.1620** MPa

$$t_{min(external)} = \frac{1.67PDK}{2SE - 0.2(1.67P)} = \frac{1.67 * 0.1000 * 750.00 * 1.00}{2 * 138.00 * 1.0 - 0.2 * 1.67 * 0.1000}$$ = Greater of(0.45(Calc),1.50(MinT))

$+ 0.00_{(corrosion)} + 0.00_{(ext.\ corrosion)} + 1.50_{(Thinout)}$

= minimum of **3.00 mm**

Nominal Head Thickness Selected = **4.00** mm
Minimum Thickness after forming, t_s (uncorroded) = **2.50** mm

Fig. 3 APV calculations for design of Head

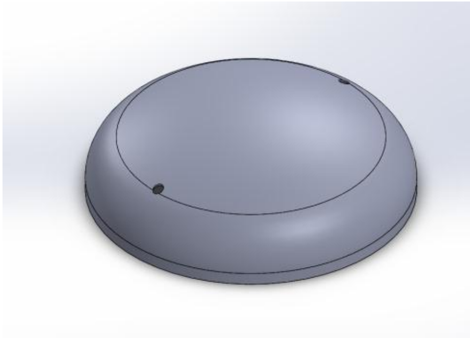

Fig. 4 Model of Ellipsoidal head on Solidworks

6. 1 Stiffening Rings under External Pressure

External stiffening rings will be attached to the shell by welding or brazing if required
The available moment of inertia of a circumferential stiffening ring shall not be less than that determined by one of the following two formulas as mentioned in the UG-30.
Compare the required moment of inertia with the actual moment of inertia of selected member. If actual exceeds that which is required then design is acceptable.
Here, for V300, the value of external pressure is under acceptable limits. Hence, no stiffener is added in the design[2] [4]

7. DESIGN OF OPENINGS AND REINFORCEMENTS

The nozzles are designed for the stresses and reinforcement material is provided for,
(a)The weakening due to the hole made for nozzle has to be compensated by sufficient additional material.
(b)The reinforcement material should be placed immediately adjacent to the hole but suitably disposed in profile and contour, so as not to introduce any stress concentration.
Holes cause discontinuity in the vessel wall which increases the stress concentration at the edge of the holes that can be reduced by increasing the thickness of the vessel in the vicinity of the nozzle. [2]

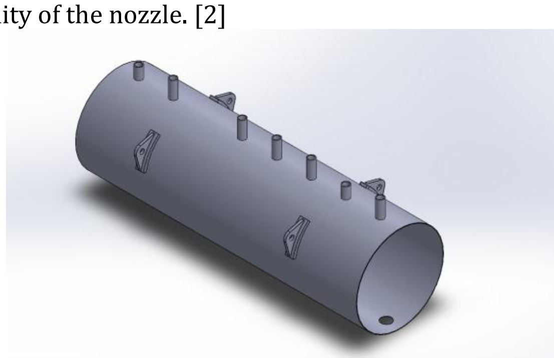

Fig. 5 Model of shell on Solidworks with nozzles and lifting Lugs

Nozzle Design Information

Design Pressure:	1.0000 MPa	Design Temperature:	70 °C
Static Head:	0.0000 MPa	Nozzle Efficiency (E):	100 %
Nozzle Material:	SA-312 TP316	Joint Efficiency (E_1):	1.00
		Factor B Chart:	HA-2
External Projection:	100.00 mm	Allowable Stress at Design Temperature (S_n):	138.00 MPa
Internal Projection:	0.00 mm	Allowable Stress at Ambient Temperature:	138.00 MPa
Inside Corrosion Allowance:	0.00 mm	Correction Factor (F):	1.00
External Corrosion Allowance:	0.00 mm	Nozzle Path:	None
Nozzle Pipe Size:	50	Nozzle Pipe Schedule:	40
Nozzle ID (new):	52.48 mm	Nozzle Wall Thickness(new):	3.91 mm
Nozzle ID (corroded):	52.48 mm	Nozzle Wall Thickness(corroded):	3.91 mm
External Limit of Reinforcement:	9.78 mm	Upper Weld Leg Size(Weld 41):	3.91 mm
Internal Limit of Reinforcement:	9.78 mm	Internal Weld Leg Size(Weld 43):	0.00 mm
Parallel Limit of Reinf (2Lpar):	104.96 mm	Outside Groove Weld Depth:	4.00 mm

Minimum Design Metal Temperature

Minimum Design Metal Temperature: -5 °C
Other Exemption

Host Component: Shell 1 - Shell

Material:	SA-240 Gr. 316	Shell wall thickness(new):	4.00 mm
Material Stress(S_v):	138.00 MPa	Shell wall thickness(corroded):	4.00 mm

Nozzle Detail Information

Backing strip if used may be removed after welding

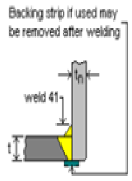

Fig. UW-16.1 (c)

Upper Weld Leg Size(Weld 41): 3.91 mm
Nozzle Wall Thickness(t_n): 3.91 mm
Outside Groove Weld Depth: 4.00 mm

Nozzle passes through the vessel, attached by a groove weld.
Pipe Size: 50 Schedule: 40
Nozzle is adequate for UG-45 requirements.
Opening is adequately reinforced for Internal Pressure.
Opening is adequately reinforced for External Pressure.
Reinforcement calculations are not required per UG-36(c)(3)(a)See Uw-14 for exceptions.
Weld Strength Paths are adequate.

Fig. 6 APV Sheet for Nozzle (A1-Liquid Inlet) Calculations

Required Shell Thickness per Paragraph UG-37(a)

$$tr = \frac{PR}{SE - 0.6P} = \frac{1.0000 * 375.00}{138.00 * 1 - 0.6 * 1.0000} = \mathbf{2.73}\text{ mm}$$

Nozzle Required Thickness Calculations

Required Nozzle Thickness for Internal Pressure per Paragraph UG-37(a)

$$trn = \frac{PRn}{SE - 0.6P} = \frac{1.0000 * 26.24}{138.00 * 1 - 0.6 * 1.0000} = \mathbf{0.19}\text{ mm}$$

Required Nozzle Thickness for External Pressure per Paragraph UG-37(a)

$$trn = \frac{3 * Do * Pa}{4B} = \frac{3 * 60.30 * 0.1000}{4 * 20.46} = \mathbf{0.24}\text{ mm}$$

Strength Reduction Factors

$$fr1 = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{138.00}{138.00}, 1.0000\right) = 1.0000 \qquad fr2 = \min\left(\frac{Sn}{Sv}, 1.0000\right) = \min\left(\frac{138.00}{138.00}, 1.0000\right) = 1.0000$$

UG-45 Thickness Calculations

Nozzle Thickness for Pressure Loading (plus corrosion)

$$t_a = \frac{PRn}{SE - 0.6P} + Ca + ext.\ Ca = \frac{1.0000 * 26.24}{138.00 * 1.00 - 0.6 * 1.0000} + 0.00 + 0.00 = \mathbf{0.19}\text{ mm}$$

Nozzle Thickness for Internal Pressure (plus corrosion) Based on Host

$$t_{b1} = \frac{PR}{SE - 0.6P} + Ca + ext.\ Ca = \frac{1.0000 * 375.00}{138.00 * 1 - 0.6 * 1.0000} + 0.00 + 0.00 = \mathbf{2.73}\text{ mm}$$

Nozzle Thickness for External Pressure (plus corrosion) Based on Host

$$t_{b2} = \frac{PR}{SE - 0.6P} + Ca + ext.\ Ca = \frac{0.1000 * 375.00}{138.00 * 1 - 0.6 * 0.1000} + 0.00 + 0.00$$

$$= \text{Greater Of } (0.27_{(Calculated)}, 1.50_{(Minimum\ Allowed)}) + 0.00_{(corrosion)} + 0.00_{(ext.\ corrosion)} = \mathbf{1.50}\text{ mm}$$

Minimum Thickness (plus corrosion) per Table UG-45

t_{b3} = minimum thickness (Table UG-45) + Ca + ext. Ca = **3.42** mm

Nozzle Minimum Thickness Based on Host and Table UG-45

$t_b = \min[t_{b3}, \max(t_{b1}, t_{b2})]$ = **2.73** mm

$t_{UG-45} = \max(t_a, t_b)$ = **2.73** mm

Wall thickness = tn * 0.875(pipe) = **3.42** is greater than or equal to UG-45 value of **2.73**

Fig. 7 APV Sheet for Nozzle (A1-Liquid Inlet) Calculations

8. SUPPORTS FOR VESSEL

Horizontally kept cylindrical pressure vessels are generally supported on twin saddle supports.

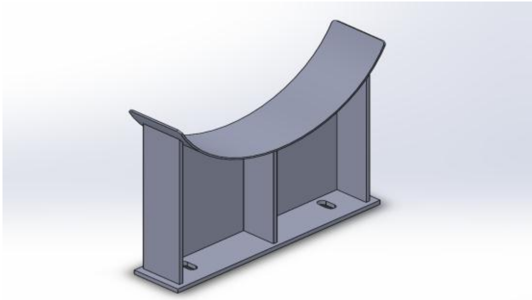

Fig. 8 Model of Saddle supports on Solidworks

ASME Flange Design Information

Host	Description	Type	Size (in.)	Material	ASME Class	Material Group	MAP (MPa)
Nozzle A1 Liquid Inlet	ASME Flange A1	Slip On	2	SA-182 Gr. F316	150	2.2	1.7520
Nozzle A2 Liquid Inlet	ASME FlangeA2	Slip On	2	SA-182 Gr. F316	150	2.2	1.7520
Nozzle B Makeup	ASME Flange	Slip On	2	SA-182 Gr. F316	150	2.2	1.7520
Nozzle C Spare with Bl	ASME Flange 4	Slip On	2	SA-182 Gr. F316	150	2.2	1.7520
ASME Flange 4	ASME Blind Flange C	Blind	2	SA-182 Gr. F316	150	2.2	1.7520
Nozzle D Liquid Inlet	ASME Flange D	Slip On	2	SA-182 Gr. F316	150	2.2	1.7520
Nozzle E Vent	ASME Flange E	Slip On	2	SA-182 Gr. F316	150	2.2	1.7520
Nozzle F	ASME Flange F	Slip On	3	SA-182 Gr. F316	150	2.2	1.7520
Nozzle G Level Guage	ASME Flange G	Slip On	1	SA-182 Gr. F316	150	2.2	1.7520
Nozzle H Level Guage	ASME Flange H	Slip On	1	SA-182 Gr. F316	150	2.2	1.7520
Nozzle I Spare with Bl	ASME Flange I	Slip On	2	SA-182 Gr. F316	150	2.2	1.7520
Nozzle J	ASME Flange J	Slip On	1	SA-182 Gr. F316	150	2.2	1.7520

Fig. 9 Flange Dimensions

Summary Information

	Dry Weight	Flooded Weight
Shell	185.65 kg	1263.61 kg
Head	39.81 kg	150.26 kg
Nozzle	7.51 kg	7.51 kg
ASME Flange	24.49 kg	24.49 kg
	0.00 kg	0.00 kg
	0.00 kg	0.00 kg
	0.00 kg	0.00 kg
	0.00 kg	0.00 kg
	0.00 kg	0.00 kg
Totals	257.46 kg	1445.86 kg

	Volume
Shell	1.08 cu. m
Head	0.11 cu. m
Nozzle	0.00 cu. m
	0.00 cu. m
	0.00 cu. m
Totals	1.19 cu. m

	Area
Shell	5.81 sq. m
Head	1.24 sq. m
Nozzle	0.33 sq. m
	0.00 sq. m
	0.00 sq. m
Totals	7.38 sq. m

Fig. 10 Summary information

9. FINAL ASSEMBLY OF V300

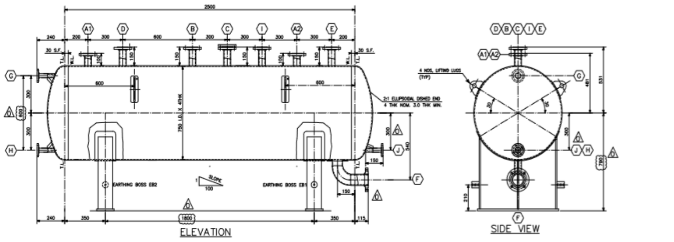

Fig. 11 Auto-cad drawing of V300

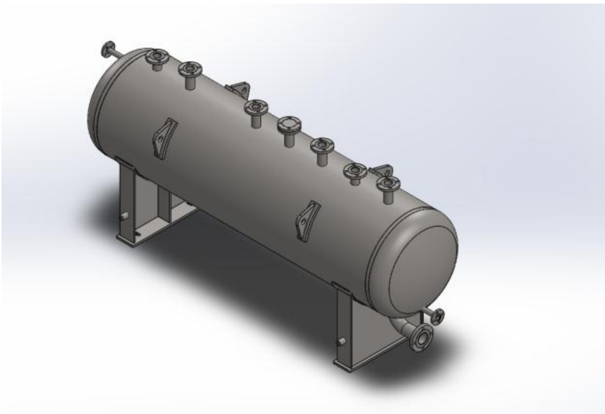

Fig. 12 Assembly of V300 on Solidworks

10. ANALYSIS ON ANSYS WORKBENCH

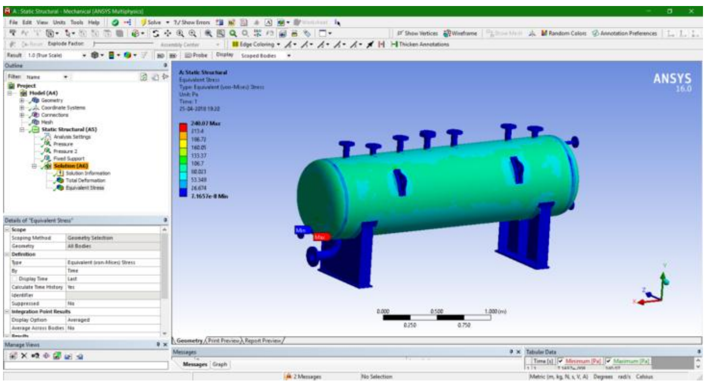

Fig. 13 Stress Analysis

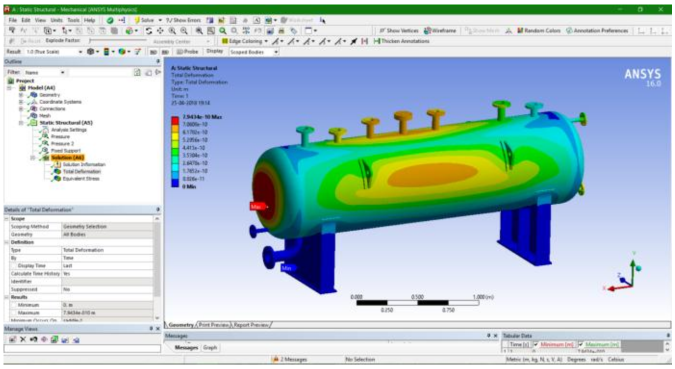

Fig. 14 Deflection Analysis

11. HYDROSTATIC TEST REPORT

Min. Test temperature= MDMT + 30°F
Max. Test temperature = 120°F
MDMT is the pressure vessel minimum design metal temperature which is 5^0C

PRESSURE TEST REPORT

CLIENT:	XYTEL INDIA PVT. LTD.	INSPECTION BY FABRICATION BY	XIPL ALPHATECH

EQUIPMENT:	Liquid Collection Tank	DRG NO:	J17-534-V-300	REV NO:	0

PO NO DATE:	[redacted]	JOB NO:	J17-534	QTY: 01	

TAG NO.	V-300	DATE OF INSPECTION	13.12.2017

DESCRIPTION	DESIGN PREURE Bar g	DESIGN TEMP. °C	TEST PRESSURE Bar g	HOLDING TIME MINUTES	TESTING MEDIA
SHELL SIDE	10	70	13	30	WATER

Pressure Gauge Details:

Certificate No :- 17.07.15.001
SR No. :- AF/PG/111
Calibration Due Date: - 14.07.2018

Calibration Date :- 15.07.2017
Range :- 0 to 18Bar g
Make :- MASS

REMARK: - No leakage found, no pressure drop observed test found satisfactory.

INSPECTION AUTHORITY
(XIPL)

VENDOR/ MANUFACTURER
(ALPHATECH ENGINEERING)

Fig. 15 Hydrostatic Test Report

12. RESULTS AND DISCUSSION

The calculated values of thickness of every component are compared with the values obtained using APV software. Following table shows the comparative study of calculated values of thickness and values obtained from software.

Table 1 Results

Sr. No	Parameter	Value Obtained from software (mm)
1.	Shell thickness	4
2.	Head thickness	4
3.	Nozzle(1”) thickness	3.42
4.	Nozzle(2”) thickness	3.91
5.	Nozzle(3”) thickness	5.49

1. Design consistent with American Pressure Vessel (APV) software design.
2. Static structural analysis done in ANSYS Workbench 16.0. Design is safe.
3. Hydrostatic pressure test carried out on V300 at a pressure of 13 bar. No leakages or failure of components.

13. CONCLUSIONS

Various materials available for pressure vessel are studied and suitable material for each component is selected. Thickness of shell, head and nozzle were calculated manually and validated using APV software.
Cad design is extremely important since it will be provided to manufacturer and will be used for three dimensional drawings. If there is a minor mistake in the drawing then it may lead to the manufacturing defects, difficulties in assembly and failure of the vessel.
Hydrostatic test is performed to check the leakages and pressure bearing capacity of the vessel, thus assured the safety.
Pressure vessel accidents are one of the most severe and explosive accidents, leading to the loss of working personnel around it. Hence, the perfect design with all the tests must be carried out.

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REFERENCES

[1] Joshi’s Process equipment design, M.V. Joshi, V.V.Mahajani, S.B. Umarji, Trinity Press, New Delhi, 1979, (Reprint-2014), pp.91-149

[2] ASME code section VIII, Div. 1 (Ed. 2015)

[3] ASME code section II, Part D, Properties

[4] Quantifying the role of stiffening rings in pressure vessels using FEA by Pushpa Khot, Dr. S.G. Taji, International Journal of Science, Engineering and Technology Research (IJSETR), Volume 5, Issue 1, January 2016.