Design Testing and Simulation of Heat treatment pipe inside Process Reactionally Vessel.

[#1]Admane Mahesh C., [#2]Dr. S. G. Dambhare

[1]Heat Power Engineering, S. P. Pune University, Pune. Maharashtra, India.
[2]Mechanical Engineering, S. P. Pune University, Pune. Maharashtra, India.

Abstract: Chemical Process is widely carried out by using heat. Naphtha is a volatile commercial product which is obtained by the distillation process in petroleum refinery of coal tar. Petroleum naphtha is a name for the petroleum distillate which contains aliphatic hydrocarbons and boiling point higher than the gasoline and lower than kerosene. In the crude oil distillation the overhead liquid distillate is called virgin or straight naphtha. The virgin naphtha has the initial boiling point of about 35 ºC and final boiling point of about 200ºC.A vertical storage column is typically used to store these liquids. To maximize the storage capacity, these columns are usually very tall typically over 27meters.Heat is used to maintain viscosity of naphtha that is stored. As naphtha has a non-linear thermal property after distillation it is in the semi-fluid form. If the temperature drops the fluid changes from semi- liquid to semi-solid form and can cause considerable damage to the system and becomes difficult for further processing. However if we heat from below, at time surface cooling occurs and this makes natural convection stop, and the heat never reaches the top layers. Hence to avoid this heat pipes are inserted from top to ensure efficient distribution of heat. In this project we will study the effect of thermal stress due the presence of embedded heat pipes and try to simulate the distribution of heat in the system using ANSYS.Furthermore, we will take a problem statement from the company on any one pressure vessel job that is currently going on in the company and design the entire vessel. This would help us in realizing the challenges that need to be overcome when taking a real time project at hand.

Keywords: **Naphtha, Process Reactionally Vessel, Heat Pipe, Performance, Design Testing and Simulation.**

I. INTRODUCTION

Naphtha is a liquid state petroleum product that boils from about 30°C (86°F) to approximately boils at 200°C (392°F), although there are different types or grades of naphtha within this extensive boiling range that have different boiling ranges (Guthrie, 1968; Good fellow, 1973; Weisermel and Aarpe, 1979; Francis and Peters, 1980; Hoff`man, 1983; Austin`s, 1984; Chenie R, 1992; Speight, 1999; Hori, 2000). The term petroleum solvent is often used synonymously with naphtha.

The so-called petroleum ether solvents are specific-boiling-range naphtha ligroin. The special value of naphtha is as a solvent lies in its stability and purity.

A process reactionally vessel is a container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. The pressure differential is dangerous, and fatal accidents have occurred in the history of process reactionally vessel development and operation. Consequently, process reactionally vessel design, manufacture, and operation are regulated by engineering authorities backed by legislation. For these reasons, the definition of a process reactionally vessel varies from country to country, but involves parameters such as maximum safe operating pressure and temperature, and are engineered with a safety factor, minimum design temperature (for brittle fracture), corrosion allowance and involve nondestructive testing, such as ultrasonic`s testing, radiographies and pressure tests, usually involving water, also known as a hydro-test, but could be pneumatically tested involving air or another gas.

II. LITERATURE REVIEW

Senthilkumar [1] has studied the finite element analysis of pressure vessels for determining the stresses in local area`s such as penetration. They found that O-ring grooves and other areas are difficult to analyze by hand. After testings the three dimensional, symmetric and axisymmetric model finite element analysis becomes a extremely powerful tool. For symmetric models shell elements and axisymmetric models using solid elements can be used. Local stress risers, unrealistic displacement becomes extremely important in this kind of analysis.

M Ram Ranjan. [2] developed pipe cooling system using three dimensional finite element programs for thermal analysis. For the modeling of pipe they adopted the line element method also the internal flow theory was applied for calculating the temperature variation of cooling water. For the model of concrete eight node isoperimetric solid element was introduced and two node iso-parametric line elements was taken to this solid element to implement pipe cooling effect into the analysis. Results shows that the temperature variation of cooling water is efficiently obtained by influence of internal flow theory and line element.

Jin Keum kim [3] studied the different factors which can affect the performance of the heat pipe while selecting the material. The properties which were considered while selecting the material are corrosion resistance, stress corrosion cracking potential, thermal and mechanical properties and temperature limitations, finally they concluded that ferrite and stainless steel alloys containing 12% chromium can be most cost effective. **Michael A. Porter Dennis H. Martens Pedro Marcal. [4]** presented the practical approach in the analysis of typical pressure vessel components. 3-D brick element was taken as the element type for solution as it takes less time and effort. Also suggested that finer mesh will produce more accurate results than a coarser mesh. Another important consideration in meshing is the aspect ratio of the elements. (1:1:1) for brick elements which means that square or cubic elements are best.

William et al. [5] has studied the improved treatment of residual stresses which are produced by welding in pipes and pressure vessels fabricated form ferritic steels. The residual stresses in piping and pressure vessels depend on the pipe wall thickness and the welding heat input. The estimate for axial residual stresses at the inside surface can be related with a linearized bending residual stress that has been shown to accurately predict the effect of welding residual stress on crack connected to the internal surface. **Khandekar et al. [6]** studied that the closed loop pulsating heat pipes are complex heat transfer systems with a very strong thermo-hydrodynamic coupling governing the thermal performance. Semi-empirical correlation has been developed to fit the available data based on non-dimensional numbers of interest.

Tian et al. [7] has experimentally characterized the performance of a cellular metal sandwich heat sink consisting of a crossed tube core with embedded heat pipes under forced air convection. The experimental results shows that on pressure loss and heat transfer rate is obtained as function of Reynolds number for selected values of input heat fluxes, and compared with those obtained for sandwich structures with woven textile cores. Also the heat pipe sample can dissipate more heat than the stainless steel wire mesh sample.

Hadi Khoramishad.[8] has given two approaches for carrying out a design-by-analysis which covers both the stress categorization method and the direct route method for checking global plastic deformation and against progressive plastic deformation. The pipe is discretized by an axisymmetric ring element with quadrilateral cross section with 4 nodes as it gives better results.

III. PROBLEM DESCRIPTION

1] As temperature drops or down the fluid changes from semi-liquid to semi-solid form. This will Cause considerable damage to the system and becomes difficult for further processing. Uneven thermal Stresses developed in the heat pipe. This could cause a serious damage to the structure.

2] Naptha is the petroleum product which is used as fuel or use as their accordance. But due to temperature variation, there is chances of pipe chock up because of the viscosity. Below the 35 degree Celsius, Naphtha get solidified state and due to this pipe will bursted or chock up.

3] To maintain Naptha in liquid state, that means to maintain it at 35 to 40 degree Celsius. In old method, it maintain by Natural convection so it is difficult to distribute the heat in proper way.

IV. CASE STUDY

4.1 THE THICK WALL CYLINDER

Thick-wall cylinders are used widely in industry as process reactionally vessels, pipes, gun tubes, etc. in many applications the cylinder wall thickness is constant and the cylinder is subjected to a uniform internal pressure p1, a uniform external pressure p2, an axial load p, and a temperature change ΔT. Often the temperature change ΔT is a function of the radial coordinate r only. Under such conditions, the deformations of the cylinder are symmetrical with respect to the axis of the cylinder.[1]

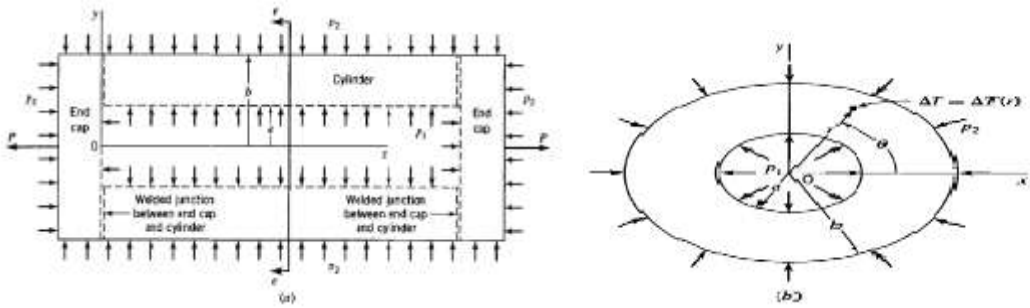


Fig 4.1 Closed cylinder with internal pressure, external pressure, and axial loads.

Furthermore, the deformations at a cross section sufficiently far removed from the junction of the cylinder and its end caps are practically independent of the axial coordinate z. In particular, if the cylinder is open and unconstrained, it undergoes ax symmetric deformations owing to pressure p1 and p2 and temperature changes ΔT= ΔT(r), which are independent of z. If the cylinder's deformation is constrained by support or end caps, then in the vicinity of the supports or junction between the cylinder and end caps, the deformation and stresses will depend on the axial coordinate z.

4.2 STRESS COMPONENTS AND RADIAL DISPLACEMENT FOR CONSTANT TEMPERATURE

4.2.1 STRESS COMPONENTS

As there is no temperature change, ΔT=0

The expressions for the stress components in a closed cylinder (cylinder with end caps) [2]

$$\sigma_{rr} = \frac{(p_1 a^2 - p_2 b^2)}{(b^2 - a^2)} - \frac{a^2 b^2}{r^2 (b^2 - a^2)}(p_1 - p_2) \qquad ..(1)$$

4.3 RADIAL DISPLACEMENT FOR A CLOSED CYLINDER

For no temperature change, ΔT=0. Then the radial displacement u for a point in a thick-wall closed cylinder (cylinder with end caps).

The resulting expression for u is

$$u_{(\text{closed end})} = \frac{r}{E(b^2 - a^2)}\left[(1 - 2v)(p_1 a^2 - p_2 b^2) + \frac{(1 + v)a^2 b^2}{r^2}(p_1 - p_2) - v\frac{p}{\pi}\right] \qquad ..(2)$$

4.4 STRESSES AND DEFORMATIONS IN A HOLLOW CYLINDER

A Thick-wall closed-end cylinder is made of an aluminum alloy (E=72GPa and v=0.33), has an inside diameter of 200mm, and has an outside diameter of 800mm. the cylinder is subjected to an internal pressure of 150 MPa. Determine the principle stresses, maximum shear stress at the inner radius(r=a=100mm), and the increase in the inside diameter caused by the internal pressure.[2]

4.4.1 ANALYTICAL SOLUTION

The principal stresses are given by Equations (1) & (2).

$$\sigma_{rr} = \frac{(p_1 a^2 - p_2 b^2)}{(b^2 - a^2)} - \frac{a^2 b^2}{r^2 (b^2 - a^2)}(p_1 - p_2)$$

For the conditions that p_2=0 and r=a, these equations give

$$\sigma_{rr} = p_1 \frac{(a^2 - b^2)}{(b^2 - a^2)} = -p_1 = -150\text{MPa}$$

The maximum shear stress, given by

$$\tau_{max} = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{170 - (-150)}{2} = 160\text{MPa}$$

The increase in the inside diameter caused by the internal pressure is equal to twice the radial displacement given by

$$u_{(\text{closed end})} = \frac{r}{E(b^2 - a^2)}\left[(1 - 2v)(p_1 a^2 - p_2 b^2) + \frac{(1 + v)a^2 b^2}{r^2}(p_1 - p_2) - v\frac{p}{\pi}\right]$$

For the conditions p_2 = p = 0 and r = a. Thus,

$$u_{(r=a)} = \frac{p_1 a}{E(b^2 - a^2)}[(1 - 2v)a^2 + (1 + v)b^2]$$

=0.3003mm

The increase in the inside diameter caused by the internal pressure is 0.6006mm.

4.5 NUMERICAL SOLUTION MODEL

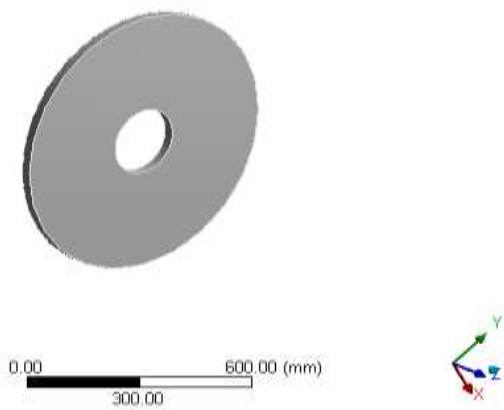


Figure 4.2 Model

Model was created having inside diameter of 200mm, and outside diameter of 800mm it is extruded to 30 mm keeping symmetry in both the direction. Then defined material properties are assigned to this geometry.

4.5.1MATERIAL AND ITS PROPERTIES

Material Selected: Aluminum alloy

4.5.2 MESHING

METHOD: Hex Dominant method
ELEMENT MIDSIDE NODES: Kept
FREE FACE MESH TYPE: Quad/Tri
NUMBER OF NODES: 29195
NUMBER OF ELEMENTS: 5696
ELEMENT TYPE: SOLID 186

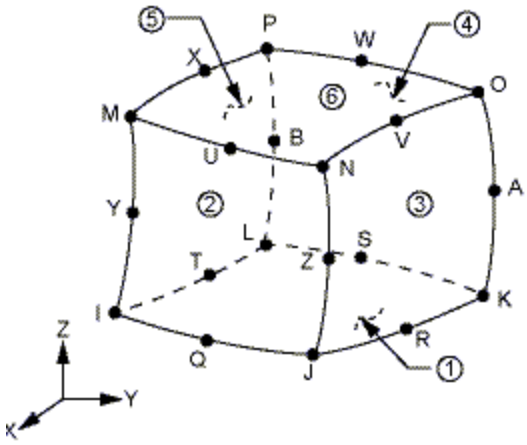


Fig 4.3 SOLID186 homogeneous Structural Solid Geometry

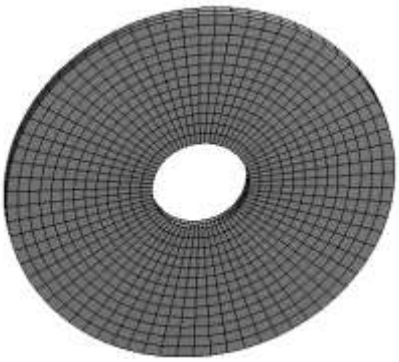


Fig 4.4 Meshing

Mapped meshing is done on both faces of the model to remove the irregularities in the meshing and making the mesh more refine.

4.6 BOUNDARY CONDITION

PRESSURE=150MPa

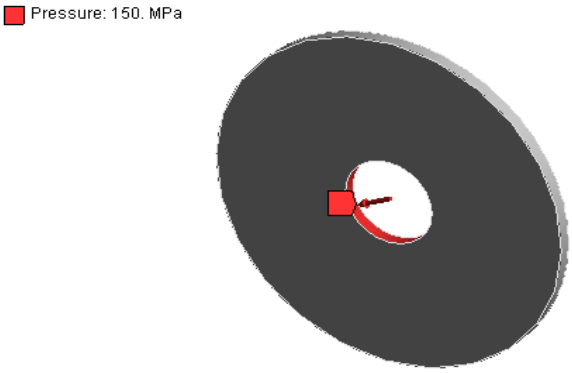


Fig 4.5Boundary Condition

Figure 4.3 shows the model with boundary conditions. Pressure of the magnitude 150MPa is applied at the inner surface of the hollow cylinder. For cylinder no fix support is required since its structure itself acts as fix boundary conditions.

V. DESIGN OF PROCESS REACTIONALLY VESSEL

5.1 Design Data

Design a pressure vessel for the following specifications

Table 5.1:Design specification for process reactionally vessel

Sr No.	Parameter Description	Parameter Code	Value
1	Internal Pressure	P	0.05MPa
2	External Pressure	Po	Atm
3	Process Volume	Vp	205m3
4	Expected Stagnant	VS	57m3
5	Buffer Volume	Vb	50m3
6	Vessel radius	R	2.5m
7	Tube porosity volume	Tp	25
8	Radius of tube sheet	R	2.5m
9	Tube diameter	Td	200mm

5.2 INTERNAL DIMENSIONS CALCULATIONS

=20.5m3<Vs(57m3)

Hence the vessel will be characterized as Process reactionary vessel.

5.3 CALCULATIONS FOR LOW PRESSURE APPLICATIONS (P< 1.4)

For P(0.05MPa)<1.4MPa andProcess reactionary vessel.

$NTD = 11.6m$

To determine L1,

$L_1 = 6.645m$

5.4 RECALCULATION OF VOLUME CONSIDERING TUBESHEET THICKNESS

Vp = 1.1(Vp'+Vr)+1.2($\pi \times Td \times Td$)(Tp/400)N

Where,

Vp'= process volume = 205 m3

Vr = residual volume =0.9754 m3

Td=tube diameter =0.2 m

Tp=tube porosity volume = 25

N=No of tubes = 4

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

Vp=226.61 m3

For Process reactionary vessel.

$NTD = 12.823\ m$

5.4.1 Internalpressure

Assume L_0 =(1.1) (NTD)

$L_0 = 1.1 \times 12.823$

$L_0 = 14.1053\ m$

Total Height ht = L0 + 2 L1

ht = 27.3953 m

$$P_i = 0.05 + \frac{\delta g h_t}{10^6}$$

= 0.3859 N/mm2 < 1.4 MPa

5.5 MATERIAL PROPERTIES

SA 516 Grade70

Maximum allowable stress (S) = 20000 psi = 138 MPa

Modulus of elasticity E = 200 Gpa

Poisson's ratio μ = 0.29

5.6 SHELL THICKNESS

S= 138MPa
El=0.7 longitudinal seam efficiency (circ stress)
Ee=0.85circ seam efficiency (long. stress)
Pi = 0.3859 MPa
R = 2500mm
From ASME SECTION VIII, div –I, UG27[3]

$$t_a = \frac{P_i \times R}{S \times E_l - 0.6 \times p_i}$$

$$t_a = 10.01$$

$$t_b = \frac{P \times R}{2SE_e + 0.4P}$$

$$t_b = 4.109$$

$$T_{req} = Max(t_a, t_b) + CA$$

$$T_{req} = 16.01$$nt = 18 mm

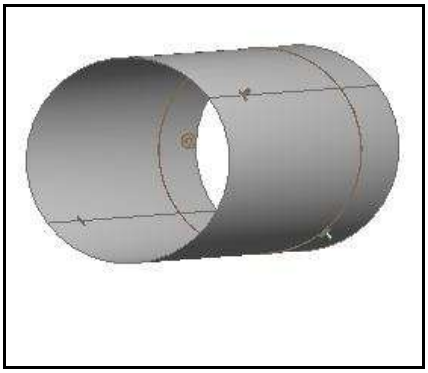


Fig 5.1 Shell render

5.6.1 Maximum Pressure

$$P_{i_1} = \frac{SE_l \times nt}{R_i + 0.6nt}$$

$$P_{i_1} = 0.6875$$

$$P_{i_2} = \frac{2SE_c \times nt}{R_i - 0.4nt}$$

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

$$P_{max} = 0.6875 > 0.3859$$

5.7 Thickness of head

Flat head

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

Fig 5.2 flat head render

5.8 NOZZLE DESIGN

Nozzle design parameters,
Nozzle Material: SA-51670
Sn=138 E = 1 (Nozzle efficiency)
nt = 18mm Nozzle diameter = 100mm
Number of nozzle = 4
Drain Nozzle =100mm
Do = 100 + 2 × 18 = 136

Nwall = 18mm
Lp = Exterior projection of nozzle = 3di
= 300mm
Lp=Exterior projection of nozzle = 0
nca = 6 (Nozzle corrosion allowance)
Pn=0.3859 MPa

$$R_n = \frac{D_o}{2} - (N_{wall} - nca)$$

= 56 mm

tn = Nwall – nca
tn = 12 mm
ti = Nwall- 2nca = 6 mm
d = Do-2tn d = 112mm

$$(Treq)n = \frac{p_n \times R_n}{S \times E - 0.6 \times p_n} + nca$$

$$= 6.15 < 18mm$$

$A_r = 1793.34$ mm2
A1=222.66 mm2
A2=461.88 mm2
A3=-72mm2
Aa=A1+A2+A3
=612.54<Ar
Reinforcement pad required

5.8.1 REINFORCEMENT PAD DESIGN

L0= NTD + (RF)OD
(RF)OD= 1.1×12.823-12.823
=1.2823×103mm
R = 641.15 mm

$$T_{reqRP} = \frac{P_n \times R_n}{S_n \times E - 0.6 \times P_n} + nca$$

=7.79mm
te=9mm
A1= 222.6602 mm2 A3= -72mm2
A2=461.8819 mm2 A5= 10341 mm2

Aa= A1+A2+A3+A5= 10953.54>Ar

For Dp= 300mm
Aa= 1530>Ar
Optimum Dp= 300mm
= 6.42016<9mm

Hence design is safe

L0=(NTD)+(Dp)RP
L0 = 13.213 m

5.9 Calculation of pressure

ht= L0+ 2L1
= 26.413 m

$$P = 0.05 + \frac{\delta gh}{10^6}$$

P = 0.3759 MPa

5.9.1 Shell thickness

$$t_a = \frac{P \times R}{S \times E - 0.6 \times P} + CA$$

= 7.19 + 6 = 13.191 < 15mm

5.9.2 For Nozzle

$$T_{reqN} = \frac{P_n \times R_n}{S_n \times E - 0.6 \times P_n} + nca$$

= 6.1157<15mm

$$T_{reqRP} = \frac{P \times R_p}{S \times E - 0.6 \times P} + nca$$

= 6.3018< 9mm

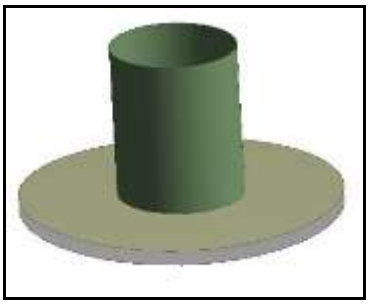


Fig 5.3 Nozzle with RF Pad render

5.10 SUPPORT SKIRT

Thickness of the skirt will be taken equal to the thickness of shell = 18mm
Total length of the skirt = 3m

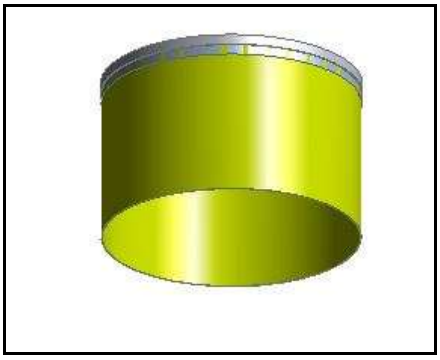
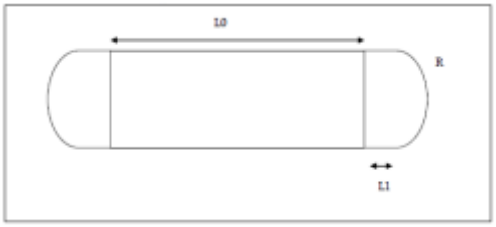
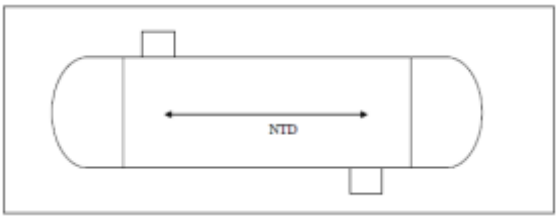


Fig 5.4 Support skirt render



Main stock Cylindrical length L0 = 11.9m
Buffer Stock Cylindrical Length L1 = 6.645m
Vessel Radius R= 2.5



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

Fig 5.5: Sectional view of vertical pressure vessel

Fig 5.6 Embedded pipes with partition plate

VI. CONCLUSION

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. After finalizing the model thermal analysis of u-shape embedded heat pipe were simulated. The temperature of naphtha heavy oil is between the ranges of 30-35 oC after 1900 seconds. So it can be concluded that these temperatures are within the acceptable limits.

REFERENCES

[1] Senthilkumar R1 Vaidyanathan S2, Sivaraman Thermal Analysis Of Heat Pipe Using Taguchi Method SenthilkumarR.Et. Al. / International Journal of Engineering Science and Technology Vol. 2(4), 2010, 564-569

[2] M Ram Ranjan, January -March 2010, “A numerical model for transport in flat heat pipes considering wick microstructure effects”

[3] JinKeumkim, Sciences (2 april1999), “Thermal Analysis of hydration heat in concrete with pipe cooling system

[4] Michael A. Porter Dennis H. Martens Pedro Marcal..”On Using Finite Element Analysis for Pressure Vessel Design”VOL .1, No 4, December 2012

[5] William C. Mohr, Panagiotis Michaleris, and Mark T. Kirk “An improved treatment of residual stresses in flaw assessment of pipes and pressure vessels fabricated from ferritic steels” Vol-3, December 2002..

[6] Zuersterschienen in: International Journal of Pressure Vessels and Piping 82 (2005), Seite 61–67 Direct finite element route for design-by-analysis of pressure components.

[7] J.Tian T.J. Lu, H.P. Hodson“Thermal-hydraulic performance of sandwich structures with crossed tube truss core and embedded heat pipes 13th International Heat pipe conference” (13th IHPC) Shanghai, china, September 21-25, 2004..

[8] HadiKhoramishad and Majid Reza Ayatollahi“Finite Element Analysis Of A Semi-Elliptical External Crack In A Buried Pipe” Received October 2008, Accepted August 2009 No. 08-CSME-43, E.I.C. Accession 3081.

[9] Qiuyuan Zhu , Xinhua Xu*, Jinhua Yu Study of Dynamic Thermal Performance of Active Pipe-embedded Building Envelopes Based on Frequency-Domain Finite Difference Method.

[10] Mohamed H. A. Elnaggar“The Effect Of Thickness And Permeability Of Wick Structure On L-Shape Heat Pipe Performance Using Different Working Fluids Frontiers In Heat Pipes” (Fhp), 3, 043004 (2012) Doi: 10.5098/Fhp.V3.4.3004
[11] Rathinasamy SENTHILKUMAR *,Subaiah VAIDYANATHAN, and Balasubramanian SIVARAMAN Thermal Analysis of Heat Pipe Using Self Reweting Fluids THERMAL SCIENCE, Year 2011, Vol. 15, No. 3, pp. 879-888

[12] Dr. Wajeeh Kamal Hasan Transient Three-dimensional Numerical Analysis of Forced Convection Flow and Heat Transfer in a Curved Pipe IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE) e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 9, Issue 5 (Nov. - Dec. 2013), PP 47-57

[13] Milind A. Patel and Krunal Patel, (September 2014)“Static Thermal Analysis And Experimental Evaluation Of Heat-Pipe Oven For Plasma Wakefield Accelerator Experiment”
[14] Peng-feiLIU†,Calculations of plastic collapse load of pressure vessel using FEAReceived Jan. 9, 2008; revision accepted Apr. 2, 2008

