International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 07 Issue: 04 | Apr 2020 www.irjet.net p-ISSN: 2395-0072

"DESIGN, DEVELOPMENT AND STRUCTURAL ANALYSIS OF EMBEDDED HEAT PIPE"

[1]Priyanka T. Randive, [2]Abhay M. Kalje

[1] Research scholar ME - Mech (Design Engg.), [2]Associate Professor and Dean Academics (ME- Production)
N. B. Navale Sinhgad College of Engineering, Solapur, Solapur University, India.

ABSTRACT

Pressure vessels are the important storage equipment in petroleum industry. Crude oil is the raw form of oil and which has to be stored at specific pressure and temperature before processing into diesel, petrol and more. Failure in maintaining pressure and temperature of the crude oil leads to solidification of oil and would be difficult to process further. So the concept is to introduce heat pipe inside the pressure vessel and which will be embedded in the pressure vessel with continuous supply of heat passing through the heat pipe in order to maintain the pressure and temperature as desired. The project contains detail design and structural analysis of heat pipe as well as pressure vessel. The type of heat pipe differs as U- shaped and C- shaped. The dissertation further contains structural analysis of both U and C shaped heat pipe with Finite element analysis and ANSIS workbench.

Keywords: Flat head Pressure vessel, embedded heat pipes, FEA

1. INTRODUCTION

1.1 Overview of Pressure Vessel

Pressure Vessel is single most important aspect of mechanical engineering in the industrial field. Pressure vessel is defined as a container with a pressure differential between inside and 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. Cylindrical pressure vessels are divided into two groups, thin and thick cylinders. A cylinder is considered thin when the ratio of its inner diameter to the wall thickness is more than 20. When the ratio of the inner diameter of the cylinder to the wall thickness is less than 20, the cylinder is called a 'thick-walled cylinder'.

2. DESIGN OF PRESSURE VESSEL

2.1 Design Data

Design a pressure vessel for the following specifications

Table 2.1 Design specification for pressure vessel

Sr No.	Parameter Description	Parameter Code	Value
1	Internal Pressure	P	0.05 MPa
2	External Pressure	P_o	Atm

3	Process Volume	V_p	205 m^3
4	Expected Stagnant Volume	V_S	57 m^3
5	Buffer Volume Requirement	V_b	50 m^3
6	Vessel radius	R	2.5 m
7	Tube porosity volume	T_p	25
8	Radius of tube sheet	r	2.5 m
9	Tube diameter	T_d	200 mm
10	Skirt Support height	h	3 m

2.3 Referring Code A2209, for full Process Reactionary Vessel

2.3.1 Calculation for nozzle to nozzle distance (NTD)

For P(0.05 MPa)<1.4 MPa and Process reactionary vessel.

$$V_p = (0.90 \times \mathrm{NTD}) \times (\pi R^2) \,..\, (2)$$

0.90 is the Reactionary stabilization Parameter, as 10% of flow volume may eventually stagnate.

$$\mathrm{NTD} = \frac{V_p}{(0.90 \times \pi R^2)} = \frac{205}{(0.90 \times \pi 2.5^2)}$$

$$\mathrm{NTD} = 11.6\ \mathrm{m}$$

2.3.2 Calculation for L_1, (Total Distance from tube sheet to flat head)

$$V_s + V_b = (0.82 \times L_1) \times (\pi R^2) \,..\, (3)$$

0.82 is the cross buffer stabilization parameter

$57 + 50 = (0.82 \times L_1) \times (\pi 2.5^2)$ 57+50=0.82×L_1×π2.5^2

$$L_1 = 6.645\ \mathrm{m}$$

2.4 Recalculation of Volume Considering Tube sheet Thickness

$$V_p = 1.1(V_p' + V_r) + 1.2(\pi \times T_d \times T_d) \times (T_p/400) N \,..\, (4)$$

Where,

V_p' = Process volume = 205 m^3 V_r = Residual volume = 0.9754 m^3

T_d = Tube diameter = 0.2 m T_p = Tube porosity volume = 25

N = Total number of tubes = 4

$$V_p = 1.1(205 + 0.9754) + 1.2(\pi \times 0.2 \times 0.2)(25/400)4$$

$$V_p = 226.61\ m^3$$

For Process reactionary vessel. using equation (2)

$$V_p = (0.90 \times \mathrm{NTD}) \times (\pi R^2)$$

$$\mathrm{NTD} = \frac{V_p}{(0.90 \times \pi R^2)} = \frac{226.61}{(0.90 \times \pi \times 2.5^2)}$$

$$\mathrm{NTD} = 12.823\ \mathrm{m}$$

2.4.1 Internal Pressure

$$L_0 = \text{L\_0=(1.1) (NTD)} = 1.1 \times 12.823 \ldots (5)$$

$$L_0 = 14.1053 \text{ m}$$

Total Height $h_t = L_0 + 2\,L_1$

$$h_t = 14.1053 + (2\times 6.645)$$

$$h_t = 27.3953 \text{ m}$$

$P_i = 0.05 + \frac{\delta g h_t}{10^6}$ ..(6) P_i=0.05+δgh_t/10^6 ..(6)

$$P_i = 0.05 + \frac{(1000\times 1.25)(9.81)(27.3953)}{10^6}$$

$$= 0.05 + 0.31436$$

$$= 0.3859 \text{ N/mm}^2 < 1.4 \text{ MPa}$$

2.5 Material Properties

SA 516 Grade70 Maximum allowable stress (S) = 20000 psi = 138 MPa

Modulus of elasticity E = 200 GPa Poisson's ratio $\mu = 0.29$

2.6 Shell Thickness

S= 138 MPa E_l=0.7 longitudinal seam efficiency (circ stress)

E_e=0.85 circ seam efficiency (long. stress) P_i = 0.3859 MPa

R = 2500 mm

From ASME Section VIII, div –I, UG27 [3]

$$t_a = \frac{P_i \times R}{S\times E_l - 0.6\times p_i} \quad \ldots(7)$$

$$t_a = \frac{0.3859 \times 2500}{138\times 0.7 - 0.6\times 0.3859} = 10.01 \text{ mm}$$

$$t_b = \frac{P\times R}{2SE_e + 0.4P} \quad \ldots(8)$$

$$t_b = \frac{0.3859\times 2500}{2\times 138\times 0.85 + 0.4\times 0.3859} = 4.109 \text{ mm}$$

$$T_{req} = \mathrm{Max}(t_a, t_b) + CA \ldots (9)$$

$$T_{req} = 10.01 + 6$$

$$T_{req} = 16.01 \approx n_t = 18 \text{ mm}$$

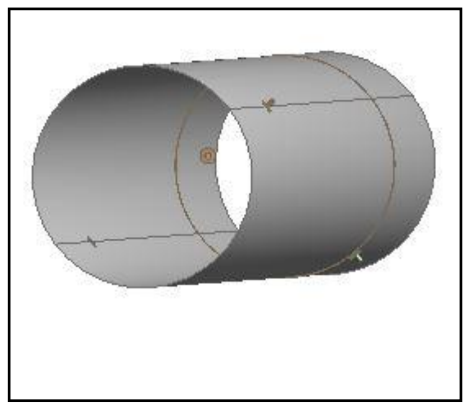

Fig. 2.1 Shell render

2.6.1 Maximum Pressure

$$P_{i_1} = \frac{SE_l \times n_t}{R_i + 0.6n_t} = \frac{138\times 0.7\times 18}{2500 + 0.6\times 18} \,..(10)$$

$$P_{i_1} = 0.6875$$

$$P_{i_2} = \frac{2SE_c \times n_t}{R_i - 0.4n_t} \,..(11)$$

$$= \frac{2\times 138\times 0.85\times 18}{2500 - 0.4\times 18}$$

$$P_{i_2} = 1.693 \text{ MPa} > 0.05 \text{ MPa}$$

Maximum allowed design pressure

$$P_{max} = \text{Min}\left(P_{i_1}, P_{i_2}\right) \,..(12)$$

$$= \text{Min}(0.6875, 1.693)$$

$$= 0.6875 \text{ MPa} > 0.3859 \text{ MPa (Acceptable)}$$

2.7 Thickness of Head

Flat head

$$t_h = 0.7 d_i \sqrt{\frac{P_i}{SE}} + CA \,..(13)$$

$$t_h = 0.7(5000)\sqrt{\frac{0.3859}{138.57}} + 6$$

$$t_h = 200 \text{ mm}$$

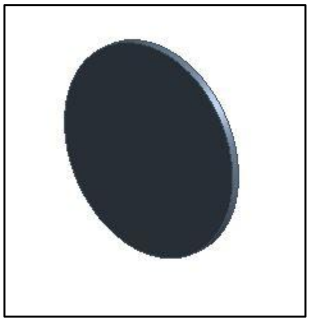

Fig. 2.2 flat head render

2.8 Calculation of pressure

$$h_t = L_0 + 2L_1 = 13.212 + 2\times 6.645 = 26.413 \text{ m}$$

$$P = 0.05 + \frac{\delta g h}{10^6} \,..(26)$$

$$P = 0.05 + \frac{(1250 \times 9.81 \times 26.5)}{10^6}$$

P = 0.3759 MPa

2.8.1 Shell thickness

$$t_a = \frac{P \times R}{S \times E - 0.6 \times P} + CA \,..(27)$$

$$t_a = \frac{0.2774 \times 2500}{138 \times 0.7 - 0.6 \times 0.2774} + 6$$

= 7.19 + 6

=13.191 < 15 mm

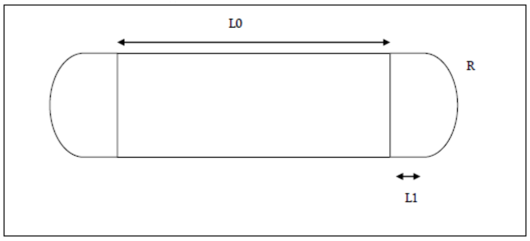

Fig. 2.3 Pressure vessel layout 1

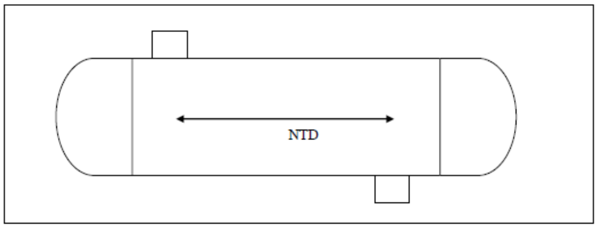

Fig. 2.4 Pressure vessel layout 2

Main stock Cylindrical length L_0 = 11.9 m
Buffer Stock Cylindrical Length L_1 = 6.645 m
Vessel Radius R = 2.5

Main Nozzle to Nozzle Centre Distance (Inlet/Outlet) NTD = 11.5 m

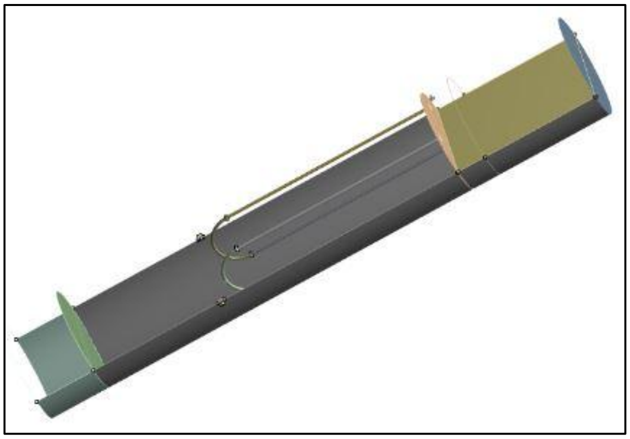

Fig. 2.5 Sectional view of vertical pressure vessel

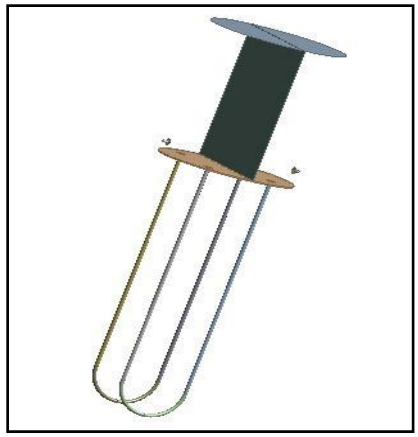

Fig. 2.6 Embedded pipes with partition plate

3. ANALYSIS

3.1 Assumptions

1. The small size nozzle, inlet-outlet pipe and other mounting and accessories are not consider for the purpose of Finite element analysis.(Because the wieght of these devices are very small compared with weight of the entire column)
2. The vertical storage column is considered as thin pressure vessel because of (diameter to thickness ratio is greater than 20)
3. The cad model is meshed with second order shell element (SHELL 93)
4. The each of vertical support are fix into individual concrete column.
5. For static analysis the welding is not require to simulate because the weld material and parent material are assumed to be same(and the stress is independent from the material).therefor entire structure is consider as a continuous structure.

3.2 Structure analysis

1. The structural analysis of the column is carried out in order to check whether the column structure is safe or not under the given boundry condition.
2. Due to the presence of contact at the junction of reinforcement pad to the cyllinderical shell of column the analysis become non linear static structural analysis.
3. Due to large structure of column the weight of column is taken into account by applying gravity to the structure.
4. Before performing the further analysis the two model of embedded Pipes are taken into consideration. And making the entire structure safe.

The two models which are taken into comparison are C-Shape and U-Shape embedded Pipes.

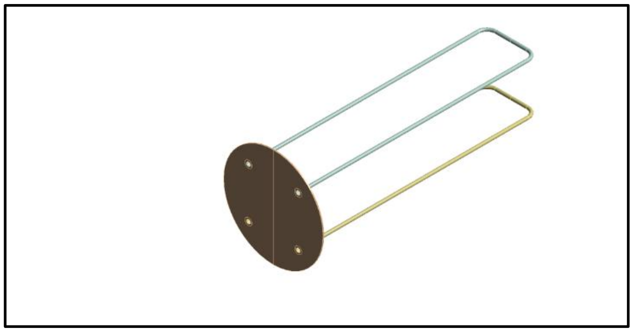

Fig. 3.1 C- shape embedded Pipe

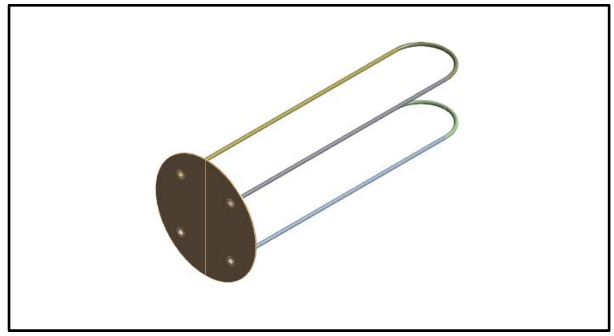

Fig. 3.2 U- shape embedded Pipe

4. STRUCTURAL ANALYSIS

4.1 Structure Analysis for C- shape and U shape embedded pipe

4.1.1 Case 1: Structure Analysis for self-weight considering the earth gravity.

The structure is simulated for the Self weight to check whether the structure is safe for no load

Boundary Condition

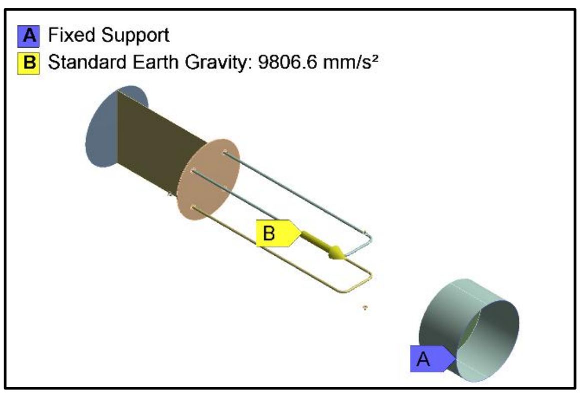

Fig. 4.1 Boundary Condition for self-weight

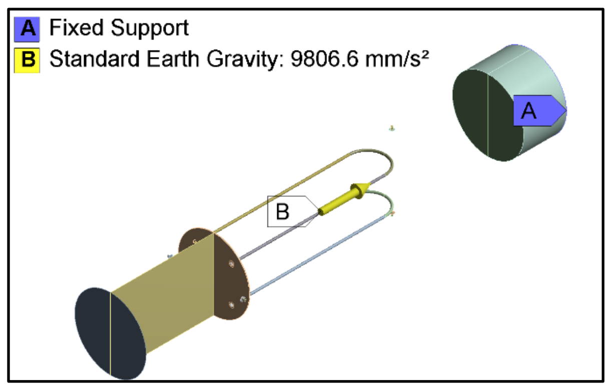

Fig. 4.2 Boundary condition for Self-weight

In figure 6.3 vessel is fixed at skirt support and standard earth gravity of 9806.6mm/s^2 is applied keeping the vessel empty

Total Deformation

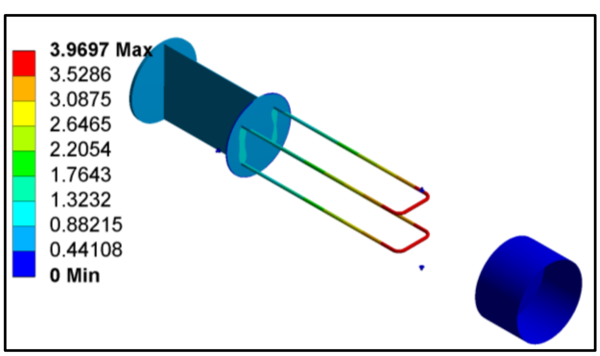

Fig. 4.3 Total deformation

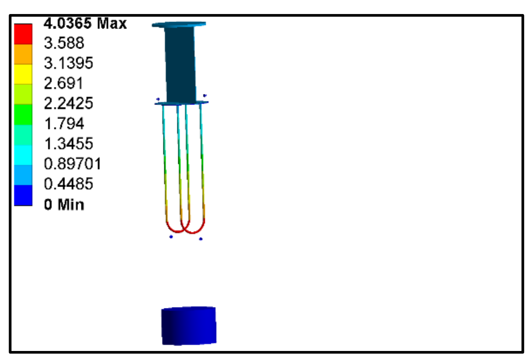

Fig. 4.4 Total deformation

The deformation developed at the C- section of the embedded pipe is 3.9697 mm and with U- section of the embedded pipe is 4.0365 mm.

(Von – mises) Stress

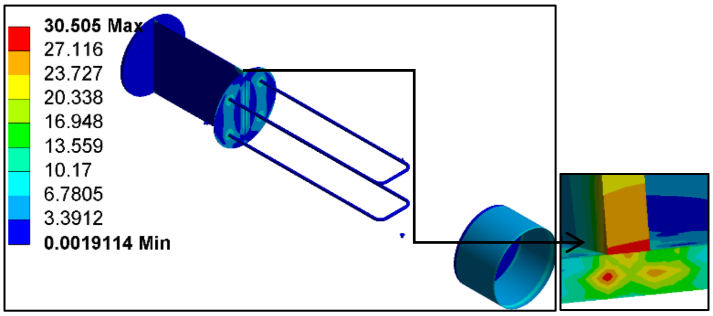

Fig. 4.5 (Von – mises) Stress

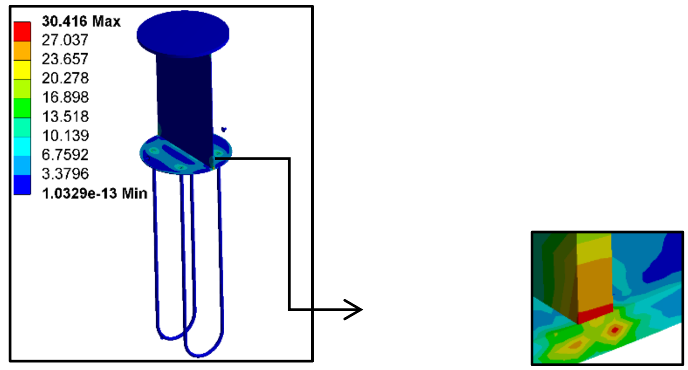

Fig 4.6 (Von – mises) Stress

The maximum Stress developed at the edge of partition plate and tube sheet of 30.505 MPa.

4.2.1 Case 2: Structure Analysis for Upper chamber. Pressure of 0.04 MPa and 0.05 MPa is applied at both sections to maintain the pressure difference in pipe.

Boundary Condition

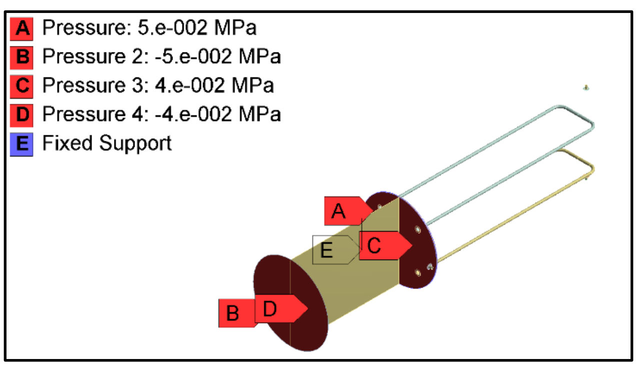

Fig. 4.7 Boundary Condition for Upper chamber

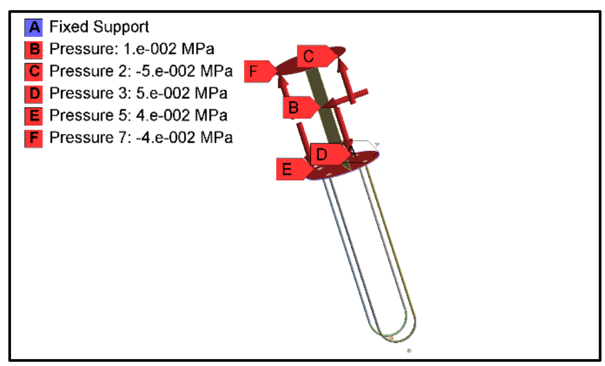

Fig. 4.8 Boundary Condition for Upper chamber

Total Deformation

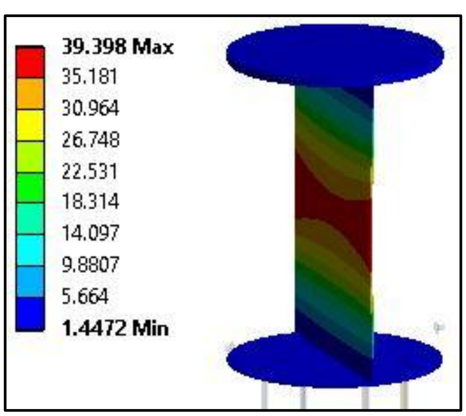

Fig. 4.9 Total deformation

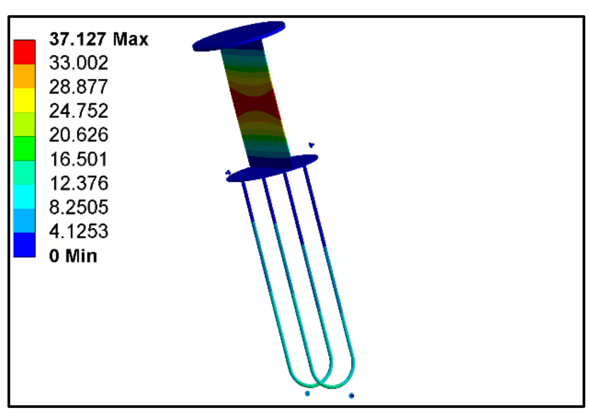

Fig. 4.10 Total deformation

The maximum deformation at upper chamber is 39.398MPa.

(Von – mises) Stress

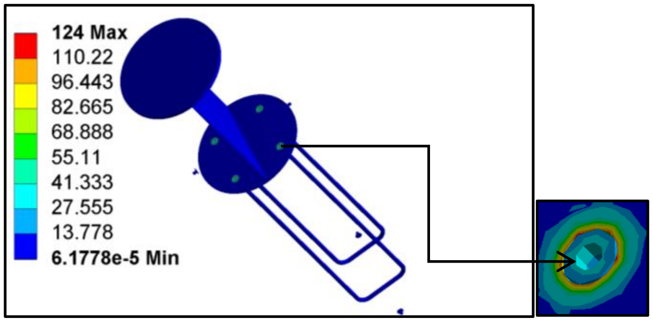

Fig. 4.11 (Von – mises) Stress

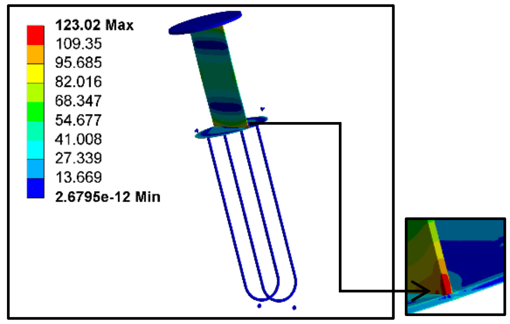

Fig. 4.12 (Von – mises) Stress

Maximum stress developed at the tubesheet is 124 MPa.

For Static loading the allowable deformation is given by L/300

Where L is the total height of the vessel = 26000mm

= 26000/300 = 86.66 mm

Total deformation is less than the allowable deformation. Allowable stress given is 138 MPa according to ASME codes.

Table 4.1 Result Table

Sr. no.	Analysis	C- Shape Embedded Pipe		U-Shape Embedded Pipe	
		Deformation	Stress	Deformation	Stress
1	Self-weight	3.9697 mm	30.505 MPa	4.0365 mm	30.416 MPa
2	Differential pressure	37.432 mm	124.89 MPa	37.432 mm	124.87 MPa
3	Upper chamber	39.398 mm	124 MPa	37.127 mm	123 MPa
4	60^{o}C Temperature to Pipe, 7333.7 N Buoyancy	5.4816 mm	124 MPa	5.4914 mm	95.359 MPa
5	60^{o}C,40 ^{o}C to Pipe 7333.3N Buoyancy	6.2065 mm	1012.8 MPa	6.1793 mm	487.39 MPa

- From above results the Deformation and Stress developed in U- shape embedded pipe is less as compared to C-shape embedded pipe

5. VALIDATION WITH EXPERIMENTAL WORK

The experiments were carried out in pressure vessel testing facility. The scaled model was developed and tested under different load conditions in pressure vessel testing facility. Results were obtained for the deformation.

5.1 C-Shape Embedded Pipe

Table 5.1 Experimental versus FEA results for deformation

Sr. no.	Analysis	Experimental	Finite Element Method	Percentage error
1	Self-weight	3.525mm	3.9697mm	11.2%
2	Differential pressure	33.323mm	37.432mm	10%
3	Upper chamber	35.398mm	39.398mm	10.25%
4	60°C Temperature to Pipe, Buoyancy	4.816mm	5.4816mm	6%
5	60°C,40 °C to Pipe	5.402mm	6.2065mm	12%

From the above table it can be seen that there is not much difference in the experimental results and results obtained by finite element method. The maximum percentage error is 11.2%. Results obtained by both the methods are almost same.

5.2 U-Shape Embedded Pipe

Table 5.2 Experimental versus FEA results for deformation

Sr. no.	Analysis	Experimental	Finite Element Method	Percentage error
1	Self-weight	3.414mm	4.0365mm	15.42%
2	Differential pressure	30.412mm	34.701mm	12.36%
3	Upper chamber	32.323mm	37.127mm	13.31%
4	60°C Temperature to Pipe, Buoyancy	4.716mm	5.4914mm	14.114%
5	60°C,40 °C to Pipe	5.321mm	6.1793mm	13.12%

From the above table it can be seen that there is not much difference in the experimental results and results obtained by finite element method. The maximum percentage error is 15.42%. Results obtained by both the methods are almost same.

6. CONCLUSION

6.1 Summary

The aim of this research work was to design and optimized the process reactionary vessel and the heat treatment pipe. The structural and thermal analysis of embedded pipes inside pressure vessel is carried out using numerical simulation. Two models are simulated at different load conditions. On the basis of the simulation results following conclusions are drawn out:

i. The U- shape embedded heat pipe is structurally stable than the C-shape embedded heat pipe. Also tube thickness of 50 mm the minimum stress is developed in the structure.
ii. After finalizing the model thermal analysis of u-shape embedded heat pipe were simulated and it is found that the maximum temperature of embedded pipe is 62.97 ^{o}C after 1900 seconds at the outlet of the pipe.
iii. The temperature of naphtha heavy oil is between the ranges of 30-35 ^{o}C after 1900 seconds. So it can be concluded that these temperatures are within the acceptable limits.

7. FUTURE WORK

i. Selecting different Material can increase the heat transfer rate in the vessel. Also the different shape of embedded pipe can be tested for proper distribution of heat flow.
ii. Fluid to Fluid interaction can also be tested for the phase change and the temperature distribution considering the same structure of embedded heat pipe.
iii. Weight minimization of structure can be done which will reduce the total manufacturing cost without compromising the heat transfer of the heat pipe.

REFERENCES

[1] **Alegre**J. M., **Ciesta** I.I.,Stress Intensity Factor Equations for Internal Semi-Elliptical Cracks in Pressurized Cylinders.*Journal of Pressure Vessel Technology*. **133**.054501-1, 2011.

[2] **Azzam** B.S., **Muhammad** M.A., **Kolkailah** F.A.; Creep relaxation in filament wound composite pressure vessel. *Computational Mechanics*, *Springer*, **32**, 2244-2249, 1995.

[3] **Bhatt** M.J., **Gohil** A., **Shah** H., **Patel** N.; Design Calculation of nozzle junction based on ASME Pressure Vessel Design Code. *International Journal of Advance Engineering and Research Development*. **1(5)**, 2348-4470, 2014.

[4] **Boo** J.H., **Lee** S.K., **Kim** J.K.; Numerical Analysis of a Thermal Storage System with Inserted Heat Pipes for Medium-High Temperature Range, *International Journal of heat pipes,***(46)**, 1582-1588, 2011.

[5] **Chavan** C., **Barve** S.; Parametric study for wind design of vertical pressure as per Indian standard. *International Journal of Modern Engineering Research*. **3(4)**, 2157-2160, 2013.

[6] **Chaaba** A; Plastic Collapse Assessment of Thick Vessels under Internal Pressure According to Various Hardening Rules. *Journal of Pressure Vessel Technology*. **132**, 051207-1, 2010.

[7] **Carbajal W.**.M.**, Panagiotis ,Michaleris, Mark T. Kirk**; An Improved Treatment of Residual Stresses in flaw Assessment of pipes and pressure vessels fabricated from ferritic steels.*International journal of engineering science and technology*. **(6)**,135-139,2004.

[8] **Jadhav** A., **Gupta** S.R., **Desai** A.; Optimize Nozzle Location For Minimization Of Stress in Pressure Vessel. *International Journal of Innovation Research. In Science, & Technology*. **3(1)**, 2349-6010,2010.

[9] **Khoramishad** H., **Ayatollahi** R.; Finite Element Analysis of a Semi-Elliptical External Crack in a Buried Pipe. *Transcation of the Canadian society forMechanical Engineering.*(**33**). No.3.2009.

[10] **Kumar** S**., Vaidyanathan** S**., Sivaraman** B.; Thermal analysis of heat pipe using Taguchi method. *International journal of engineering science and technology*. **2(4)**, 564-569. 2010.

[11]**Khazaeinejad P.**; On the Buckling of Functionally Graded Cylindrical Shells Under Combined External Pressure and Axial Compression.*Journal of Pressure Vessel Technology*. **132**,064501-1, 2010.

[12] **Liu** M.A., **Peng** F.U., **Jin-yang** Z., **Cun** J. M., **Lin-lin** W.U.; Calculation of Plastic collapse load of pressure vessel using FEA. *Journal of Zhejiang University*. **9(7)**, 1673-565X, 2009.

[13] **Mohamed** H., **Elnaggar** A.; The Effect of thickness and permeability of wick structure on L-shape heat pipe performance using different working fluid, *Frontiers in heat pipes*, **3**, 2155-658X,2012.

[14] **Pandey** S.C., **Ralli** D.K., A K **Saxena**, W K **Alamkhan**.; Physicochemical characterization and application of naphtha. *Journal of scientific and Industrial Research*, **5(5)**, 346-351, 2004.
[15] **Staat** M., **Heitzer** M., **Lang** H., **Wirtz** K., Direct Finite element route for design-by-analysis of pressure components, *International journal of Pressure Vessel and Piping* .**82**, 61-67.2005.
[16] **Tian** J., **Lu** T.J., **Hodson** H.P.; Thermal-Hydraulic performance of sandwich structures with crossed tube truss core and embedded heat pipes, *13th International Heat pipe Conference*, 2004.
[17] **William** B.J., **Zimmerman**; Process Modeling and Simulation with Finite Element Methods. World Scientific Publishing Co., 2004.
[18]**Boresi** A.P., **Schmidt** R.J.; Advanced Mechanics of Materials. Sixth edition. John Wiley & Sons, Inc.2003.
[19]**Ball** B.E., **Carter** W.J.; CASTI Guidebook to ASME section VIII Div. 1- Pressure Vessels. Third edition, CASTI Publishing Inc., Canada, 2002.
[20] Ansys 14.5. Ansys manual.