International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 07 Issue: 08 | Aug 2020 www.irjet.net p-ISSN: 2395-0072

Design and Analysis of Filter Tube Sheet of Pressure Vessel Against Fatigue and Fouling

Mr. Vaibhav A. Sapate[1], Prof. Ravikant K. Nanwatkar[2], Prof. Sainand M. Jadhav[3]

[1]PG Student, Department of Mechanical Engineering, NBNSSOE, SPPU, Pune, Maharashtra.
[2,3]Assistant Professor, Department of Mechanical Engineering, NBNSSOE, SPPU, Pune, Maharashtra.

Abstract - *Many Petroleum industries use Filter sheet vessel for various applications. It generally deals with filtering the Natural gas immediately after it is mined. During these operations, natural gas contains common but serious contaminant e.g. sulphur in the form of H2S, SO2, CH4S, OCS and CS2. Of these sulphur combines with sand to create a rough particulate impurity like slogging, clogging, fouling etc. Filter tubes in pressure vessels provides a large surface for filtration. These tubes are made up of ceramics, steel etc and provide filtration up to 0.1 micro mm. But due to continuous use, these filter tubes gets layer of clog on surface which raises the overall pressure in the vessel. This pressure rise has some limits which cant exceeds beyond certain limits because of stability issues. Due to this reason the filter tubes are cleaned once a month/ week. This causes shut down of whole plant during cleaning which affects the economic costs due to loss of production. This problem can be reduced by designing a self cleaning filter unit which is divided into two sections and after every 4 to 5 seconds one of the compartments receives a reverse pressure of 1 sec and washes out the filter tubes. This system of cleaning of tubes without stopping the plant gives good results without reduction in production and since it is a regular cycle so its needs to be shut down the plant once a year for full clean-up. However, due to reverse pressure, the stress profile in the filter changes from tensile to compressive, creating possibility of fatigue. The main aim of this project work is to design of filter tube sheet on basis of working parameters, analyze the fatigue variation with modified concept using finite element analysis method to avoid fouling, slog and fatigue.*

Key Words: **FEA, tube sheet, fatigue, cyclic loads, de-clogging, Fouling.**

1. INTRODUCTION

The present work predicting the fatigue life of the tube sheets (used as main supporting element in large filter vessels), by design modification using proposed working cycle and transient analysis of the same. The application of the filters which is considered in this project is of petro chemical industries. This analysis finds application in natural gas filtering when it is immediately mined from the ores.

A tube sheet can be defined as perforated circular sheet or a plate with a pattern of holes (triangular / rectangular) specially designed to accept pipes or tubes. These tubesheet facilitate the smooth flow of fluid inside the tubes and also support and isolate the boilers, heat exchangers and other types of filter elements. The variety of material use to make tube sheet ranges from different types of metal, resin composites, ceramics or plastic. Cladding / insulating material / sacrificial /galvanic anode method is used as a corrosion barrier and insulator and may also be fitted with a galvanic anode which covers the tubesheet during its manufacturing.

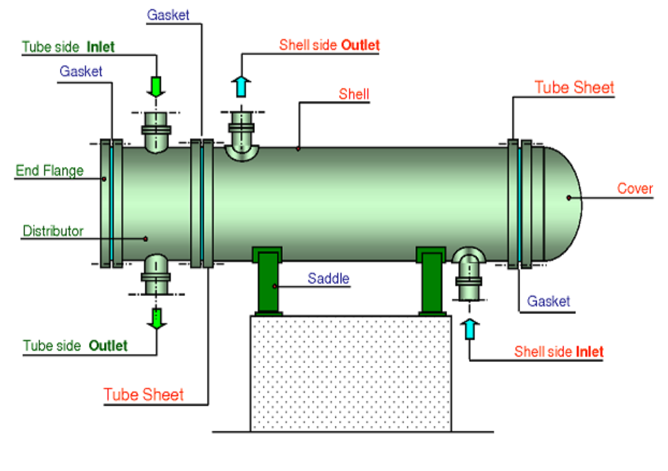


Figure1: Assembly of Pressure Vessel & tube sheet

Generally tubesheet are used in bundles in filters, boilers and heat exchangers with thin walled dense arrangements. These Tubes are supported on either end or in between according to specified design by sheets which drilled in specific pattern, generally triangular or rectangular to allow the tube ends to pass through the sheet. The ends of the tubes which are fixed in the tube sheet are expanded by means of flowing fluid and getting fixed in closed chamber to form a seal and makes a complete unit between the tube sheets. Flanges and tubesheet are bolted to make rigid unit to avoid any further design issues. The shell of heat exchanger extends beyond the end of tube sheet and is sealed, to form two closed chambers on the non-tube ends. An arrangement is formed where the heat exchanger / boiler consists of different chambers joined by tubes. Heated fluid generally in gaseous or liquid forms including other foreign materials from ore i.e. minerals etc. is then passed from one end to the other end of chamber through the tubes using various metallurgical processes like hydrometallurgy, pyrometalurgy or electrometallurgy. Generally these flowing fluid are high in temperature which creates to increase in temperature of tubesheet. Attempt can be made to increase the velocities of these fluids which otherwise can create fouling and creating cloggy surface on the tubesheet or on the surface of tubes. Also this fouling can be reduced by analysing the fatigue life behaviour of the tubesheet by transient analysis. Design calculation of tubesheet and its pattern to fixe tubes is somewhat complex process. These patterns include triangular or rectangular holes to have maximum tubes on the tubesheet with reduction in fouling and fatigue. Many computer aided technologies like CAD, CAM, and FEA are used now a day for safer design and analysis of the tubes and tubesheet. Tube sheets have pattern of holes with designed diameter and pitch. The portion between these holes is called ligament and the cross sectional area of the ligament w.r.to the area in a normal unpierced cross section of width is called ligament efficiency. The Stress Concentration Factor (SCF) is defined as the ratio of maximum principal stress in the stressed model to the nominal stress applied at the boundary of the plate. As tubesheet plays a vital role in design and analysis of the pressure vessel it should be carefully studied for various loads and working conditions. The thickness of the tubesheet varies directly to the costing and procurement of various component of pressure vessel. Thicker tubesheet results in longer tube length inside the tubesheet that do not take part during working operations. The primary aim of this project is to evaluate the fatigue life design of tubesheet by determining and analysing the effects of instant back pressure for cyclic loadings using Finite Element Analysis. The diameter, thickness and other design parameters are studied for given mechanical and working parameters for efficient and safe performance of the pressure vessel. Also various possible causes of fouling and clogging are theoretically analysed to find out suitable solution to avoid risk of fluing in tubesheet. This Research paper includes various design calculations using standard ASME codes for pressure vessel tubesheet. A mathematical modelling has been prepared by considering tubesheet as a flat plate with center hole, for verifying the designed and FEA solution. Further dynamic and transient analysis has been done with FEA software ANSYS for evaluating the fatigue life of the tube sheet.

1.1 Problem Definition

Tube Sheet filters is cleaned by applying back pressure, and the pollutants are then collected at the opposite end of the vessel. Industrial filters do this operation once a month. However in a Coal gas plant, the level of impurities is high, and this results in dense clogging of the filters within 5 days, and severe damage is caused to the filter and assembly due to the heavy built up of pressure as the tube get clogged. To tackle this problem, an instantaneous back pressure mechanism has been developed, which delivers 5 seconds of back pressure after 14 seconds of front pressure. This de-clogs the filters and reduces chances of pressure built up. However, due to back pressure, the stress profile in the filter changes from tensile to compressive, creating possibility of fatigue.

1.2 Objectives

1) To analyse the filter sheet for the load cycle specified.
2) To Study the possible causes and remedial measure of fouling on tube sheet surface.
3) To Study the effect of variation of parameters on the performance of the system. Since Flammable gases are involved in the process, safety is of prime importance.
4) To perform an ideal case analysis to benchmark the normal stresses during operation.
5) To conduct Transient analysis to check performance under a single cycle operation.
6) To perform Fatigue analysis to be undertaken to check the life of the component.
7) To modify the design if the fatigue life is not up to requirements.
8) To perform Tests will be to benchmark the modifications.

1.3 Scope

1) Three dimensional modeling of the tube sheet using any suitable CAD modeler, Viz. CATIA /ANSYS Workbench.
2) Finite element analysis (dynamic and transient) using suitable solver viz. ANSYS14.0.
3) Load cycle analysis by graphical methods using MS Office Excel and word.
4) Determination of fatigue and the stress profile on the filter due to pressure variation from tensile to compressive.
5) Comparative analysis with existing results of similar products.
6) Cost estimation for optimized solution.

1.4 Methodology

During normal operation, air flows inward through the intake duct and passes through the filter elements. Duct is collected on the outside surfaces of the elements. Clean air flows through the center of the elements into the clean air storage elements.

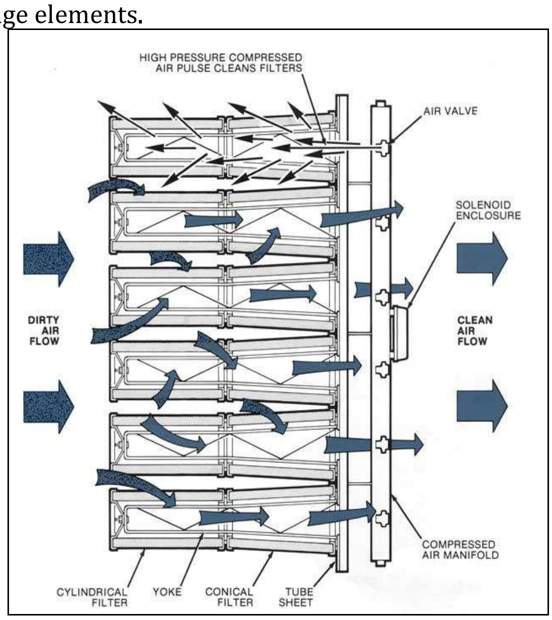


Figure 2: Basic operation of filter tube sheet

2. LITERATURE REVIEW

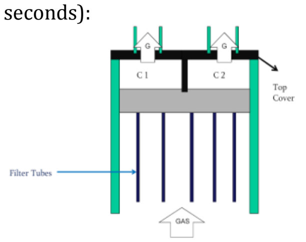
Title of Publication	Author and Publisher	Findings / Concluding remarks	Work to be completed / scope
Design of Perforated Plates	W.J. O'Donnell, B.F. Langer; JEIT, ASME; Volume 84. (2019)	Analysis of the stresses and deflection in the perforated plates with a triangular penetration pattern.	Fatigue and transient analysis with rectangular pattern. Fouling analysis by increasing the velocity of flowing fluid in tubes.
A finite element Benchmark for the Dynamic Analysis of Perforated Plates with a Square Penetration Pattern	D.L. Kaap, M.A. Sprague, R.L. Engelstad (2018)	for the dynamic analysis of perforated plates with a square penetration pattern	Effect of reduction in thickness of the tubesheet and its result on tubesheet stresses and design modification considering this changes.
Gas-side fouling, erosion and corrosion of heat exchangers for middle/low temperature waste heat utilization: A review on simulation and experiment.	Ming-Jia Li, Song-Zhen Tang, et.al. in ELSEVIER (2017)	The simulations and experimental studies for the fouling, erosion and corrosion of heat exchangers.	Fatigue and transient analysis with effect to reduce the fouling. Design modifications to reduce the stresses due to variation in thickness of tubesheet.
Strength Analysis of Tube to Tubesheet Joint in Shell and Tube Heat Exchanger"	Kotcherla Sriharsha, Venkata Ramesh Mamilla and M.V. Mallikarjun in (IJSETR),	The strength analysis of a typical tube to tube sheet joint in shell and tube heat exchanger.	Fatigue and fouling behavior with design modifications and FEA analysis though transient and dynamic approach.
Analysis of the Tubesheet Cracking in Slurry Oil Steam Generators.	L.K. Zhu,L.J. Qiao, X.Y. Li, B.Z. Xu, W. Pan, L. Wang, Alex A. Volinsky;in ELSEVIER.	Analysis of the tube sheet cracking in slurry oil steam generators Tubesheet to increase the service life.	Same as above with further work related to tubesheet in pressure vessel.

Conventional and Proposed Filtration Process

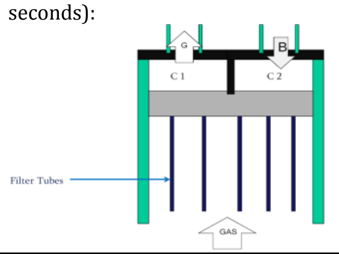
When natural gas is mined it contains contaminants. The most common and serious contaminants is sulphur, usually in the form of oxides, these sulphur at times combines with sand to create a rough particulate impurity. The gas is usually considered sour if the hydrogen sulphide content exceeds 5.7 milligrams of H_2S per cubic meter of natural gas. Filter tubes are the most commonly used filter options as they provide a large surface for filtration. They are typically made of ceramic and provide filtration up to 0.1 micro mm. However these filters do get clogged over a period of time. Once they get clogged they obstruct the flow of flow of gases, creating pressure rises. Once the allowable value is reached it need to shut down the plant and apply back pressure to clean the filter tubes. However when the filter tubes are being cleaned, the entire plant line has to be shut down. This comes with economic costs as the production of the pant reduces. Furthermore after restarting the system takes 30 mins to reach full capacity which further augments the production losses. Furthermore this cleaning is needed every month creating schedule losses.

To reduce the production losses and scheduled losses in the system a new system has been incorporated which reduces the compulsion of shut down of entire plant line for cleaning of filter tubes and also increases the overall productivity of the plant. The newly designed compartments are divided into two compartments as shown in figure after every 5 seconds one of the compartments receives a back pressure of 1 second, and cleans the filter tubes. This ensures cleaning without stopping the plant, plus since this is in regular cycles, plant shutdowns for full clean up are needed only once a year. During filtration process the compressed gas is passed through the gas inlet chamber to the Filter elements. This gas is collected at the top of the filter compartment. Blow pipes with nozzles are provided for providing the back pressure and clean the filter elements. A compressed air supply is provided for creating a high intensity back pressure. The process of creating back pressure is divided into number of cycles for cleaning number of chambers.

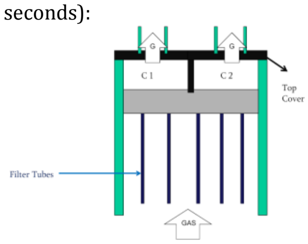
Initial operation ($0^{th}$to 5^{th} seconds):



Second operation (5^{th} to 6^{th} seconds):



Third operation ($6^{th}$to 11^{th} seconds):



Final operation (11^{th} to 12^{th} seconds):

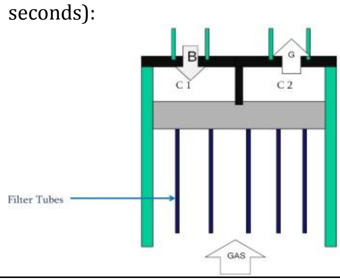


Figure 3: The proposed filtration process

The process repeats itself over and over again. However one of the crucial components is the filter sheet itself. This sheet stress reversals from positive to negative and is susceptible to fatigue.

3. DESIGN AND MATERIAL SELECTION

3.1 Material Selection

Materials are selected according to the following criteria.

1) Corrosive or noncorrosive service
2) Contents and its special chemical/physical effects
3) Design condition (temperature)
4) Design life and fatigue affected during the plant life
5) Referenced codes and standards
6) Low temperature service
7) Wear and abrasion resistance
8) Welding and other fabrication processes

During analysis and testing of tubesheet of pressure vessel in this case the material used is SA 516 GR70 having following properties,

Table1: Properties of the tubesheet material

Requirement	Grade 55	Grade 60	Grade 65	Grade 70
Carbon, max 0.5" and under	0.18%	0.21%	0.24%	0.27%
Manganese 0.5" and under	0.60-0.90%	0.60-0.90%	0.85-1.20%	0.85-1.20%
Tensile strength in ksi	55-75	60-80	65-85	70-90
Yield strength in ksi	30	32	35	38

Careful examination reveals that a material with maximum carbon content of 0.18%, manganese content of 0.90%, 70 ksi tensile strength and yield strength of 38 ksi will satisfy the requirements for all grades of SA-516. If the material also meets all other requirements of the specification, then it may be marked for all four grades of SA-516. The material taken for tubesheet is SGR 590, as per the specifications of ASME codes.

3.2 Design Calculation

For design calculations, ASMEVIII, Div 1 , TEMA codes and UHX of ASME BPVC part VIII-1 is used.

All necessary design values for the calculation are listed.

Table 2: Tubesheet Parameters for Design Calculations (Instant Back Pressure)

Sr. No.	Parameter Description	Notations	Given Value
1	Internal Pressure	P	0.14 MPa
2	External Pressure	P_0	Atmospheric
3	Process Volume	V_p	126 cu m
4	Expected Stagnant Volume	V_s	Not Specified
5	Buffer Volume Requirement	V_b	Not Specified
6	Tube Porosity Volume	T_p	70
7	Tube Length	T_L	5.5m
8	Radius of tube sheet	r	2m
9	Tube Diameter	d	0.15m

A 5% Gap will be maintained on the Tube Sheet radius to allow for welding. Tubes shall be spaced in a manner such that they form a 60 deg Equilateral Triangle.

Calculations for unknown parameters:

Total volume = Pressure Volume + Expected Stagnant Volume + Buffer Volume

V = Vp + Vs + Vb……………………… (1)

V = Vp + 0.1V + 0.01V…………………(2)

V (1 - 0.1 – 0.01) = Vp

V (0.89) =126 x 109.

V = 1.415730 x 1011mm3

Buffer Volume = Vb = 0.01V = 1.415730 x 109mm3

Stagnant Volume = Vs = 0.1V = 1.415730 x 1010mm3

Here, Vs > 0.1Vp

Hence, the vessel is characterized as a full process reactionary vessel.

Referring A2209, for full process reactionary vessel,

Vp = (0.90NTD) x (πr2) (NTD is nozzle to nozzle distance in meters).

126 x 109= 0.90 x NTD x π x (20002)

NTD = 11146.4968 mm

i.e. NTD = 11.146 metres

Now, Vs + Vb = (0.82L1) x (π r2)

Here, Vs > Vb, hence considering Vb = Vs

2 x 1.415730 x 1010 = (0.82L1) x (π x 20002)

Gives, L1 = 2749.203 mm.

i.e. NTD = 2.7492 metres

Calculations for Tube sheet volume (Tv):

$$Tv = \frac{\pi}{4}(3800)^2 \times T_t$$

Assuming Tt = 1mm

Tv = 11341.1494 x 103

The above volume is reduced value of actual tube sheet volume by 5% for welding space).

Calculation for tube volume:

$$\text{Total volume} = \frac{\pi}{4}{T_d}^2 \times T_L \qquad \ldots\ldots T_L = Tt$$

Tv = 17671.458mm3

Calculations for 'n' no of holes,

T / Tv = 0.3

$$0.3 = 1 - \frac{\textit{Tube volume}}{\textit{Tube sheet volume}} = 0.7$$

$$0.7 = \frac{\textit{Tube volume}}{\textit{Tube sheet volume}}$$

11341.1494 x 103 x 0.7 = 17671.458 x n

n = 449.244 nos≅ 450 number of holes.

Calculations for ligament efficiency,

$$\frac{\textit{Area remaining after drilling holes}}{\textit{area before drilling the holes}} = \frac{D - nd}{D}$$

$$= \frac{4000 - (22 \times 150)}{4000} = 0.175.$$

Calculations for Tubesheet thickness,

Referring ASME section VIII, div-I, page no. 34.

$$t = d\sqrt{\frac{C.P}{S.E}}$$

For, 1000C i.e. SA516 GR70
S = 20.0KSi
S = 137.895 N/mm2

$$t = 4000\sqrt{\frac{0.2\times 0.14}{137.895\times 0.175}} = 136.253m = 137mm$$

Recalculation of volumes considering tube sheet thickness,
Vp = 1.1 (Vp' + Tr) + 1.2 (Pi * Td * Td) * (Tp/400)N
But Tr/Tv = 0.3
Tr = 5301.43 mm3
Td = 150mm (Tube Diameter)
Tp = 70 (Tube Porosity Volume)
N = 450 nos. (No of tubes)
Vp = 1.14191781 x 1011 mm3
For full process reactionary vessel,
Vp = (0.90 NTD) x (πr2)
1.41 x 1011 = (0.90 x NTD) x (π x 20002)
NTD = 12548.290mm
i.e. NTD = 12.54m.
Now Total volume, V = Vp + Vb + Vs
Where, Vb = 0.01V and Vs = 0.1V
V = 1.41 x 1011 + 0.01V + 0.1V
V = 159.44943 x 109 mm3
Vb = 0.01V = 1594.4943 x 106 mm3
Vs = 0. 1V = 15944943 x 106 mm3
Now, Vs + Vb = (0.82 x L1) x (πr2)
Here, Vs > Vb,
Hence considering Vb = Vs
2Vs = (0.82 x L1) x (π x 20002)
L1 = 3094.776 mm = L1 = 3.094776 m

Design of Shell:
According to ASME Section-VIII, Division-I, UG27,

$$\text{Thickness of Shell (tshell)} = \frac{P.R}{SE-0.6P}$$

$$\text{tshell} = \frac{0.14\times 2000}{137.895-(0.6\times 0.14)} = 2mm$$

According to ASME Section-VIII, Division-I, UG32,

$$\text{Thickness of ellipsoidal head (thead)} = \frac{P.D}{2SE-0.2P}$$

$$\text{thead} = \frac{0.14\times 4000}{2\times 137.895-(0.2\times 0.14)} = 2.029mm$$

But according to According to ASME Section-VIII, Division-I, the thickness of shell as well as ellipsoidal head should be taken minimum 6mm.

Design of Nozzle:
According to ASME Section-VIII, Division-I, UG36,

$$\text{tm}= \frac{P_i d_i}{2\sigma J - P_i} = \frac{0.14\times 300}{(2\times 137.85\times 0.6)-0.14}$$

tm= 0.154mm
Similarly,

$$\text{tr}= \frac{P_i d_{ishell}}{2\sigma J - P_i} = \frac{0.14\times 2000}{(2\times 137.85\times 0.6)-0.14}$$

tr= 3.388mm
But as per ASME codes minimum thickness should be taken as 6mm.
h = 2.5(tr – CA) = 2.5(6 – 3) = 7.5mm
OR
h = 2.5(trn – CA) = 2.5(6 – 3) = 7.5mm
d= di + 2CA = 300+ (2 3) = 306mm.
x = d or whichever is maximum
x = 306mm
OR

$$x = \frac{d_i}{2} + t + t_n - 3CA$$

x = 153mm
Therefore, taking maximum value as, 306mm

Area calculation,
Area pertaining to material removed
i.e. A = d.tr = 306 3.388 = 1036mm2
Excess area in the shell,
A1 = (2x – d) (t – tr – CA)
= [(2 153)-306] [6 – 3.388 – 3] = 0
Excess area in the nozzle,
A2 = 2h1(tn – trn– CA) = 2x7.5(6 – 0.154 – 3)
A2 = 42.69 mm2
Excess area in the nozzle inside the shell,
A3 = 2h2(tn – 2.CA) = 2 7.5[6 – (2x 3)] = 0.
Therefore required area,
Ar = A – (A1 + A2 + A3) = 1036 – (0 + 42.69 + 0)
Ar = 993.31 mm2

Design of Reinforcement pad,
dip = internal diameter of RF pad
dop= external diameter of RF pad
t = thickness of RF pad
dip = di + 2tn = 300 + (2x 6) dip=312 mm.

$$\text{dop} = \frac{A_r}{t_p} + d_{ip} = \frac{993.31}{6} + 312 \quad \text{dop} = 477.55\text{ mm } 478\text{ mm.}$$

Design of flange,
(Referring PV Engineering design calculation as per ASME standards)
Load on projected area = pressure x projected area.
P = (0.14 x 4000) i.e. P = 560N.
Nomenclature,
A = Flange overall diameter = 162”
Bn= Flange internal diameter =157.5”
Tflange=Flange thickness = 0.5”
Rf = Hub corner radius = Tflange/2 = 0.25”
Gof = Hub thickness = 6mm = 0.25”
G1 = Hub base thickness = 6mm = 0.25”
m = gasket factor
Referring PV Engineering Data sheet,
B = Bn + 2CA = 157.5 + (2 0) = 157.5”
VarR = [(VarC – VarB)/2] – g1
VarC = Bolt circle diameter = (PCD) bolt= 162 – 2 = 160”
VarB = 157.5”
VarR = [(160 – 157.5)/2] – 0.25= 1”
GID = 157.5 + (G0/2)= 157.5 + (0.25/2)= 157.625”
GOD = 159”

$$\text{VarN} = \frac{G_{OD}-G_{ID}}{2} = \frac{159-157.625}{2} = 0.687''$$

G0 = N/2 = 0.687/2 = 0.3437”
b = if (b0> 0.25 then, , b0)
Here, b0 > 0.25

VarB = 0.293”
VarG = if [b0> 0.25, GOD – (2 VarB)] = 159 – (2 0.293)
VarG = 158.414”
Hub corner radius = Rf = Length of hub = 0.12”

Design of bolt

H = 0.785 (VarG)2 Pr
= 0.785 (158.414)2 20.305
H = 399.99 103Pounds……………end load
He to be considered wind pressure here,
He = [0.785 (VarG) 2 0.0265] < H
Hp = 2 VarB 3.14 VarG m P
= 2 0.293 3.14 158.414 3 20.305
= 17755.9974 Pounds……………..contact load
Wm1 = H + Hp = (399.99 103) + 17755.9974
Wm1 = 417745.99 Pounds……………..bolt load.

Bolt area required,

$$Am=\frac{W_{m1}}{\text{Allowable Load}}=\frac{417745.99}{20000}=20.887in^2$$ ……………..bolt area.

Now, Am= 0.87”……………for bolt area.

$$\text{No. of bolts} = \frac{A_m}{\text{Bolt root area}}=\frac{20.887}{0.87}=24nos.$$

Total length of the shell,

L0 = NTD + (RF Pad)OD = 12.54 + 0.3556
= 12895.6mm.
Reinforcement pad diameter = 14” = 355.6mm

Approved Design Modification after Testing,

- L1 which is currently calculated to be 3094 has been updated to 3200, as per site conditions.
- Filter Cover designers have made the following changes
- On top Side the no of nozzle required will be 4, 2 for positive pressure and 2 for negative (back) pressure.
- The back pressure nozzles are bigger than the other two nozzles, of size 700mm dia. The other two nozzles are to be made 500mm dia.
- Nozzle reinforcement pads are kept at 1400mm dia (2 x nozzle dia), the repad is kept same for all 4 nozzles.
- Previous tube SGR506, failed the back pressure tests, and it was found that larger dia filter tubes will be needed. Hence the tubes will be replaced by SGR590 tubes, of Dia 1400 mm.
- The Tube Sheet will be divided into two sections. Hence the central band of 100mm width will have no tube holes as it will be resting point for the portion.
- To compensate for this loss, the outer requirement of 5% space is reduced to 2.5% for the Tube Sheet.
- SGR 590 Tubes have the following properties, Dia 1400mm, length 4600mm, and weight 72.5kg.
- The Tube sheet will be welded to the shell from both sides.
- Top Nozzles will be at a distance of 1800mm from the Flange.
- NTD will be maintained at 12550 mm, therefore the bottom nozzles will be shifted and overall length of the vessel will increase by about 1100mm (approx. value, calculate exact value for modelling).
- Bottom Nozzle size will be increase to 700mm and there will be 2 nozzles at 180 degree orientation.
- Nozzle on the dish end will also be upgraded to 500mm dia; this is because SGR 590 has tendency of coagulating impurities into larger blocks.

Pressure Cycle

For SGR 590 the following pressure cycle is present for one chamber of the sheet.
0 - 5 sec 0.14 MPa (Up) A
5 - 6 sec 0 MPa
6 - 10 sec 0.145 MPa (Back) B
10 -11 sec 0 MPa
When chamber 1 is at cycle A, chamber 2 will be at cycle B. Both will be shut off at the same time.
During Shutoff the accumulation of gas will increase the upward pressure up to 0.143 MPa.
Since the entire process is working at differential of 0.005 MPa, Max upward pressure will be 0.145 MPa.
Design Pressure is hence considered to be 1.2 x 0.145 = 0.174 MPa which will be the test pressure.
Corrosion allowance is considered to be 3mm.

Modified design calculations:

Calculation for tubesheet thickness,
Referring ASME codes section VIII, division-I, UG 34, t = d
Where, C = factor considering the method of attachment (0.20 for fillet welding).
d = diameter of vessel
P = internal pressure
S = Allowable stress
E = efficiency (summation of ligament and joint efficiency).
For given material, i.e.SA516GR70, allowable stress = S = 20.0ksi = 137.895 N/mm2.

Volume of tubesheet (Tv),

Tv = (π/4)d2 Tt
Where, Tt = tubesheet thickness
Tv = total volume of the tubesheet
TL = tube length.
Tv = (π/4) (4000)2 Tt
Considering, Tt = 1mm
Tv = 12566.37 103mm3.
Reducing 2.5% space for welding, i.e. taking diameter = 3900 mm.
Tv = (π/4) (3900)2 Tt

Volume of tube (Tv),

Now tube diameter = 0.14 m = 140mm
Considering length of the tube = length of tube sheet,

$$\text{Tubesheet volume} = \frac{\pi}{4}T_D^{\,2}\times T_L$$ ………………..(TL = Tt)

$$\text{Tube volume} = \frac{\pi}{4}(140)^2\times 1$$

Tube volume = 15393.804 mm3.

Calculation for no. of holes on tubesheet,

Volume of holes = 15393.804 n
Residual volume = tubesheet volume – tube volume
TR = TV - TT
But TR/TV = 0.3

i.e. $0.3 = 1 - \frac{Tube\ volume}{Tube\ sheet\ volume} = \frac{15393.804 \times n}{11945.906 \times 10^3}$

n = 543.214 544 holes.

Total pitch = total diameter – (2.5%diameter) – diameter of hole = Total pitch = 3900 –140 = 3760mm.

for locating the holes,

Considering pitch as 1.1d, we get,

Pitch = 1.1 140 = 154mm.

i.e. 2b = 154mm

b = 77mm

a = b (tan60)

a = 77(tan60)

a = 133.3679mm.

Number of holes arranged = 509

Ligament efficiency, $\eta = \frac{D - nd_h}{D} = \frac{4000 - (24 \times 140)}{4000}$

$\eta = 0.16$

Now,thickness of tubesheet, $\sqrt{\frac{0.2 \times 0.14}{137.895 \times 0.16}}$

t = 4000

t = 142.49 150mm (minimum to be taken for tubesheet design as per ASME code).

4. EXPERIMENTAL ANALYSIS

4.1 FEA Analysis

Filter sheet bending failure usually follows a path from the center of filter holes pattern to outside surface of filter sheet. Fractures are usually asymmetric and it is often possible to decide whether the bending loads leading to fracture are the results of gas loads. High gas loads induce tensile stresses in the filter sheets.

In the project four different cases of loadings on tubesheet has been considered, to analyze the effects of cyclic loadings on tubesheet as follows, CAD Model of Tubesheet using CAD Modeler. Import this model in ANSYS workbench for Meshing. Mesh generation has been done using HEX and TET element of ANSYS 14.5. Results has been tabulated for both elements considering different load conditions to evaluate the convergence factor and nature of curve between number of nodes vs deformation & stress value. Thus Fatigue life has been evaluated using transient analysis in ANSYS 14.5 solver. Various testing results have been compared with FEA solution to determine infinite fatigue life behavior.

Case I: Self Weight and Gravity Load (TET element)

No. of Nodes	Element Size	Max Deformation (mm)	Max Stress (Mpa)
110850	95	0.77228	38.015
149744	64	0.80331	38.231
222416	54.5	0.81815	38.352
250544	50	0.82115	38.361
298837	37.5	0.82449	38.36

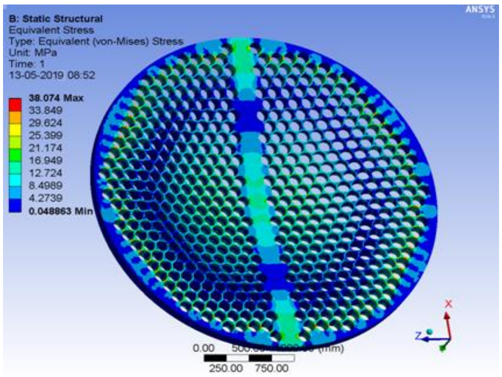


Figure4: Maximum stress in MPa (meshing with 2.5L nodes)

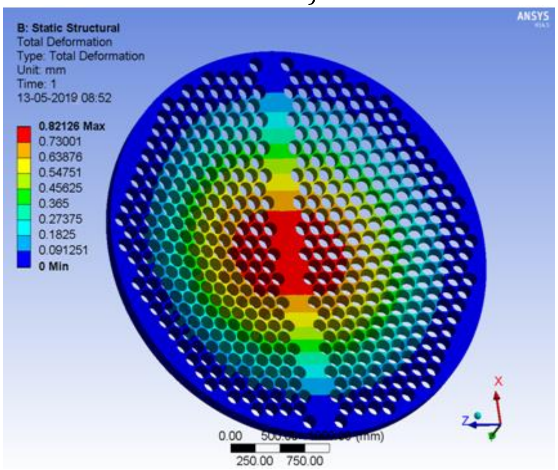


Figure 5: Maximum Deformation in MPa (with 2.5L nodes)

Case II: Gravity with Design Loads in opposite direction of gravity (TET element)

No. of Nodes	Element Size	Max Deformation (mm)	Max Stress (Mpa)
110814	95	0.67284	34.065
150019	64	0.70022	34.037
222053	54.5	0.71235	35.264
250225	50	0.71512	35.013
297126	37.5	0.71817	35.94
355759	35	0.71837	35.148

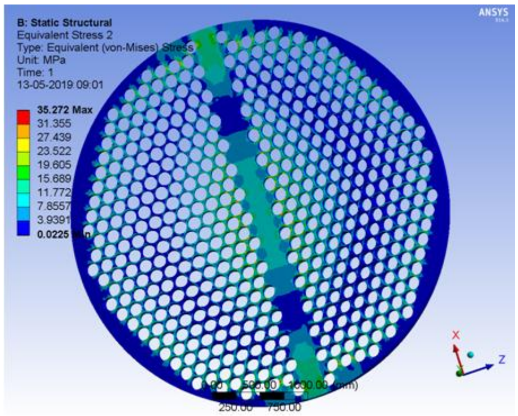


Figure 6: Maximum stress in MPa (meshing with 2.5L nodes)

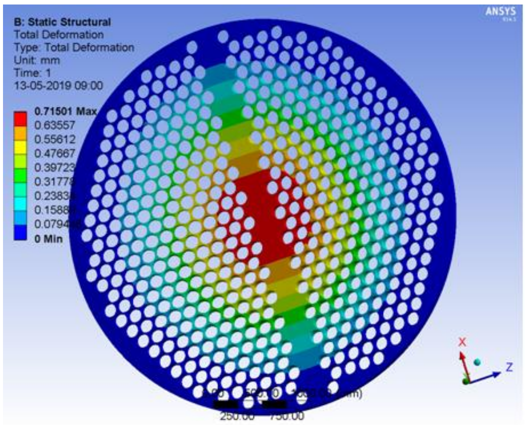


Figure 7: Maximum Deformation in MPa (meshing with 2.5L nodes)

Case III: Gravity with Back Pressure in the direction of gravity (TET element)

No. of Nodes	Element Size	Max Deformation (mm)	Max Stress (Mpa)
110814	95	1.9762	97.342
150019	64	2.0559	96.735
222053	54.5	2.029	97.457
250225	50	2.1	99.829
297126	37.5	2.1083	98.742
355759	35	2.1091	98.327

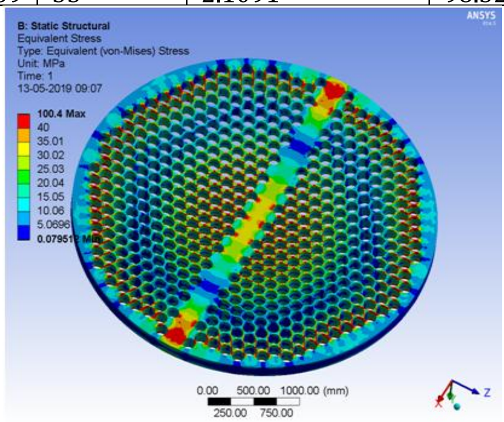


Figure8: Maximum stress in MPa (meshing with 2.5L nodes)

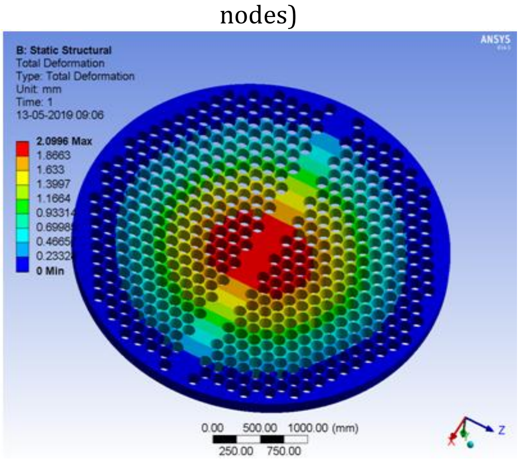


Figure 9: Maximum deformation in MPa (with 2.5L nodes)

Case IV: Positive and Negative Pressure both at a Time (TET element)

No. of Nodes	Element Size	Max Deformation (mm)	Max Stress (Mpa)
110853	95	0.9539	58.766
149744	64	0.99743	60.717
222416	54.5	1.0172	61.575
250544	50	1.0212	62.146
298837	37.5	1.0255	62.056
355381	35	1.0256	62.101

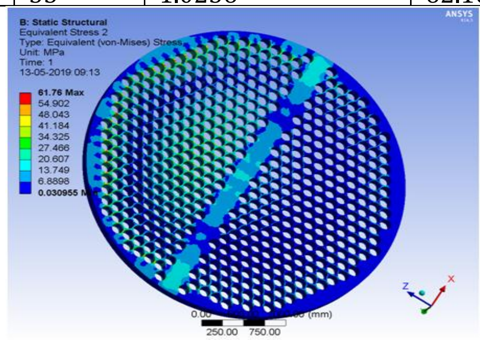


Figure 10: Maximum stress in MPa (with 2.5L nodes)

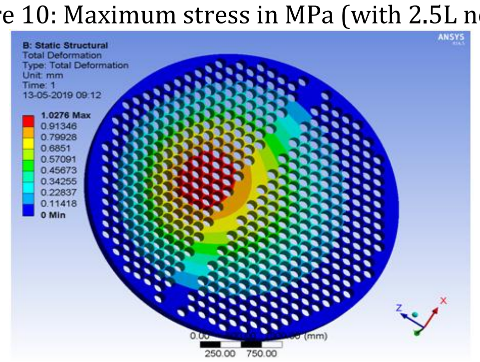


Figure 11: Maximum deformation in MPa (2.5L nodes)

Similar results were obtained for the same case (1 to 4) with "Hex" element for comparative analysis.

4.2 Transient Analysis

CASE I: (Without gravity)

Time	Wmax	Max Stress	Time	Wmax	Max Stress
0.2	1.15E-02	1.1286	6.2	1.32E-02	1.2333
0.4	2.31E-02	2.2638	6.4	2.17E-02	2.0087
0.6	3.34E-02	3.2746	6.6	3.52E-02	3.2843
0.8	4.50E-02	4.4112	6.8	4.44E-02	4.13
1	5.64E-02	5.5351	7	5.84E-02	5.4384
1.2	6.80E-02	6.6711	7.2	6.80E-02	6.3314
1.4	7.95E-02	7.7958	7.4	8.15E-02	7.5926
1.6	9.11E-02	8.9308	7.6	9.17E-02	8.5312
1.8	0.10254	10.056	7.8	0.10471	9.7498
2	0.1141	11.19	8	0.11524	10.728
2.2	0.12559	12.317	8.2	0.12793	11.911
2.4	0.13715	13.45	8.4	0.13878	12.92
2.6	0.14864	14.577	8.6	0.15119	14.076
2.8	0.16019	15.709	8.8	0.16228	15.11
3	0.17169	16.837	9	0.17447	16.244
3.2	0.18323	17.968	9.2	0.18576	17.296
3.4	0.19474	19.097	9.4	0.19779	18.416
3.6	0.20628	20.228	9.6	0.20921	19.48

3.8	0.21779	21.357	9.8	0.22112	20.589
4	0.22932	22.487	10	0.23264	21.662
4.2	0.24084	23.616	10.2	0.24448	22.764
4.4	0.25237	24.746	10.4	0.25606	23.843
4.6	0.26389	25.875	10.6	0.26784	24.94
4.8	0.27542	27.005	10.8	0.27947	26.024
5	0.28694	28.134	11	0.29121	27.118
5.2	4.91E-04	4.38E-	11.2	4.98E-04	4.18E-
5.4	1.27E-03	0.1135	11.4	1.32E-03	0.11098
5.6	1.74E-03	0.15567	11.6	1.82E-03	0.15286
5.8	2.40E-03	0.21472	11.8	2.51E-03	0.21115
6	2.56E-03	0.2296	12	2.68E-03	0.22592

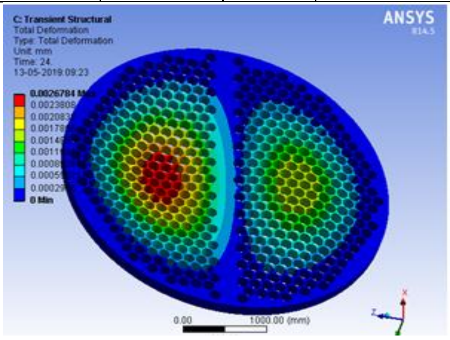


Figure 12: Maximum deformation in mm(Transient)

CASE II: Considering Positive and Negative Pressure both at a Time (HEX Element)

Time	Deforn	Stress	Time	Deforn	Stress
0.2	0.8152	43.661	6.2	0.82089	43.108
0.4	0.8338	45.269	6.4	0.82521	43.789
0.6	0.8169	45.389	6.6	0.83221	45.082
0.8	0.8510	47.759	6.8	0.83748	45.906
1	0.8269	47.556	7	0.84438	47.155
1.2	0.8629	50.014	7.2	0.84982	48.025
1.4	0.8411	49.885	7.4	0.85773	49.229
1.6	0.8746	52.164	7.6	0.86535	50.141
1.8	0.8606	52.276	7.8	0.87481	51.306
2	0.8885	54.284	8	0.88272	52.254
2.2	0.8807	54.672	8.2	0.89195	53.386
2.4	0.9026	56.414	8.4	0.90006	54.363
2.6	0.9003	57.048	8.6	0.90912	55.47
2.8	0.9174	58.57	8.8	0.91754	56.468
3	0.9196	59.396	9	0.9267	57.557
3.2	0.9332	60.754	9.2	0.93527	58.571
3.4	0.9384	61.717	9.4	0.94432	59.646
3.6	0.9496	62.963	9.6	0.95297	60.672
3.8	0.9567	64.015	9.8	0.96196	61.737
4	0.9664	65.192	10	0.97093	62.77
4.2	0.9750	66.296	10.2	0.98036	63.869
4.4	0.9844	67.434	10.4	0.98997	64.949
4.6	0.9938	68.565	10.6	1	66.046
4.8	1.0035	69.685	10.8	1.0101	67.13
5	1.014	70.828	11	1.0206	68.223
5.2	0.8142	42.737	11.2	0.81481	42.725
5.4	0.8149	42.699	11.4	0.81379	42.784
5.6	0.8140	42.842	11.6	0.81555	42.709
5.8	0.8150	42.675	11.8	0.81347	42.836
6	0.8141	42.874	12	0.81574	42.704

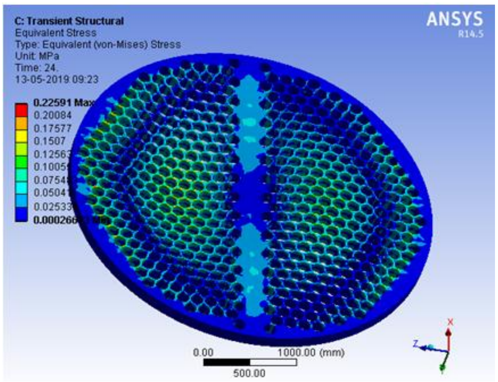


Figure 13: Maximum stress in MPa (Transient)

4.3 Experimental validation

1) All new product equipments were be tested at 2.5 times the operating pressure using Hydro test.
2) The Hydro test has slow built up of pressure, from base pressure to test pressure over a period of 120 min.
3) The equipment is maintained at test pressure of 30 min.
4) The pressure gradually reduced to base pressure within a period of 45 min.
5) After test, all components subjected to NDT as below,
   a) No surface irregularities must be present.
   b) Pre Dyed components should have no loss of dye due to leakage.
   c) Ultra Sonic Testing – Post Test, internal damage shall get amplified if any, and shall be recorded in an Ultra Sonic Test.
6) The test performance of the assembly should be completely elastic; this shall be verified by checking the dimensions of product for any permanent yield.

Hydro Test Condition

Working fluid : Water with Anti Scaling Additives
Test Pressure : 2.5 x 0.07 MPa
Leak Inspection : Sensors (LDR) on the top side of Filter assembly.
Method : Visual Inspection on top side after completion of test.
Remark : Simultaneous testing of all 7 chambers was done. Filter holes were plugged with caps of SA 204.

Test Execution Details

- Begin Time : 09.00 hrs
- Base Pressure : 0 MPa (Empty vessel)
- Peak Pressure Time : 11.00 hrs
- Peak Pressure : 0.175 MPa
- Pressure relief begins Time : 11.30 hrs

Visual Inspection Details

- No leak observed on Top Side of Assembly
- No visible damage observed after test.
- Plug Adhesion intact after test.

Auditors Remarks:

- Code requirements have been met by the analysis.
- The Mesh is satisfactorily fine enough to generate accurate results with considering boundary conditions.

- The maximum Stress in Filter sheet is 32 MPa, however nominal value if calculated is much lower, it satisfy FOS is 5.
- Gasket plate shows peak pressure of 34 MPa. However it is observed to significant stress raiser due to vicinity of contact and relatively less thickness of the plate compared to the other components.
- Material Non Linearity may not be modeled in future analysis as it will have negligible effect on accuracy and unnecessary increases solution time.
- FEA processing has been done in line with requirements of SA 516 GR70, FEA and the component maintains a FOS greater than 5 for the current boundary conditions.

5. CONCLUSION AND FUTURE SCOPE

5.1 Conclusions

- The filter tube sheet is analyzed for proposed load cycle of four stages compared to conventional and found satisfactory results with FEA and transient analysis to reduce fatigue.
- Theoretical approach of studying possible causes and remedial measure of fouling on tube sheet surface conclude that fouling can be reducing by design modification and implementing the mechanism by increasing the velocity of flue gases.
- Modified design calculations shows acceptable effect of variation of parameters on the performance of the system.
- Analysis results are reliable as seen in Mesh Sensitivity convergence and actual Testing.
- Concerned with FEA analysis more accurate results are achieved using HEX element compared to TET element with fine meshing but increased time.
- FEA Validation shows we can increase efficiency of Filter sheet by increasing number of tubes and maintaining Factor of Safety 5, to benchmark the normal stresses during operation.
- Transient analysis done for predicating the fatigue life shows satisfactory results.
- Transient analysis illustrates the scope for enhancement of infinite fatigue life of the tubesheet which will increase the overall efficiency.

5.2 Future Scope

- Further analysis can be done for different components of the pressure vessel such as shell, flange, support etc for evaluating the results to improve efficiency and life of the pressure vessel.
- Similar transient and dynamic analysis can be performed for thickness optimization of tubes as well as tubesheet.
- Design modification and its FEA analysis can be performed for different patterns of tubesheet hole and insertion in between the tube pathways to support the tubes with different materials, distances and thickness.
- Further Transient analysis can be performed for velocity and pressure calculations of flowing fluid effect on pressure vessel components using CFD analysis, so as to reduce time for fouling by the suit particles and other fouling materials.

ACKNOWLEDGEMENT

It is indeed a great pleasure and moment of immense satisfaction for me to present a seminar report on **“Design and Analysis of Pressure Vessel Filter Tube Sheet against Fouling and Fatigue”** amongst a wide panorama that provided us inspiring guidance and encouragement, I take the opportunity to thanks to thanks those who gave us their indebted assistance. I wish to extend my cordial gratitude with profound thanks to our internal guide **Prof. R. K. Nanwatkar and Prof. S. M. Jadhav.** It was their inspiration and encouragement which helped me in completing my work. My sincere thanks and deep gratitude to Head of Department, **Prof. D. H. Burande** and other faculty member; but also to all those individuals involved both directly and indirectly for their help in all aspect of the project.

At last but not least I express my sincere thanks to the Institute’s Principal **Dr. S. D. Markande,** for providing us infrastructure and technical environment.

REFERENCES

[1] “Design of Perforated Plates”W.J. O'Donnell, B.F. Langer; Journal of Engineering for Industry; Trans. ASME;Volume 84 (August 1962); Paper no.61-WA-115.

[2] “A finite element Benchmark for the Dynamic Analysis of Perforated Plates with a Square Penetration Pattern”, D.L. Kaap, M.A. Sprague, R.L. Engelstad; Fusion Technology Institute University of Wiscons;UWFDM-1034 May 1997.

[3] “Comparisen and Design and Analysis of Tubesheet Thickness by using UHX code of ASME and TEMA standerds”,Kalepesh D. Shirode, Dr. S.B. Rane, Mr. Yashawant Naikinternational journal of mechanical engineering and technology (IJMET), ISSN 0976-6359,volume-4, issue-4, july-august 2013, pp 105-117.

[4] “Strength Analysis of Tube to Tubesheet Joint in Shell and Tube Heat Exchanger”Kotcherla Sriharsha, Venkata Ramesh Mamilla and M.V. Mallikarjun in International Journal of Science, Engineering and Technology Research (IJSETR), Volume 1, Issue 4, October 2012, ISSN: 2278-7798.

[5] “Finite Element Analysis of a Tubesheet with considering effective geometric properties through design methodology validated by Experiment”, Ravivarma. R, Azhagiri. Pon, International Journal of Computational Engineering Research (IJCER), ISSN (e):2250-3005, volume-4, Issue-4, April 2014.

[6] “An Approch to Finite element Analysis of Boiler Tubesheet”R.D. Patil, Dr. Bimlesh Kumar,American Journal Of Engineering Research(AJER),e-ISSN:2320-

0847, p-ISSN:2320-0936,Volume-2, Issue-8, PP-8-11.

[7] "Thickness and Shape Optimization of Filter" Naik, Shweta, Global Journal of Researches in Engineering; Mechanical and Mechanics Engineering; Volume-13; Issue-1;Version-1.0;Year-2013.

[8] "ASME Section III Stress Analysis of a Heat Exchanger Tubesheet with a Misdrilled Hole and Irregular or thin ligaments" Dr. Enrique Gomez, Mr. Roberto Ruiz, Mr. Robort M. (CON) Wilson, proceedings of the ASME 2013 pressure vessels and piping conference PVP 2013 July 14-18, 2013 paris, France.

[9] "Analysis of the Tubesheet Cracking in Slurry Oil Steam Generators" L.K. Zhu, L.J. Qiao, X.Y. Li, B.Z. Xu, W. Pan, L. Wang, Alex A. Volinsky; ELSEVIER science directory journal, engineering failure analysis 34(2013) 379-386.

[10] "3-D finite element analysis of roller-expanded heat exchanger tubes in over-enlarged tubesheet holes"N. Merah, A. Al-Aboodi, A. N. Shuaib, Y. Al-Nassar in article; Appl Petrochem Res (2012) 1:45-52; DOI 10-1007/s 13203-011-0005-z.

[11] "Stress Analysis of Tubesheet in waste Heat Boiler" Minshan Liu, Qiwu Dong, XinGu, journal of pressure equipment and systems 4(2006) 48-51;Thermal Energy Engineering Research Centre of Zhengzhou University, Zhengzhou 450002, China,

[12] "Optimization of Advanced filter system"R. A. Newby, G. J. Bruck, M. A. Alvin, T. E. Lippert, Westinghouse Science & Technology Centre; Worked Under Contract Number DE-AC26-97FT3300703 August 20, 1997 - April 1998.

[13] "Advance Mechanics of Materials" Arthur P. Boresi, Richard J. Schmidt, John Wiley & sons, Inc; sixth edition.

[14] "Pressure Vessel Design Manual" Dennis Moss, ELSEVIER publications, third edition.

[15] "Pressure Vessel Design" Guides and Procedures, P.V. Engineering.

[16] "CASTI Guidebook Series - Vol. 4 to ASME Section VIII Div. 1 for Pressure Vessels" Bruce E. Ball, Will J. Carter.

BIOGRAPHIES

Prof. Vaibhav Sapate, PG Student, Design Engineering of NBNSSOE, Ambegaon, Pune and Ex-Faculty of Department of Mechanical Engineering, of Sou. Venutai Chavan Polytechnic college, Vadgon.

Prof. Ravikant K. Nanwatkar, Faculty of Department of Mechanical Engineering, of STES's NBNSSOE, Ambegaon, Pune. His area of specialization is CAD/CAM and research area is CAD/CAM, Design, Mechanical Vibration, Material Science and Metrology.

Prof. Sainand M. Jadhav, Faculty of Department of Mechanical Engineering, of STES's NBNSSOE, Ambegaon, Pune. His area of specialization is CAD /CAM and research area is CAD/CAM, Mechatronics, Vibration.