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Design Calculation for Horizontal Type of Heat Exchanger & Its Components

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ABSTRACT

Based on various ASME & Indian standard codes, For the design of shell and tube heat exchangers, the tube sheet are usually very heavy. The reason is attributed to the oversimplified mechanical model for the calculation of the tube sheet thickness especially the insufficient consideration of the tube support to the tube sheet. Due to its interaction with pressure vessel and the resulting stress that it generates, the design of tube sheet becomes extremely complex. The location where the tube sheet is attached, radial expansion of the vessel is halted; this creates bending stresses in the vicinity of the tube sheet hence the resulting stress profile becomes increasingly complex..This paper deals with the design calculation related to heat exchanger and its various components .Detail calculation of various parameter and the factors associated with the components are calculated and analyzed.

KEYWORDS: ***Heat exchanger, Effectiveness, corrugated tube, saddle***

1. INTRODUCTION

Heat exchangers are frequently called as the workhorses in process and petrochemical plants and more than 65% of these are tubular heat exchangers. Tubular heat exchangers exemplify many aspects of the challenges in the mechanical design of pressure vessels. Their design requires a thorough grounding in several disciplines of mechanics and a broad understanding of the interrelationships between the thermal and the mechanical performance of the heat exchanger [1].There are many codes and standards available to design the heat exchanger components. Widely known codes are ASME Sect. VIII Div.1, 2, EN13445, TEMA, CODAP etc. Large heat exchangers are made of expensive materials which are cost effective components in industrial plants.For these, a thorough understanding of code differences is of paramount importance. Results of code comparisons exist and a few are published .A tube sheet is sheet, a plate, or bulkhead which is perforated with a pattern of holes designed to accept pipes or tubes. These sheets are used to support and isolate to tubes in heat exchangers, filter and boilers support elements. Depending on the application, a tube sheet may be made of various metals or of resin composites or plastic or steel.. A tube sheet may be covered in a cladding material which useful for a corrosion barrier, temperature and pressure effects and insulator and may also be fitted with a galvanic anode. Tube sheets may be used in pairs in heat exchange applications or singularly when supporting elements in a filter.

The studies of existing system in pressure vessel one or two tube are used with small size vessel. Here in this project is totally new design that is proposed there are two tube sheets at equal intervals and combination of three pressure vessel. In this design arrangement of tube-sheets are equally spacing distance and vessel size will be large as compare to existing .design of all model by using ASME Code Section-8, Div-2. Finite Element Analysis based Structural analysis of Horizontal Tube Design and analysis is very critical in refinery and heavy equipment industries. The optimized designs parameters are not only reduce the material cost but also help to define scope of research. In shell and tube heat exchanger tube sheet acts as the main pressure boundaries between shell side and tube side chambers. It is therefore exposed to the operating transients of both fluids of heat exchanger. Tube sheet is a key component of heat exchanger since it is directly connected to three major components of heat exchanger. This subjects the tube sheet to reactive loads in addition to pressure and thermal loads. The magnitude of the reactive loads is a result of complex interaction between the tube sheet and corresponding connected parts. Theoretically evaluation and analysis of tube sheet is one of the important and challenging tasks for designer. The possibility of optimizing the thickness of the tube sheet with better knowledge of its state of stress has fuelled the researchers and engineers towards refinement in design and analysis procedure. The thickness of the tube sheet affects the cost of the heat exchangers in many ways.

Increased thickness necessitates procurement of heavy and costly forging plate with difficult to achieve uniform acceptable mechanical properties. Thicker tube sheet results in longer tube length inside the tube sheet that do not take part in heat transfer. This unused tube length adds to the procurement length and cost. The primary aim of tube sheet design is to determine and optimize the pitch pattern of the tube holes, the diameter and the thickness for known mechanical and thermal loads for efficient and safe performance of the heat exchanger. To obtain optimum tube sheet thickness, codes and standards are to be compared which makes the tube sheet design as an iterative process. In these work the codes and standards used in the investigation are UHX part rules for design of tube sheet from ASME section VIII div. 1 and Appendix A from TEMA standards which was then analysed by FEA software ANSYS 13. Structural integrity of pressure vessels and heat The exchangers depends on proper mechanical design arrived at after detailed stress analysis keeping in view all the static, dynamic, steady, and transient loads[8]. Therefore, an optimum mechanical design of various components of heat exchangers is of paramount importance. Mechanical design involves the design of pressure-retaining, non-pressure retaining components and equipment's to withstand the design loads, the deterioration in service. It is also possible that formulae developed by some standards have higher factor of safety, which leads to some times over design. Over the course of time, finite element models have gained significant importance, and research has been ongoing to establish a supportive results to hand or software calculations. Computer models (CAE), have been developed to provide timely and economical simulations for results of a component under extremely sever loading conditions [3]. Many manufacturing industries nowadays, prefer finite element analysis of pressure vessel & heat exchanger component, because the simulations can be used to target sensitive parameters that affect the overall design, cost and safety of equipment. Going through literature review, many authors presents their work by using different methodologies for design and analysis of tube sheet. K. Behseta, S. Schindler has present the work on the design of the tube sheet and the tube sheet-to-shell junction of a fixed tube sheet heat exchanger in which they compare the ASME Sect. VIII Div.1 and EN 1344-3 clause 13and Annex J for their investigation.

1.1 Heat Exchanger

The heat exchanger considered in this paper is a Fixed Tube Sheet Shell-and-Tube heat exchanger of Waste Heat Boiler. In this heat exchanger the flue gases are flowing from tubes and the steam is from the shell. The geometrical dimensions and design data for the investigation are:

Shell Inner Diameter: 1984mm
Inlet channel Inner diameter: 3510mm
Tube outer diameter: 88.9mm
Tube hole Pitch: 124.5mm
Tube hole diameter: 89.3mm
Tube thickness: 7.62mm
Inner tube sheet to tube sheet distance: 6706mm
Shell thickness: 45mm
Channel thickness: 20mm
No. Of tubes: 148.
Baffle thickness: 20mm
Shell side pressure: 4.75 MPa
Tube side pressure: 0.35 MPa
Tube sheet design temperature: 343 ^{0}C
Shell design temperature: 343 ^{0}C
Channel design temperature: 343 ^{0}C
Tube design temperature: 343 ^{0}C
Tube inlet: 1416 ^{0}C
Tube outlet: 649 ^{0}C
Shell operating: 248 ^{0}C

As the flue gas inlet temperature is very high, to protect the welded tube-to-tube sheet joint from high temperature the ferrules are used. The channel and tube sheet are insulated from inside. The operating temperatures of tube sheet, channels are obtained from the thermal analysis. The material properties such as modulus of elasticity, allowable stresses, yield strength, coefficient of thermal expansion, thermal conductivity etc. for different materials are obtained from ASME Sect. 2 Part D. The geometrical details of fixed tube sheet heat exchanger are shown in Fig.1.

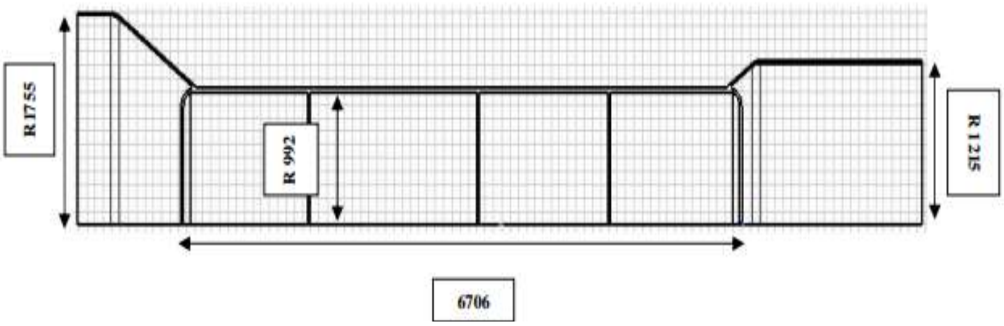

Figure 1:- Structure of heat exchanger

1.2:-Tube sheet

A tube sheet is sheet, a plate, or bulkhead which is perforated with a pattern of holes designed to accept pipes or tubes. These sheets are used to support and isolate to tubes in heat exchangers, filter and boilers support elements. Depending on the application, a tube sheet may be made of various metals or of resin composites or plastic or steel. A tube sheet may be covered in a cladding material which useful for a corrosion barrier, temperature and pressure effects and insulator and may also be fitted with a galvanic anode. Tube sheets may be used in pairs in heat exchange applications or singularly when supporting elements in a filter. Tube sheets are also used on cartridge-type filter devices to support the individual filter elements. They are similar in design to the high heat boiler varieties except they are typically made of resin composites or plastic and are generally used as single units. There are usually fewer tubes involved in a filter application although the tube sheet design still has to be carefully calculated to ensure optimal performance.


Figure 2:-CAD model of tubesheet

Generally, Holes arranged in the uniform pattern are in a plate is popularly known as tube plate ortubesheet. Holes in the tube sheet can be arranged in three different patterns.

- Equilateral Triangular Pattern
- Square Pattern
- Staggered Square Pattern

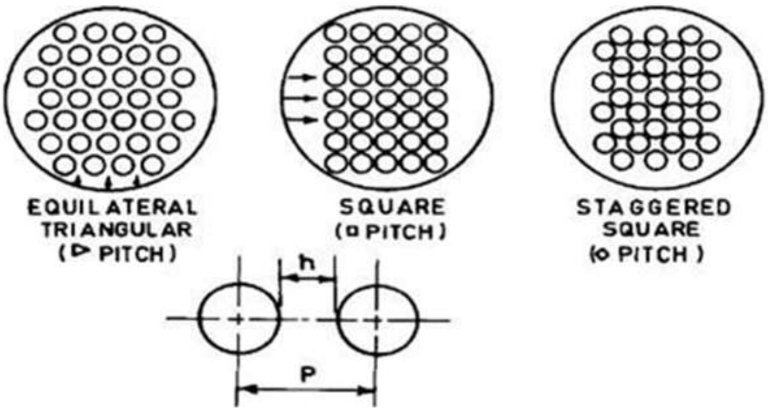

Figure 3:-Three Different Types of Hole Patterns

Out of these patterns, the equivalent triangular arrangement is the most widely used as it is the most effective packing arrangement

2. LITERATURE REVIEW

H.F. Liinvestigate the possible mechanical causes of a real tube-sheet cracking by simulates the tube sheet under different loading condition. They took three different loading conditions, namely residual expansion stress, crack face pressure and transverse pressure, and three crack growth patterns were considered .They conclude that;

- If the tube-to-tube-sheet expansion is through-thickness, the residual contact pressure or the residual expansion stress will be a long distance force acted on the tube-sheet along thickness.
- In all the cases studied, cracks at the tube-sheet on the tube side keep open once initiated, which is especially dangerous under some aggressive media conditions by leading to stress corrosion cracking of the tube-sheet.
- Surface crack propagation may be driven by all the three loadings especially the transverse pressure.
- When surface cracks come into the interior of the tube-sheet along the thickness, instead of the transverse pressure, the residual expansion stress would play key roles in crack propagation.

V. G. Ukadgaonker works on review on analysis of tube sheets. They analyze for the different types of whole pattern in the tube sheet.

- Equilateral Triangular
- Square
- Staggered Square

They use Analytical technique, Experimental Technique as well as Numerical technique to analyze the tube sheet. And also compare the stress concentration factor for all three patterns of holes.

W.J. O'Donnell has described the method for calculating the stresses and deflection in the perforated plates with a triangular penetration pattern. The method is based partly on theory and partly on experiments. Average ligament stresses are obtained from purely theoretical considerations but effective elastic constants and peak stresses are derived from strain measurement and photo elastic tests. Acceptable limits for pressure stresses and thermal stresses in heat-exchanger tube sheets are also proposed. Effective Elastic Constants for Perforated Plates when a perforated plate is used as a part of a redundant structure, the values used for the effective elastic constants will affect calculated stresses in the remainder of the structure, as well as in the perforated plate itself. For example, the amount of rotation at the periphery of a steam-generator tube sheet depends on the relative rigidity or the tube sheet with respect to the rest of the heat exchanger. If effective elastic constants (particularly E*) which are too low are used in the analysis, the theoretical rotation at the periphery of the tube sheet due to pressure loads across the tube sheet will be greater than the actual rotation. The calculated stresses at the periphery will then be lower than the actual stresses. Correspondingly, if an effective elastic modulus which is too high is used in the analysis, the calculated pressure stresses at the center of the tube sheet would be low.

2010 ASME Codes Section-VIII Division-II ASME is one of the oldest standards- developing organizations in America. It produces approximately 600 codes and standards covering many technical areas, such as fasteners, plumbing fixtures, elevators, pipelines, and power plant systems and components. ASME's standards are developed by committees of subject matter experts using an open, consensus-based process. Many ASME standards are cited by government agencies as tools to meet their regulatory objectives. ASME standards are therefore voluntary, unless the standards have been incorporated into a legally binding business contract or incorporated into regulations enforced by an authority having jurisdiction, such as a federal, state, or local government agency. ASME's standards are used in more than 100 countries and have been translated into numerous languages. The largest ASME standard, both in size and in the number of volunteers involved in its preparation, is the ASME Boiler and Pressure Vessel Code (BPVC). The BPVC provides rules for the design, fabrication, installation, inspection, care, and use of boiler , pressure vessels and nuclear components. The code also includes standards on materials, welding and brazing procedures and qualifications, non-destructive examination, and nuclear in- service inspection.

Kotcherla Sriharshainvestigate the strength analysis of a typical tube to tube sheet joint in shell and tube heat exchanger. In this work the joint between tube and tube sheet joint in shell and tube heat exchanger is designed and analyzed using ANSYS 9.0 software for the combination of admiralty brass and steel as tube and tube sheet materials respectively. Contact analysis is performed to analyze the gap between tube and tube sheet element. Stress concentrations at various parts of tube and tube sheet at various stages of loading and unloading are obtained and displayed pictorially. The value of stress at interface of tube and tube sheet joint is also obtained. The value thus obtained is compared with the result value obtained by executing readily available computer code. Further the Pull out load of tube to tube sheet joint of shell and tube heat exchanger is calculated using mathematical calculations and the obtained value is compared with the pullout load obtained in laboratory. It is found that there is no much difference between the pullout load obtained by mathematical calculation and that of experimental results.

Dr. Enrique Gomez works on ASME section III stress analysis of a heat exchanger tube-sheet with a mis-drilled hole and irregular or thin ligaments**.** A stress analysis is described for a nuclear steam generator tube-sheet with a thin or irregular ligament associated with a misruled hole using the rules of ASME B&PV Section III and Non-Mandatory Appendix A, Article A-8000 for Stresses in Perforated Flat Plates. The analysis demonstrates the proper application of the NB-3200 rules for this special application with discussion of the differences between an actual tube hole deviation from what is assumed in ASME Appendix. They conclude that the presence of misdialed holes or locally thin ligaments do not affect the primary stress margin in the tube-sheet and do not reduce its overall structural integrity. Therefore, there is no consequence of a fatigue crack at any point within the tube-sheet drilling-pattern because it could not propagate beyond the local ligament. Consequently, the presence of misdialed holes within the tube-sheet drilling pattern, although it may complicate tube installation, is a structurally acceptable condition.

3. DESIGN CALCULATIONS

3.1 Parameters for Tube- sheet Calculations

Table 1:-Process parameters

Sr. No.	Parameter Description	Notations	Given Value
1	Internal Pressure	P	0.32 MPa
2	External Pressure	P0	Atmospheric
3	Process Volume	Vp	286 cu m
4	Expected Stagnant Volume	Vs	Not Specified
5	Buffer Volume Requirement	Vb	Not Specified
6	Tube Porosity Volume	Tp	70
7	Tube Length	TL	5.5 m
8	Radius of tube-sheet	R	3 m
9	Tube Diameter	Td	0.15 m

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.

3.2 Mathematical Formulation for unknown parameters

Total volume =Pressure Volume +Expected Stagnant Volume+ Buffer Volume V =Vp +Vs+ Vb

Here buffer volume is not specified hence it has to be considered 0.01 of total volume or stagnant volume whichever is higher.

Also stagnant volume is not specified hence it should be taken as 0.1 of total volume. Therefore ,above equation 1 becomes,

V =Vp +0.1V+0.01V… (2)

V(1-0.1–0.01)= Vp V(0.89)=286x10^9.

V= 3.21348 x$10^{11}$$mm^3$

Buffer Volume= Vb =0.01V =3.213481x$10^9$$mm^3$

Stagnant Volume= Vs =0.1V =3.213481x$10^{10}$$mm^3$ Here, Vs>0.1Vp

Hence, the vessel is characterized as a full process reactionary vessel.ReferringA2209,for full process reactionary vessel, Vp=(0.90NTD) x(πr^2) (NTD is nozzle to nozzle distance in meters).

286 x10^9= 0.90 x NTD x π x (3000^2)

NTD =11239.08 mm

NTD =11.239 m

Now, Vs +Vb=(0.82L1) x(πr^2)

Here, Vs>Vb, hence considering Vb=Vs

2 x3.213481 x10^{10}= (0.82L1) x(πx3000^2)

Gives,L1 = 2772.041 mm

i.e. NTD = 2.772041 m

3.3 Calculations for Tube sheet volume (Tv)

Tv=$\frac{\pi}{4} D^2 \times Tt$

Tv=$\frac{\pi}{4} (5700)^2 \times 1$

Assuming Tt =1 mm

Tv =25517.5863 x10^3 mm^3

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

3.4 Calculation for tube volume

Now, tube diameter (Td)is 0.15 m =150mm. Considering ,

Length of the tube =length of the tube sheet.

$\text{Tv} = \frac{\pi}{4}\text{Td}^2 \times \text{Tt}$

$\text{Tv} = \frac{\pi}{4} D(150)^2 \times 1$ ....$TL=Tt$

TL=Tube length Tv =17671.458 mm^3

3.5 Calculations for 'n' no of holes

Volume of holes =(total volume) x n

Residual volume=Tube sheet volume–Tube volume TR=Tv–Tube volume

But T_R/ Tv=0.3

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

$0.7 = \frac{17671.458 \times n}{25517.58 \times 10^3}$

n = 1010.8 nos≈1012 number of holes.

$0.7 = \frac{17671.458 \times n}{25517.58 \times 10^3}$

n = 1010.8 nos 1012 number of holes various other parameters were calculated .

3.6 Calculations for 'n' no of holes

Volume of holes =(total volume) x n

Residual volume=Tube sheet volume–Tube volume TR=Tv–Tube volume

But T_R/ Tv=0.3

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$0.7 = \frac{17671.458 \times n}{25517.58 \times 10^3}$

n = 1010.8 nos 1012 number of holes.

3.7 Calculations for ligament efficiency

Ligament efficiency (η) = $\frac{D-nD}{D} = \frac{6000-(36\times150)}{6000} = 0.1$

3.8 Calculations for Tube-sheet thickness

Referring ASME section VIII, div-II, page no. 34 $t = d\sqrt{\frac{C.P.}{S.E.}}$

Where,

C = factor considering the method of attachment (0.20 for fillet welding)

d = diameter of the vessel.

p = internal pressure

S =allowable stress

E = Efficiency (summation of ligament and joint efficiency) For, 100 C i.e. SA516 GR70

S = 20.0KSi

S = 137.895 N/ mm^2

$t = 6000\sqrt{\frac{0.2\times0.32}{137.895\times0.1}} =$ 409 mm

3.9 Recalculation of volumes considering tube sheet thickness

Vp=1.1[(Vp'+Tr) +1.2 (Pi* Td * Td) *(Tp/400)]N

Where,

Vp''=Process Volume=286m^3

Tr=Residual Volume ButTr/Tv=0.3

Tr=0.3x17671.458

Tr=5301.43mm^3

Td=150mm (Tube Diameter)

Tp=70 (Tube Porosity Volume)

N=1012 nos.(No of tubes)

Vp =1.1[(286x10^9) +5301.43]+[1.2(πx150^2)x (70/400)]x1012

Vp= (314600.005x10^6)+ (15022153.59) Vp= 3.14615027x$10^{11}$$mm^3$

For full process reactionary vessel,

Vp=(0.90 NTD) x(πr^2)

3.14 x10^{11}= (0.90xNTD)x(πx3000^2)

NTD=12339.420 mm
i.e. NTD = 12.33 m
Now Total volume,
V=Vp+Vb+Vs
Where,Vb=0.01VandVs=0.1V
$V=3.14 \times 10^{11}+0.01V+0.1V$ $V=3.528089\times10^{11} mm^3$
$Vb=0.01V=3528.0898\times10^6 mm^3$
$Vs=0.1V=35280.89\times10^6 mm^3$
Now, $Vs+Vb=(0.82 \times L1)\times(\pi r^2)$ Here, Vs>Vb,
Hence considering Vb=Vs $2Vs=(0.82 \times L_1)\times(\pi \times 3000^2)$
L_1=3043.429mm
L_1 =3.043429 m

3.10 Design of Shell

According to ASME Section-VIII, Division-I, UG27,

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

$$=\frac{0.32\times3000}{137.895-(0.6\times0.32)}$$

Thickness of Shell (tshell) =6.97 mm 7 mm

3.11 Design of Ellipsoidal Head

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

Thickness of ellipsoidal head (t_{head}) $=\frac{P.D.}{2S.E.-0.2P}$

$$=\frac{0.32\times6000}{(2\times137.895)-(0.6\times0.32)}$$

(t_{head}) =6.96mm

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

Thickness of ellipsoidal head (t_{head}) $=\frac{P.D.}{2S.E.-0.2P}$

$$=\frac{0.32\times6000}{(2\times137.895)-(0.6\times0.32)}$$

But according to ASME Section-VIII ,Division-I, the thickness should be taken minimum 9mm.
Find length of Ellipsoidal Head L=?

$\frac{L}{D}= 0.75$

$\frac{L}{6000}= 0.75$

L = 4500 mm

3.12 Design of Nozzle

According to ASME Section-VIII Division-I, UG-36,
Nozzle design parameters,

- Number of nozzles=2
- Diameter of nozzle=0.3m
- Drain nozzle=0.1m
- Head type=ellipsoidal
- Support =saddle support
- No. of support =3
- d_i= internal diameter of nozzle
- d = external diameter of nozzle
- CA =corrosion allowance
- t = actual thickness of shell in mm
- t_r= required thickness as per calculations in mm
- t_n= actual thickness of nozzle in mm
- t_{rn}= required thickness as per calculations in mm.

$t_{rn} = \frac{PiDi}{2\sigma J - Pi}$

$= \frac{0.32 \times 300}{(2 \times 137.85 \times 0.6) - 0.32}$

t_{rn}= 0.5817 mmt_r $= \frac{\text{Pi x Dishell}}{2\sigma J - \text{Pi}}$

$= \frac{0.32 \times 6000}{(2 \times 137.85 \times 0.6) - 0.32}$

t_r=11.63 mm

But as per ASME codes minimum thick thickness should be taken as 12 mm.

h= 2.5(t_r–CA) =2.5(12– 3)

h = 22.5 mm

OR

h= 2.5(t_{rn}–CA) =2.5(6– 3)

h = 22.5 mm

d=d_i+ 2CA

d= 300+ (2 x 3)

d = 306 mm

Now Let, „x" be the nozzle distance till the effect of the persists in mm on each side of the centerline.

$$x = d \text{ or } \frac{di}{2} + t + t_n – 3CA$$

Whichever is Maximum x = 306 mm

or

$$x = \frac{300}{2} + 12 + 12 – (3 \times 3)$$

x = 165 mm

Therefore, taking maximum value as, 306mm

3.13 Area Calculation

Area pertaining to material removed i.e. A=d.t_r =306x 11.62=3555.72 mm^2

Excess area in the shell,

A1= (2x– d) (t– t_r– CA) =[(2x165)-306]x [12 – 11.62-3] A1 = -62.88 mm^2

Excess area in the nozzle, A2=2h1(t_n– t_{rn}–CA) =2x 22.5(12– 0.5817 – 3) A2 =378.8235 mm^2

Excess area in the nozzle inside the shell, A3=2h2 (t_n– 2.CA) =2x 22.5 [12– (2x 3)] A3= 270 mm^2

Therefore required area, A_r=A– (A1+A2+A3) = 3555– (-62.88+378.82+270) A_r=2969 mm^2

3.14:-Design of Reinforcement pad

d_{ip}= internal diameter of RF pad

d_{op}= external diameter of RF pad

t = thickness of RF pad=12 mm d_{ip}=d_i+ 2t_n

d_{ip}=300 + (2 x12)

d_{ip}=324mm.

$$dop = \frac{Ar}{tp} + d_{ip} = \frac{2969}{12} + 324$$

dop= 572 mm

3.15:-Design off Lange:-

Load on projected area =pressure x projected area. P = (0.32 x6000)

P =1920 N.

Internal diameter of flange = 324mm External diameter of flange =600mm thickness off lange=12mm

3.16:-Design Modification after Approval and Testing

- NTD which is currently calculated to be 11239 mm has been updated to 11000 mm, as per site conditions.
- Total length which is currently calculated to be 16905 mm has been updated to 16000 mm, as per site conditions.
- Discussion and Conclusion: -In this chapter all the dimensions of the vessel were calculated, also the dimensions were modified after confirming from the company.

The final modified dimensions are as follows:-

- Thickness of Tube-sheet = 409 mm
- Total length of vessel = 16000 mm 3. NTD=11000mm

3.17 Design Of Saddle

Design of saddle support for horizontal pressure vessel, standard values of saddle is directly taken from Joshi "s process equipment design data book as per requirement of shell diameter of vessel. Following table shows the all standard values of saddle support for 6000 mm. vessel.

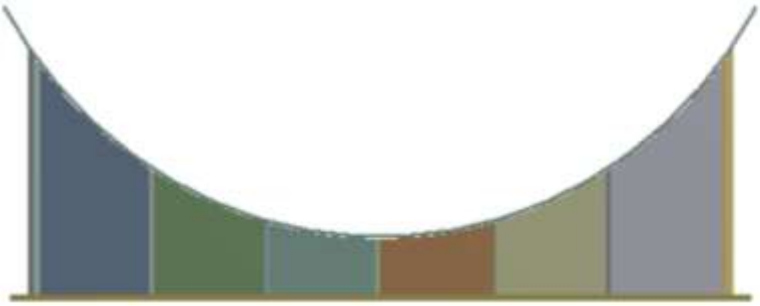

Figure 4: Design of saddle.

Table No 2. Saddle Support Design

Notation	Element Name	Thickness & Width (mm)
A	Thickness of Saddle	23 mm
B	Width of Saddle	500 mm
C	Thickness of Rib	32 mm
D	Width of Rib	230mm
E	Thickness of Rib Plate	32 mm
F	Length of Base Plate	5150 mm
G	Thickness of Base Plate	40 mm

4. CONCLUSION

- The project is basically focused on an Analysis and optimization of space sequential tube-sheet in pressure vessel. Design of pressure vessel are done by ASME Code Section-8,Div-2.
- The Analysis of pressure vessel model was done in ANSYS 15.0 workbench. The results were supported with an experimental validation for verifying the actual deformation and FEA results. Following are concluding remarks based on the analysis performed on vessel.
- Firstly analysis of pressure vessel model is done to develop the standard operating procedure. From the comparison of results at different mesh size.it is concluded that variation in results is within acceptable limit, hence approximately 100000 nodes mesh size is fixed for further analysis.in that maximum stress is 66.837MPa and deformation is 1.6501 mm.
- Result of optimization of Tube-Sheets we found that Thickness is reduced from 215 mm to 45 mm and also Reduced Weight from 13988 kg to 2927.7kg.
- 5-Tube-sheet optimization including point mass weight of 2.5 kg of each tube for reducing weight and material, 45 mm of tube-sheet is finalized and reduced weight up to 21%.
- 6-Stress Result for Complete Vessel Analysis considering all tubes masses, pressure on tube-sheets 0.01MPa and internal Pressure on all other components as 0.32 MPa with 3 saddle supports is 123.18 MPa which is within the limit of allowable stress.
- 7-Above all the conclusion shows the optimization of stress, space, and optimization of tube-sheets is reduced weight up-to 21 % and also reduced material is done.For this condition model is safe as per ASME Section-VIII, Div-II. and FEA results and Experimental results are in close resemblance and proved that FEA analysis is correct and is validated by experimental deformation results Manufactured tested values are within 20% of FEA for large vessel calculations ,hence our FEA results 8.17% are reliable.No leakage and No damage is detected by using Strain Gauge Sensor and ultrasonic testing machine. Finalized vessel satisfies ASME Criteria and this has been validated through FEA.

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