Mechanical Design and Analysis of shell and tube type fixed tube sheet Heat Exchanger

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Major Project

Submitted in partial fulfillment of the requirements

For the degree of

Master of Technology in Mechanical Engineering(Design Engineering)

By

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Declaration

This is to certify that

I) The thesis comprises my original work towards the degree of Master of Technology in Mechanical Engineering(Design Engineering) at Nirma University and has not been submitted elsewhere for Degree.

II) Due Acknowledgment has been made in the text to all other material used.

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Undertaking for Originality of the Work

I, **Rupen S Shah** ,Roll.No.**10MMED15** , give undertaking that the Major Project entitled "**Mechanical Design and Analysis of shell and tube type fixed tube sheet Heat Exchanger**" submitted by me,towards the partial fulfillment of the requirements for the degree of Master of Technology in **Mechanical Engineering(Design Engineering)** of Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made.I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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This is to certify that the Major Project entitled **" Mechanical Design and Analysis of shell and tube type fixed tube sheet Heat Exchanger"** submitted by **Mr. Rupen S. Shah (10MMED15)**,towards the partial fulfillment of the requirements for the degree of **Master of Technology** in Mechanical Engineering **(Design Engineering)** of Nirma University,Ahmedabad is the record of work carried out by him under my supervision and guidance. In my opinion,the submitted work has reached a level required for being accepted for examination.The results embodied in this major project,to the best of my knowledge,haven't been submitted to any other university or institution for award of any degree or diploma.

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Abstract

Heat exchangers are widely used in process industries. As these devices subjected to high pressure and temperature,it is necessary to design their key parts with special care. Under different loading conditions higher stresses will generate at the areas of discontinuity in the geometry. Most cases the area subjected to higher stresses are regions like shell to nozzle, shell to saddle and shell to head junctions. These higher stresses regions may affect the efficiency of the heat exchanger and may lead to dangerous results sometimes.These being a weakest part it is highly recommended to design and analyze them with proper precautions. No literature highlights a standard procedure for such type of crucial parts analysis.

The aim of the present work is to mechanically design and analyze the weak shell to nozzle junction of shell and tube type fixed tube sheet Heat Exchanger. This task is carried out by doing a theoretical calculation and making model in PV Elite.Thickness of shell,head,nozzle etc.and stresses are obtained which is helpful in making 3D model of Shell to nozzle junction and subsequent finite element analysis of shell and tube type fixed tube sheet heat exchanger.Results obtained from theoretical calculations and analysis performed is within the permissible limits and variations are not exceeding more than five percentage.This shows that the adopted procedure gives the satisfactory results.

Keywords:*fixed tube sheet heat exchanger, shell to nozzle junction*

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Nomenclatures

$P_i(P_c)$	Internal(Channel) Design Pressure,kg/cm^2
P_s	Shell Design Pressure,kg/cm^2
D	Inside Diameter of shell,mm
D_i	Inside Diameter of shell-corroded ,mm
E	Joint Efficiency
C	Internal Corrosion Allowance,mm
S	Max. Allowable Stress at Design Temperature,kg/cm^2
t	Nominal thickness,mm
D_0	outside diameter of shell,mm
b_1, b_2	width of the shell and ring,mm
A_1, A_2	c/s area of shell and ring,mm^2
y_1, y_2	distance from neutral axis for shell and ring,mm
A	Flange outside diameter,mm
b	Flange inside diameter,mm
S_b	Bolt Allowable Stress ,kg/cm^2
F_{id}, Fod	Flange Face Inside and Outside Diameter,mm
G_i, G_o	Gasket Inside and Outside Diameter,mm
m	Gasket Factor
y	Gasket Design Seating Stress, kg/cm^2
t_g	Gasket Thickness,mm
w	Flange Face Nubbin Width,mm
l_p	Length of Partition Gasket,mm
m_{part}	Partition Gasket Factor
y_{part}	Partition Gasket Design Seating Stress ,kg/cm^2
c_b	Conversion Factor
N_t	Number of Tubes Holes
V_s, V_t	Poisson's Ratio of Shell and Tube Material
L	Straight Tube Length b/w inner tube sheet faces,mm
p	Tube Pitch ,mm
a_f	Fillet Weld Leg ,mm
a_g	Groove Weld Leg ,mm
f_r	ASME Tube Joint Reliability Factor
h	Thickness of Tube sheet ,mm
h_g	Tube side Pass partition Groove Depth ,mm

Chapter 1

Introduction

1.1 Preliminary Remarks

A heat exchanger is a heat-transfer device that is used for transfer of internal thermal energy between two or more fluids available at different temperatures. In most heat exchangers, the fluids are separated by a heat-transfer surface, and ideally they do not mix. Heat exchangers are used in the process, power, petroleum, air conditioning, refrigeration, heat recovery and other industries. Common examples of heat exchangers familiar to us in day-to-day use are automobile radiators, condensers, evaporators, air pre-heaters and oil coolers.

1.2 Objective

In industries people faces many problems regarding the discontinuous junction in the heat exchanger. Heat exchangers have many discontinuous regions in their structure such as manhole connections, nozzles, supports, joints etc. They are also subjected to different loadings such as internal pressure, external pressure, thermal loads etc. These discontinuous regions will become weakest point and more chance to failure of the whole structure. Thus a reliable and accurate analysis for shell-nozzle junction is necessary.

1.3 Aim and Scope of Work

The aim of the present work is to obtain general solutions for determining the stresses at the nozzle to head junction of a shell and tube type fixed tubesheet heat exchanger. The model is done with the help of PV Elite. The theoretical calculation is done with the help of ASME (American Society of Mechanical Engineer) code. The theoretical calculation is verified with PV Elite calculation.The solid model of the shell to nozzle junction is prepared in PRO-E and finite element analysis is carried out in ANSYS.

1.4 Thesis organization

Chapter 2includes the overview of the literature which has been reviewed during project work.
Chapter 3 covers theoretical calculation and PV Elite calculation of shell and tube type fixed tubesheet heat exchanger
Chapter 4 describes the result obtained by theoretical calculation and it is compared with the PV Elite calculation and it is further analyze in ANSYS.
Chapter 5 explains the conclusion of project and future scopes.

Chapter 2

Literature Review

2.1 Classification of Shell and Tube Heat Exchanger

According to shell and tube, heat exchangers are classified as:[1]

1. Fixed tubesheet heat exchanger
2. U-tube heat exchanger
3. Floating head heat exchanger

2.1.1 Fixed Tube Sheet Heat Exchanger

A fixed tubesheet heat exchanger has straight tubes that are secured at both ends to tubesheets welded to the shell. The construction may have removable channel covers (e.g. AEL)[2], bonnet-type channel covers (e.g. BEM), or integral tubesheets (e.g. NEN) fig2.1. The principal advantage of the fixed tubesheet construction is its low cost of its simple construction. In fact, the fixed tubesheet is the least expensive construction type, as long as no expansion joint is required. Other advantages are that the tubes can be cleaned mechanically after removal of the channel cover or bonnet, and that leakage of the shell side fluid is minimized since there are no flanged joints.

A disadvantage of this design is that since the bundle is fixed to the shell and cannot

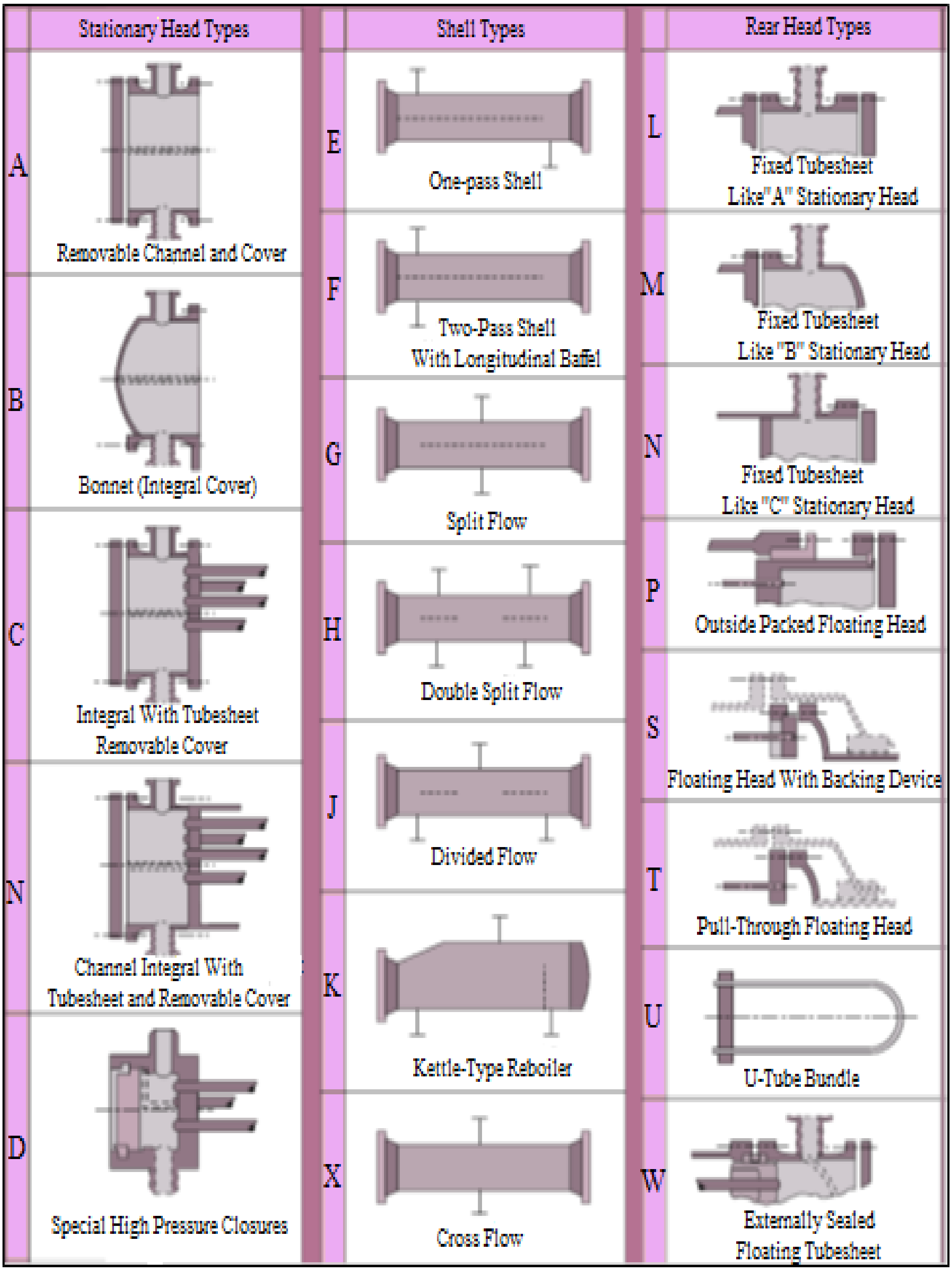

Figure 2.1: TEMA designation for shell and tube heat exchangers

be removed, the outsides of the tubes cannot be cleaned mechanically. Thus, its application is limited to clean services on the shell side. However, if a satisfactory chemical cleaning program can be employed, fixed tube sheet construction may be selected for fouling services on the shell side.

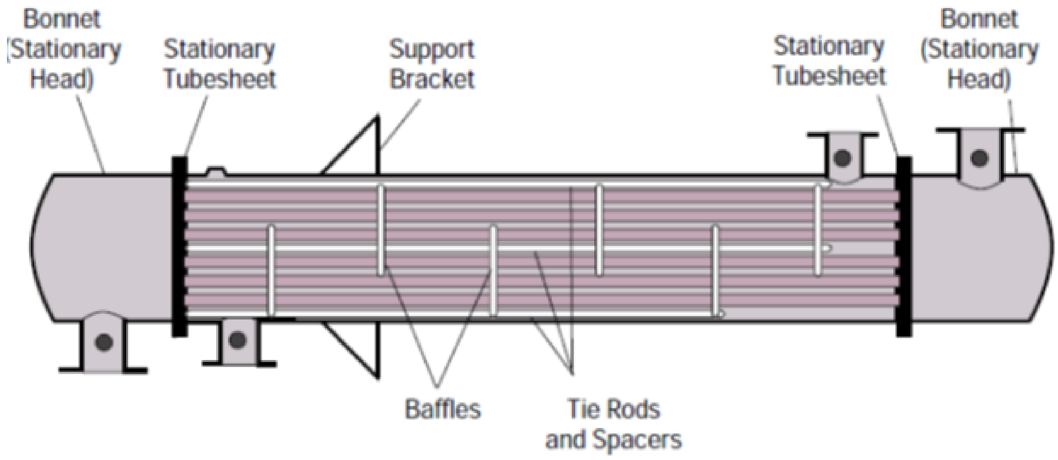

Figure 2.2: fixed tube sheet heat exchanger

In the event of a large differential temperature between the tubes and the shell, the tubesheets will be unable to absorb the differential stress, thereby making it necessary to incorporate an expansion joint. This takes away the advantage of low cost to a significant extent.

2.2 Parts of Heat Exchanger

There are four main parts of heat exchanger shown in figure 2.3.

1. Shell
2. Head
3. Support
4. Nozzle

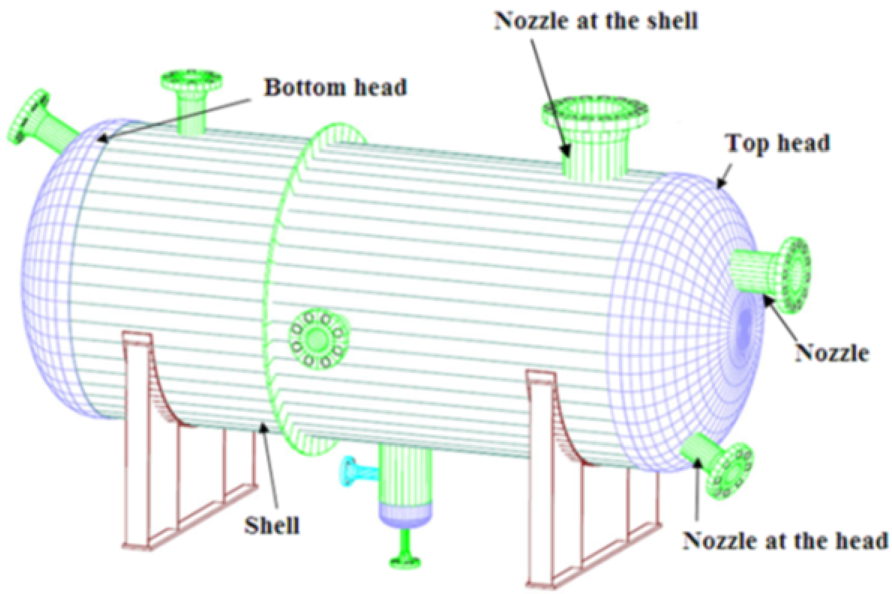

Figure 2.3: parts of heat exchanger

2.2.1 Shell

It is a major component of a heat exchanger. There are two types of shell

1. Cylindrical shell
2. Spherical shell

2.2.2 Head

Head[3] is the end closer of a heat exchanger. Types of heads are shown in figure 2.4.

Types of head:

1. Hemispherical head
2. Torispherical head or Ellipsoidal head
3. Conical head
4. Flat head

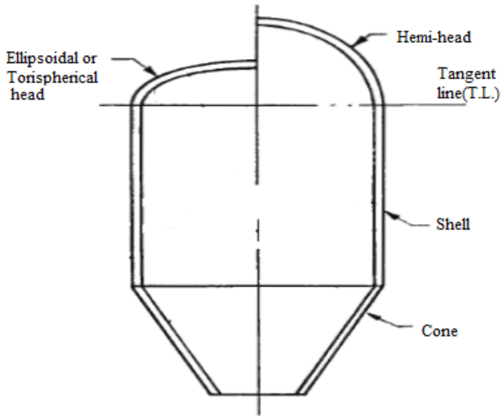

Figure 2.4: Types of head

2.2.3 Supports

Supports are used for vertical or horizontal heat exchangers. Types of supports are shown in figure 2.5.

1. Skirt (straight or conical)
2. Saddles (attached or loose)
3. Rings
4. Lugs

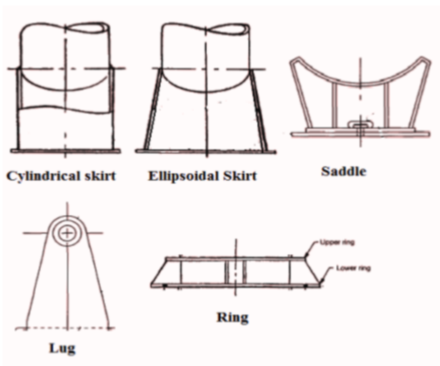

Figure 2.5: Types of support

2.2.4 nozzles

A nozzle is often a pipe or tube of varying cross sectional area and it can be used direct or modified the fluid (liquid or gas). Nozzles are frequently used to control the flow rate, speed, direction, mass, shape and the pressure of the stream that emerges from them. Types of nozzle are shown in figure 2.6.

1. Radial nozzle: A nozzle which is perpendicular to orientation line is called radial nozzle.

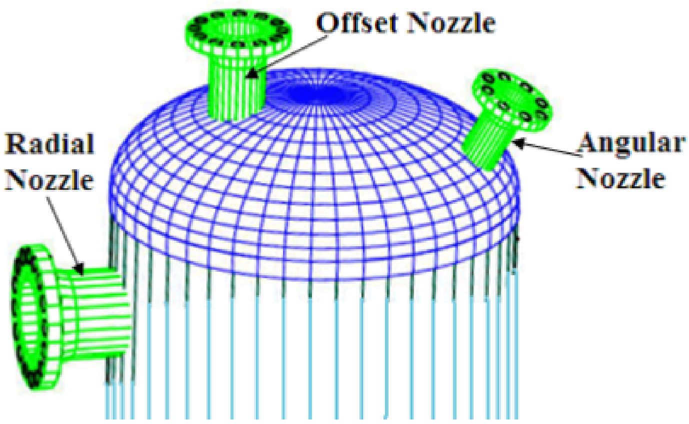

Figure 2.6: Types of nozzle

2. Offset nozzle: A nozzle which has distance from the orientation line is called offset nozzle.

3. Angular nozzle: A nozzle which is at some angle from orientation line is called angular nozzle.

2.3 Basic Components of Shell and Tube Heat Exchanger

While there is an enormous variety of specific design features that can be used in shell and tube heat exchangers, the number of basic components is relatively small.[4]

1. Tubes : The tubes are the basic component of shell and tube heat exchanger, providing the heat transfer surface between one flowing inside the tube and other fluid flowing across the outside of the tubes. The tubes may be seamless or welded and most commonly made of copper or steel alloys.

2. Tube Sheets: The tubes are held in place by being inserted into holes in the tube sheet and there either expanded into grooves cut into the holes or welded to the tube sheet. The tube sheet is usually a single round plate of metal and that has been suitably drilled and grooved to take the tubes, the gaskets, the spacer rods, and bolt circle where it is fastened to the shell.

3. Pass Divider: A pass divider needed in one channel or bonnet for an exchanger having two tube side passes, and they are needed in both channels and bonnets for an exchanger having more than two passes. If channels or bonnets are cast, the dividers are integrally cast and then faced to give a smooth bearing surface on the gasket between the divider and the tube sheet. If the channels are rolled from plate or built up from pipe, the dividers are welded in place.

4. Baffles: Baffles serve two functions: most importantly, they support the tubes in the proper position during assembly and operation and prevent the vibration of the tubes, and secondly, they guide the shell side flow and increasing the velocity and the heat transfer coefficient.

Z.F. Sang, Z.L. Wang, L.P. Xue and G.E.O. Widera [5][6] have discussed plastic limit moment of nozzles in cylindrical vessels with different d/D ratios under out-of-plane moment loading. Three full size test model were designed and fabricated. A 3D nonlinear finite element numerical simulation was also performed. A twice elastic slope plastic moment on the nozzle was obtained approximately by load-displacement and load-strain curves. The results show the plastic loads determined by test and numerical simulation methods are in good agreement. The results can serve as a basis for developing an advanced design guideline by limit analysis for cylindrical vessel with a nozzle under external loads.

V.N.Skopinsky and A.B.Smetankin [7] discussed stress analysis of reinforced nozzle connections in ellipsoidal heads of pressure vessels was carried out using shell theoryand the finite element method. A parametric study of the effects of the reinforcements on the maximum stresses in head-nozzle intersections under internal pressure loading was performed.

Aleksandar petrovic [8] has discussed stress analysis of cylindrical pressure vessel loaded by axial and transverse forces on the free end of a nozzle. The nozzle is placed such that the axis of the nozzle does not cross the axis of the cylindrical shell. The method of finite elements was applied to determine the state of stress in the cylindrical shell. The value obtained for the stress in the nozzle region was used to determine the following: envelope for maximum stress values; maximum values of this envelope; and distance between maximum values on the envelope and outer edge of the nozzle.

C.J. Dekker and H.J. Brink [9][10]have discussed internal pressure stresses at nozzle vessel junction, obtain by various analysis methods on thin shell theory, will be compared in this study. To evaluate an improved shell based analysis method capable of accounting for the effect of the additional material in the weld, an axisymmetric 3D finite element analysis is also included. The main conclusions are that any outward weld area offers little reinforcement and that stress analysis based on thin shell theory is quite acceptable. A close comparison of local load stress calculation methods reveals considerable differences. To investigate we performed many finite element analyses of nozzles on cylinders concentrating not on the shell stresses but also on the stresses in the nozzle wall. Local load stresses were sometimes found to be much higher in the nozzle than in the shell. This led us to formulate a 'modified improved shrink ring method' and to devise multiplication charts for deriving local load nozzle stresses from local load shell stresses. Being important for a proper assessment, pressure induced stresses were investigated too. This resulted in non-dimensional parameter graphs to determine pressure induced stresses at nozzle.

Chapter 3

Theoretical Calculation

3.1 Classification of Stress

3.1.1 Stress Applied

The total elastic stress, which occurs in the vessel shell, is considered to be composed of three different type of stress primary, secondary and peak. In addition primary stress has three specific sub categories. The ASME stress categories and the symbols used to denote them in the code are given below[11]:

1. Primary Stress

   General primary membrane stress, Pm

   Local primary membrane stress, PL

   Primary banding stress, Pb
2. Secondary Stress, Q
3. Peak Stress, F

• **Normal Stress**

The normal stress is the component of the stress normal to the plane of the reference; this is also referred to as a direct stress. Usually the distribution of the stress is not uniform through the thickness of the part, so this stress is considered to be made up in turn of two components one which is uniformly distributed and equal to the average value of stress across the thickness of the section under the consideration, and the other of which varies with the location across the thickness.

• **Shear Stress**

The shear stress is the component of the stress acting in the plane of the reference.

• **Membrane Stress**

The membrane stress is the component of stress that is uniformly distributed and equal to the average value of the stress across the thickness of the section under consideration.

• **Bending Stress**

The bending stress is the component of stress that varies linearly across the thickness of the section under consideration. With this terminology as background, we now can define primary, secondary and peak stress properly.

3.1.2 Stress Categorizations

• **Primary Stresses**

A primary stress is a stress produced by the mechanical loading only and it is so distributed in the structure that no redistribution of load occurs as a result of yielding. The basic characteristic of the stress is that it is not self-limiting. Primary stresses that considerably exceed the yield strength will result in failure, or at least in gross distortion. A thermal stress is not classified as primary stress.

Typical examples of general primary stresses are:

• The average stress due to internal pressure or to distributed live loads;

• The bending stress of a flat cover without supporting moment at the periphery due to internal pressure.

Primary stresses are divided in to the 'general' and 'local' categories.

- **Primary Local Membrane Stress**

Case arise in which the membrane stress produced by pressure or other mechanical loading and associate with a primary together with a discontinuity effect produces excessive distortion in the transfer of load to the other portion of the structure. An example of primary local stress is the membrane stress in a shell produce by external load and moment at a permanent support or at a nozzle connection.

- **Secondary Stresses**

Secondary stresses are stresses developed by constraints due to geometric discontinuities, by the use of the material of the different elastic modulus under external loads, or by constraints due to differential thermal expansion. The basic characteristic of the stress is that it is self-limiting. Examples are the bending stresses at dished end to shell junction, generally thermal stresses.

- **Peak Stresses**

Peak stress is that increment of stress which is additive to the primary to the secondary stresses by reason of local discontinuities or local thermal stress including the effect of stress concentration. The basic characteristic of peak stress is that they do not cause any noticeable distortion and only important to fatigue and brittle fracture in conjunction with primary and secondary stress.

3.2 Theoretical Calculations

As per ASME CODE SECTION VII DIVISION-1 EDITION-2007, ADDENDA 2009[11]

3.2.1 Internal Pressure Calculation (Channel/Tube Side)

Design Inputs as per UG-27

- **LHS/RHS Channel Shell**

Thickness due to internal pressure (t_r):

$$t_r = \frac{P \times R_i}{S \times E - 0.6 \times P} = 8.4471 \quad mm$$

$$t_d = Design\ Thickness = (t_r + c) = 11.4471 \quad mm$$

Max. Allowable Working Pressure at given thickness (MAWP):

$$MAWP = \frac{S \times E \times (t - c)}{R_i + 0.6 \times (t - c)} = 19.1398 \quad kg/cm^2$$

$$MAWP = MAWP - Liquid\ Head = 18.9228 \quad kg/cm^2$$

Max. Allowable Pressure, new and cold (MAPNC):

$$MAPNC = \frac{S \times E \times t}{R + 0.6 \times t} = 22.9928 \quad kg/cm^2$$

Actual Stress at given pressure and thickness (S_a):

$$S_a = \frac{P \times (R_i + 0.6 \times (t - c))}{E \times (t - c)} = 794.6889 \quad kg/cm^2$$

- **RHS CHANNEL DISH END**

K = 1 for 2:1 ellipsoidal dish end

t = Nominal thickness = 12 mm

Thickness due to internal pressure (t_r):

$$t_r = \frac{P \times (D + 2 \times C) \times K}{2 \times S \times E - 0.2 \times P} = 8.4146 \quad mm$$

$$t_d = Design\ Thickness = (t_r + C) = 11.4146 \quad mm$$

Max. Allowable Working Pressure at given thickness (MAWP):

$$MAWP = \frac{2 \times S \times E \times (t - c)}{K \times (D + 2 \times c) + 0.2 \times (t - c)} = 11.5689 \quad kg/cm^2$$

$$MAWP = MAWP - Liquid\ Head = 11.3519 \quad kg/cm^2$$

Max. Allowable Pressure, new and cold (MAPNC):

$$MAPNC = \frac{2 \times S \times E \times t}{K \times D + 0.2 \times t} = 15.4634 \quad kg/cm^2$$

Actual Stress at given pressure and thickness (S_a):

$$S_a = \frac{P \times (K \times (D + 2 \times C) + 0.2 \times (t - C))}{2 \times E \times (t - C)} = 1314.7463 \quad kg/cm^2$$

3.2.2 Internal Pressure Calculation(Shell Side)

Design Inputs as per UG-27

- **MAIN SHELL**

Thickness due to internal pressure (t_r):

$$t_r = \frac{P \times R_i}{S \times E - 0.6 \times P} = 13.7052 \quad mm$$

$$t_d = Design\ Thickness = (t_r + c) = 16.7052 \quad mm$$

Max. Allowable Working Pressure at given thickness (MAWP):

$$MAWP = \frac{S \times E \times (t - C)}{R_i + 0.6 \times (t - C)} = 19.1398 \quad kg/cm^2$$

$$\text{MAWP} = \text{MAWP - Liquid Head} = 19.0398 \quad kg/cm^2$$

Max. Allowable Pressure, new and cold (MAPNC):

$$MAPNC = \frac{S \times E \times t}{R + 0.6 \times t} = 22.9928 \quad kg/cm^2$$

Actual Stress at given pressure and thickness (S_a):

$$S_a = \frac{P \times (R_i + 0.6 \times (t - c))}{E \times (t - c)} = 1285.6667 \quad kg/cm^2$$

3.2.3 Stiffening Ring Calculation for Ring 1 and Ring 2

Design Inputs as per UG-27

$$C.G. = \frac{A_1 \times Y_1 + A_2 \times Y_2}{A_1 + A_2} = 79.101 \quad mm$$

$$I_{yy1} = \frac{b_1 \times {d_1}^3}{12} = 56404.7438 \quad mm^4$$

$$I_{yy2} = \frac{b_2 \times {d_2}^3}{12} = 20000000 \quad mm^4$$

$$h_1 = ABS \;\; (C.G - y_1) = 71.601 \quad mm$$

$$h_2 = ABS \;\; (C.G - y_2) = 35.899 \quad mm$$

Available moment of inertia, Ring plus Shell for both Ring 1 and Ring 2:

$$I = I_{yy1} + A_1 \times {h_1}^2 + I_{yy2} + A_2 \times {h_2}^2 = 43211254.23 \quad mm^4$$

Required moment of inertia, Ring plus Shell for Ring 1:

$$I_{s'} = {D_0}^2 \times L_s \times \left(t + \frac{A_s}{L_s}\right) \times \frac{A}{10.9} = 2780000 \quad mm^4$$

Required moment of inertia, Ring plus Shell for Ring 1:

$$I_{s'} = {D_0}^2 \times L_s \times \left(t + \frac{A_s}{L_s}\right) \times \frac{A}{10.9} = 2807095 \quad mm^4$$

3.2.4 Nozzle Calculation

Design Inputs per UG-37

Required thickness of Cylindrical Shell(t_r):

$$t_r = \frac{P \times (R_i + C)}{S \times E - 0.6 \times P} = 8.2769 \quad mm$$

Required thickness of Nozzle Wall (T_{rn}):

$$t_{rn} = \frac{P \times R_n}{S \times E + 0.4 \times P} = 2.2547 \quad mm$$

Finished diameter of circular opening(d):

$$d = D_n - 2 \times t_n + 2 \times C_n = 570 \quad mm$$

$$f_{r1} = MIN\left(\frac{S_n}{S_v}, 1\right) = 1$$

$$f_{r2} = MIN\left(\frac{S_n}{S_v}, 1\right) = 1$$

$$f_{r3} = MIN\left(\frac{S_n}{S_v}, \frac{S_p}{S_v}, 1\right) = 1$$

$$f_{r4} = MIN\left(\frac{S_p}{S_v}, 1\right) = 1$$

Reinforcement Area Required for Nozzle (A_r):

$$A_r = d \times t_r \times F + 2 \times t_{nc} \times t_r \times F \times (1 - f_{r1)} = 4717.833 \quad mm^2$$

Area Available in Shell (A_1):

$$A_1{}^0 = d \times (E \times t_{sc} - F \times t_r) - 2 \times t_{nc} \times (E \times t_{sc} - F \times t_r) \times (1 - f_{r1)}$$
$$= 3832.167 \quad mm^2$$
$$A_1{}^1 = 2 \times (t_{sc} + t_{nc}) \times (E \times t_{sc} - F \times t_r) - 2 \times t_{nc} \times (E \times t_{sc} - F \times t_r) \times (1 - f_{r1})$$
$$= 403.386 \quad mm^2$$
$$A_1 = MAX\left(A_1{}^0, A_1{}^1\right) = 3832.167 \quad mm^2$$

Area Available in Nozzle Wall (A_2):

$$A_2{}^0 = 5 \times (t_{nc} - t_{rn}) \times f_{r2} \times t_{sc} = 955.8975 \quad mm^2$$

$$A_2{}^1 = 2 \times (t_{nc} - t_{rn}) \times (2.5 \times t_{nc} + t_p) \times f_{r2} = 1414.7283 \quad mm^2$$

$$A_2 = MIN\left(A_2{}^0, A_2{}^1\right) = 955.8975 \quad mm^2$$

Area Available in Inward Nozzle (A_3):

$$A_3 = 0$$

Area Available in Welds ($A4$):

$$A_{41} = W_0{}^2 \times f_{r3} = 90.7256 \quad mm^2$$

$$A_{42} = W_p{}^2 \times f_{r4} = 144 \quad mm^2$$

$$A_{43} = W_i^2 \times f_{r2} = 0$$

$$A_4 = A_{41} + A_{42} + A_{43} = 234.7256 \quad mm^2$$

Area Available in Element(A_5):

$$A_5 = (MIN\,(D_p, D_L) - (Nozzle\ OD)) \times MIN\,(t_p, t_{lwp}) \times f_{r4} = 9720 \quad mm^2$$

Now,

$A_1 + A_2 + A_3 + A_4 + A_5 \quad = \quad 14742.7901 \quad mm^2$

$A_1 + A_2 + A_3 + A_4 + A_5 > A_r - - - - - - - - > 14742.7901 > 4717.833$

Therefore, Opening is Adequately Reinforced

Minimum Nozzle Neck Thickness Requirement

Wall Thickness per UG-45 (a)

$$t_{ra} = t_{rn} + c_n = 5.2547 \quad mm$$

Wall Thickness per UG-16 (b)

$$t_{r16b} = \frac{1}{16} in + C_n = 4.5875 \quad mm$$

Wall Thickness per UG-45 (b) (1)

$$t_{rb1} = t_r + C = 11.2769 \quad mm$$

Wall Thickness per UG-45 (b) (2)

$$t_{rb2} = t_r \left(Ext.\ pre\right) + C = 3 \quad mm$$

Wall Thickness per UG-45 (b) (3)

$$t_{rb3} = MAX \ \left(t_{rb1}, t_{rb2}, t_{r16b}\right) = 11.2769 \quad mm$$

Std. Wall Pipe per UG-45 (b)(4)

$$t_{rb4} = 8.34 + C_n = 11.34 \quad mm$$

Wall Thickness per UG-45 (b)

$$t_{rb} = MIN\,(t_{rb3}, t_{rb4}) = 11.2769 \quad mm$$

Final Required Thickness

$$t_{UG-45} = MAX(t_{ra}, t_{rb}) = 11.2769 \quad mm$$

Available Nozzle Neck Thickness = 18 mm

Available Nozzle Neck Thickness > Required Thickness

3.2.5 Blind Flange Calculation

• **LHS Channel Cover Flange**

Design Inputs as per Mandatory Appendix-2

Gasket Contact Width (N):

$$N = \frac{G_0 - G_i}{2} = 16 \quad mm$$

Basic Gasket Width (b_0):

$$b_0 = N/2 = 8 \quad mm$$

Effective Gasket Width (b):

$$b = C_b \times \sqrt{b_0} = 7.0711 \quad mm$$

Gasket Reaction Diameter (G):

$$G = G_0 - 2 \times b = 2248.858 \quad mm$$

- **Basic Flange and Bolt Loads**

Hydrostatic End Load due to Pressure (H):

$$H = 0.785 \times G^2 \times P = 420823.0425 \quad kg$$

Contact Load on Gasket Surfaces (H_p):

$$H_p = 2 \times b \times \pi \times G \times m \times P = 31756.7272 \quad kg$$

Operating Bolt Load (W_{m1}):

$$W_{m1} = H + H_p = 452579.7697 \quad kg$$

Gasket Seating Bolt Load (W_{m2}):

$$W_{m2} = \pi \times b \times G \times y = 351056.6378 \quad kg$$

Required Bolt Area (A_m):

$$A_m = MAX\left(\frac{W_{m1}}{S_b}, \frac{W_{m2}}{S_a}\right) = 257.487 \quad cm^2$$

Minimum Gasket Contact Width (N_{min}):

$$N_{min} = \frac{A_b \times S_a}{y \times \pi \times (G_0 + G_i)} = 4.8829 \quad mm$$

Flange Design Bolt Load for Gasket Seating (w):

$$W = S_a \times \left(\frac{A_m + A_b}{2}\right) = 468508.7252 \quad kg$$

Gasket Seating Force (H_G):

$$H_G = W_{m1} = 452579.77 \quad kg$$

Distance to Gasket Load Reaction (h_g):

$$h_g = (C - G)/2 = 33.0711 \quad mm$$

Tangential Flange Stress, Flat Head, Operating (S_{to}):

$$S_{to} = \frac{1.9 \times W_{m1} \times h_g \times B_{cor}}{t^2 \times G} + \frac{C \times Z \times P \times G^2}{t^2} = 705.8925 \quad kg/cm^2$$

Tangential Flange Stress, Flat Head, Seating (S_{ta}):

$$S_{ta} = \frac{1.9 \times W \times h_g \times B_{cor}}{t^2 \times G} = 57.4122 \quad kg/cm^2$$

3.2.6 Integral Weld Neck Flange Calculation

- **LHS Channel Cover Flange**

Design Inputs as per Mandatory Appendix-2

Hub Small End Required Thickness due to Internal Pressure(t_r):

$$t_r = \frac{P \times (R + c)}{S \times E - 0.6 \times P} + c = 11.2769 \quad mm$$

Corroded Flange ID:

$$B_{cor} = B + 2 \times F_{cor} = 2186 \quad mm$$

Corroded Large Hub:

$$g_{1cor} = g_1 - F_{cor} = 20 \quad mm$$

Corroded Small Hub:

$$g_{0cor} = g_0 - F_{cor} = 15 \quad mm$$

Code R Dimension:

$$R = \frac{C - B_{cor}}{2} - g_{1cor} = 44.5 \quad mm$$

- **Basic Flange and Bolt Loads**

Hydrostatic End Load due to Pressure (H):

$$H = 0.785 \times G^2 \times P = 420823.0425 \quad kg$$

Contact Load on Gasket Surfaces (H_p):

$$H_p = (2 \times b \times \pi \times G \times m \times P) + (2 \times l_p \times b_{part} \times m_{part} \times P) = 40437.3976 \quad kg$$

Hydrostatic End Load at Flange ID (H_d):

$$H_d = \frac{\pi \times {B_{cor}}^2 \times P}{4} = 397816.9224 \quad kg$$

Pressure Force on Flange Face (H_t):

$$H_t = H - H_d = 23006.1201 \quad kg$$

Operating Bolt Load (W_{m1}):

$$W_{m1} = H + H_p = 461260.4401 \quad kg$$

Gasket Seating Bolt Load (W_{m2}):

$$W_{m2} = (\pi \times b \times G \times y) + (y_{part} \times b_{part} \times l_p) = 420112.34 \quad kg$$

Required Bolt Area (A_m):

$$A_m = MAX\left(\frac{W_{m1}}{S_b}, \frac{W_{m2}}{S_a}\right) = 262.4257 \quad cm^2$$

Minimum Gasket Contact Width (N_{min}):

$$N_{min} = \frac{A_b \times S_a}{y \times \pi \times (G_0 + G_i)} = 4.8806 \quad mm$$

Flange Design Bolt Load for Gasket Seating (w):

$$W = \left(\frac{A_m + A_b}{2}\right) \times S_a = 472849.0523 \quad kg$$

Gasket Seating Force (H_G):

$$H_G = W_{m1} - H = 40437.3976 \quad kg$$

Distance to Gasket Load Reaction (h_g):

$$h_g = (C - G)/2 = 33.0711 \quad mm$$

Distance to Face Pressure Reaction (h_t):

$$h_t = \frac{R + g_{1cor} + h_g}{2} = 48.7856 \quad mm$$

Distance to End Pressure Reaction (h_d):

$$h_d = R + \frac{g_{1cor}}{2} = 54.5 \quad mm$$

End Pressure Moment (M_d):

$$M_d = H_d \times h_d = 21681.0223 \quad kg * m$$

Face Pressure Moment (M_t):

$$M_t = H_t \times h_t = 1122.3674 \quad kg * m$$

Gasket Load Moment (M_g):

$$M_g = H_g \times h_g = 1337.3092 \quad kg * m$$

Gasket Seating Moment (M_a):

$$M_a = W \times h_g = 15637.6383 \quad kg * m$$

Total Moment for Operation (M_{op}):

$$M_{op} = M_d + M_t + M_g = 24140.6989 \quad kg * m$$

Total Moment of Gasket Seating (M_a):

$$M_a = 15637.6383 \quad kg * m$$

Effective Hub Length (h_0):

$$h_0 = \sqrt{B_{cor} \times g_{0cor}} = 181.0801 \quad mm$$

Hub Ratio:

$$\frac{h}{h_0} = \frac{H_L}{H_0} = 0.1657$$

Thickness Ratio:

$$\frac{g_1}{g_0} = \frac{g_{1cor}}{g_{0cor}} = 1.2778$$

Flange Factors for Integral Flange:

Factor F per 2-7.2: F =0.9

Factor V per 2-7.3: V = 0.458

Factor f per 2-7.6: f = 1.262

Factors from Figure 2-7.1

K = 1.088 T = 1.881 U = 25.115 Y = 22.855 Z = 11.848

$$d = \frac{U \times h_0 \times {g_{0cor}}^2}{V} = 2234194.345 \quad mm^3$$

$$e = \frac{F}{h_0} = 0.005 \quad mm^{-1}$$

$$L = \frac{(t \times e) + 1}{T} + \frac{t^3}{d} = 2.5757$$

Stress Factors:

$\alpha = 1.765$ $\beta = 2.021$ $\gamma = 0.938$ $\delta = 1.636$ $\lambda = 2.574$

Longitudinal Hub Stress, Operating (S_{Ho}):

$$S_{H0} = \frac{f \times M_{op}}{B_{cor} \times L \times {g_{1cor}}^2} = 1352.7071 \quad kg/cm^2$$

Longitudinal Hub Stress, Seating (S_{Ha}):

$$S_{Ha} = \frac{f \times M_a}{B_{cor} \times L \times {g_{1cor}}^2} = 876.2441 \quad kg/cm^2$$

Radial Flange Stress, Operating (S_{Ro}):

$$S_{Ro} = \frac{\beta \times M_{op}}{B_{cor} \times L \times t^2} = 36.5367 \quad kg/cm^2$$

Radial Flange Stress, Seating (S_{Ra}):

$$S_{Ra} = \frac{\beta \times M_a}{B_{cor} \times L \times t^2} = 23.6674 \quad kg/cm^2$$

Tangential Flange Stress, Operating (S_{to}):

$$S_{to} = \frac{Y \times M_{op}}{t^2 \times B_{cor}} - Z \times S_{Ro} = 631.3529 \quad kg/cm^2$$

Tangential Flange Stress, Seating (S_{ta}):

$$S_{ta} = \frac{Y \times M_a}{t^2 \times B_{cor}} - Z \times S_{Ra} = 408.972 \quad kg/cm^2$$

Average Flange Stress, Operating (S_{Ao}):

$$S_{Ao} = \frac{S_{H0} + MAX\left(S_{R0}, S_{H0}\right)}{2} = 992.03 \quad kg/cm^2$$

Average Flange Stress, Seating (S_{Aa}):

$$S_{Aa} = \frac{S_{Ha} + MAX\left(S_{Ra}, S_{Ha}\right)}{2} = 642.6081 \quad kg/cm^2$$

Bolt Stress, Operating (B_{So}):

$$B_{So} = \frac{W_{m1}}{A_b} = 1673.5862 \quad kg/cm^2$$

Bolt Stress, Seating (B_{Sa}):

$$B_{Sa} = \frac{W_{m2}}{A_b} = 1524.289 \quad kg/cm^2$$

3.2.7 Tube Sheet Calculation

Tube Required Thickness under Internal Pressure (Tube side Pressure)(t_r):

$$t_r = \frac{P_c \times (R_t - C_t)}{S_t \times E + 0.4 \times P_c} = 0.0957 \quad mm$$

Tube Required Thickness under External Pressure (Shell side Pressure):

$$EMAP = \frac{4 \times B}{3 \times \frac{D}{t}} = 166.8613 \quad kg/cm^2$$

- **Results for ASME Fixed Tubesheet Calculation:**

- **UHX-13.5.1 Step 1:**

Compute the Tube Expansion Depth Ratio (ρ):

$$\rho = \frac{l_{tx}}{h} = 0.8$$

Compute the Effective Tube Hole Diameter (d^*):

$$d^* = MAX\left(D_t - 2 \times t_t \times \frac{E_{tT}}{E} \times \frac{S_{tT}}{S} \times \rho, D_t - 2 \times t_t\right) = 17.488 \quad mm$$

Compute The Equivalent Outer Tube Limit Circle Diameter (D_0):

$$D_0 = 2 \times R_0 + D_t = 2179.1 \quad mm$$

Determine The Basic Ligament Efficiency for Shear (μ):

$$\mu = \frac{p - D_t}{p} = 0.2$$

Compute The Equivalent Outer Tube Limit Radius (a_0):

$$a_0 = \frac{D_0}{2} = 1089.55 \quad mm$$

Compute the Effective Tube Pitch (p^*):

$$p^* = \frac{P}{\left(1 - \frac{4 \times MIN\ [A_L, 4 \times D_0 \times p]}{\pi \times D_0{}^2}\right)^{0.5}} = 25.7642 \quad mm$$

Compute the Effective Ligament Efficiency for Bending (μ^*):

$$\mu^* = \frac{p^* - d^*}{p^*} = 0.3212$$

Compute the Ratios (ρ_s):

$$a_s = \frac{D_s}{2} = 1090 \quad mm^2$$

$$\rho_s = \frac{a_s}{a_0} = 1.0004$$

Compute the Ratios (ρ_c):

$$a_c = \frac{D_c}{2} = 1124.3753 \quad mm^2$$

$$\rho_c = \frac{a_c}{a_0} = 1.032$$

Compute Parameter (x_t):

$$X_t = 1 - N_t \times \left(\frac{D_t - 2 \times \ t_t}{2 \times a_0}\right)^2 = 0.6592$$

Compute Parameter (x_s):

$$X_s = 1 - N_t \times \left(\frac{D_t}{2 \times a_0}\right)^2 = 0.4675$$

- **UHX-13.5.2 Step 2:**

Determine the Axial Shell Stiffness (K_s):

$$K_s = \frac{\pi \times t_s \times (D_s + t_s) \times E_s}{L} = 27918713.72 \quad kg/cm^2 * mm$$

Determine the Axial Tube Stiffness (K_t):

$$K_t = \frac{\pi \times t_t \times (D_t + t_t) \times E_t}{L} = 25403.7431 \quad kg/cm^2 * mm$$

Compute Stiffness Factor (K_{st}):

$$K_{st} = \frac{K_s}{N_t \times K_t} = 0.1739$$

Compute Factor (J):

$$J = \frac{1}{1 + \frac{K_s}{K_j}} = 1 \ For No Expansion Joint$$

Compute Shell Coefficient (β_s):

$$\beta_s = \frac{\sqrt[4]{12 \times \left(1 - {V_s}^2\right)}}{\sqrt{(D_s + t_s) \times t_s}} = 0.0091$$

Determine Shell Coefficient (K_s):

$$K_s = \frac{\beta_s \times E_s \times {t_s}^3}{6 \times \left(1 - {V_s}^2\right)} = 19312668 \quad kg/cm^2 * mm^2$$

Determine Shell Coefficient (λ_s):

$$\lambda_s = \frac{6 \times D_s \times K_s}{h^3} \times \left(1 + h \times \beta_s + \frac{h^2 \times {\beta_s}^2}{2}\right) = 1146902.291 \quad kg/cm^2$$

Compute Shell Coefficient (δ_s):

$$\delta_s = \frac{{D_s}^2}{4 \times E_s \times t_s} \times \left(1 - \frac{V_s}{2}\right) = 0.0282 \quad mm/kg/cm^2$$

- **UHX-13.5.3 Step 3:**

Looking up E*/E and v* from Table UHX-11.2 using h/p = 3.000

$E^*/E = 0.293828$

$E^* = 583802 \ kg/cm^2$

$v^* = 0.347485$

Compute the Tube Bundle Stiffness Factor (X_a):

$$X_a = \left(24 \times \left(1 - V^{*2}\right) \times N_t \times \frac{E_t \times t_t \times (D_t - t_t) \times {a_0}^2}{E^* \times L \times h^3}\right)^{0.25} = 8.4917$$

Values from Table UHX-13.1

$Z_d = 0.002421$ $Z_v = 0.014069$ $Z_m = 0.169999$

- **UHX-13.5.4 Step 4:**

Compute Diameter Ratio (K):

$$K = A/D_0 = 1.0917$$

Compute Coefficient (F):

$$F = \frac{1 - V^*}{E^*} \times (\lambda_s + \lambda_c + E \times \ln(K)\ ) = 1.4767$$

Calculate Parameter (φ):

$$\varphi = (1 + V^*) \times F = 1.9898$$

Compute Parameter (Q_1)]:

$$Q_1 = \frac{\rho_s - 1 - \ \times Z_v}{1 + \varphi \times Z_m} = 0.0206$$

Compute Parameter (Q_{z1}):

$$Q_{z1} = \frac{(Z_d + (Q_1 \times Z_v)) \times {X_a}^4}{2} = 5.5407$$

Compute Parameter (Q_{z2}):

$$Q_{z2} = \frac{(Z_v + (Q_1 \times Z_m)) \times {X_a}^4}{2} = 27.4727$$

Compute Parameter (U):

$$U = \frac{[Z_v + (\rho_s - 1) \times Z_m] \times {X_a}^4}{1 + (\varphi \times Z_m)} = 54.928$$

- **UHX-13.5.5 Step 5:**

Determine Factor (γ_b):

$$\gamma_b = \frac{G_c - C}{D_0} = -0.0304$$

Compute Parameter (γ):

$$\gamma = [\alpha_{t,m} \times (T_{t,m} - T_a) - \alpha_{s,m} \times (T_{s,m} - T_a)] \times L = 0$$

Calculate Parameter (ω_s):

$$\omega_s = \quad \rho_s \times K_s \times \beta_s \times \delta_s \times (1 + h \times \beta_s) = 8341.8338 \quad mm^2$$

Calculate Parameter (${\omega_s}^*$):

$${\omega_s}^* = {a_0}^2 \times \frac{({\rho_s}^2 - 1) \times (\rho_s - 1)}{4} - \omega_s = -8341.7388 \quad mm^2$$

Calculate Parameter (ω_c):

$$\omega_c = \quad \rho_c \times K_c \times \beta_c \times \delta_c \times (1 + h \times \beta_c) = 0$$

Calculate Parameter (${\omega_c}^*$):

$${\omega_c}^* = {a_0}^2 \times \left[\frac{({\rho_s}^2 + 1) \times (\rho_c - 1)}{4} - \frac{(\rho_s - 1)}{2}\right] - \omega_c = 19374.0133 \quad mm^2$$

- **UHX-13.5.6 Step 6:**

Compute the Pressure ($P_{s'}$):

$$P_{s'} = \left(x_s + 2 \times (1 - x_s) \times v_t + \frac{2}{K_{s,t}} \times \left(\frac{D_s}{D_0}\right)^2 \times v_s - \frac{{\rho_s}^2 - 1}{J \times K_{s,t}} - \frac{(1-J)}{2 \times J \times K_{s,t}} \times \frac{\left[{D_j}^2 - (2 \times a_s)^2\right]}{{D_0}^2} \right) \times P_s$$

$$= \quad 73.6978 \quad kg/cm^2$$

Compute the Pressure ($P_{t'}$):

$$P_{t'} = \left(x_t + 2 \times (1 - x_t) \times v_t + \frac{1}{J \times K_{s,t}} \right) \times P_c = 70.1096 \quad kg/cm^2$$

Compute the Pressure (P_γ):

$$P_\gamma = \frac{N_t \times K_t \times \gamma}{\pi \times {a_0}^2} = 0$$

Compute the Pressure (P_W):

$$P_W = -\frac{U}{{a_0}^2} \times \frac{\gamma_b}{2 \times \pi} \times W = 10.4985 \quad kg/cm^2$$

Calculate the Pressure (P_{rim}):

$$P_{rim} = -\frac{U}{{a_0}^2} \times ({\omega_s}^* * P_s - {\omega_c}^* * P_c) = 16.2181 \quad kg/cm^2$$

Calculate Effective Pressure (P_e):

$$P_e = \frac{J \times K_{s,t} \times (P_s^{'} - P_t^{'} + P_\gamma + P_w + P_{rim})}{1 + J \times K_{s,t} \times [Q_{z1} + (\rho_s - 1) \times Q_{z2}]} = 2.6813 \quad kg/cm^2$$

- **UHX-13.5.7 Step 7:**

Determine Factor (Q_2):

$$Q_2 = \frac{({\omega_s}^* \times P_s - {\omega_c}^* \times P_c) + \frac{\gamma_b}{2 \times \pi} \times W}{1 + \varphi \times Z_m} = -4314.6062 \quad kg$$

Calculate Factor (Q_3):

$$Q_3 = Q_1 + \frac{2 \times Q_2}{P_e \times {a_0}^2} = -0.2917 \quad kg$$

F_m value from Table UHX-13.1

$$F_m = 0.145881$$

The Tubesheet Bending Stress-Original Thickness (σ):

$$\sigma = \left(\frac{1.5 \times F_m}{\mu^*}\right) \times \left(\frac{2 \times a_0}{h - h'_g}\right)^2 \times P_e = 1821.864 \quad kg/cm^2$$

The Allowable Tubesheet Bending Stress ($\sigma_{allowed}$):

$$\sigma_{alloewd} = 1.5 \times S = 2109.21 \quad kg/cm^2$$

- **UHX-13.5.8 Step 8:**

The Tubesheet Average Shear Stress - Original Thickness (τ):

$$\tau = \left(\frac{1}{2 \times \mu}\right) \times \left(\frac{a_0}{h}\right) \times P_e = 97.3803 \quad kg/cm^2$$

The Allowable Tubesheet Shear Stress ($\tau_{allowed}$):

$$\tau_{allowed} = 0.8 \times S = 1124.912 \quad kg/cm^2$$

- **UHX-13.5.9 Step 9:**

Determine Coefficient (F_q):

$$F_q = (Z_d + (Q_3 \times Z_v)) \times \left(\frac{{X_a}^4}{2}\right) = -4.3754$$

The Tube Stress in the Outermost Tube Row ($\sigma_{t,o}$):

$$\sigma_{t,o} = \frac{(P_s \times X_s - P_t \times X_t) - (P_e \times F_q)}{X_t - X_s} = 67.1817 \quad kg/cm^2$$

The Allowable Tube Stress ($\sigma_{t,a}$):

$$\sigma_{t,a} = S_t = 1103.82 \quad kg/cm^2$$

Tube Weld Size Results per UW-20:

Tube Strength (F_t):

$$F_t = \pi \times t_t \times (d_t - t_t) \times S_t = 1247.758 \quad kg$$

Design Strength Ratio (F_d):

$$F_d = 1$$

Weld Strength Factor (F_w):

$$F_w = \frac{S_a}{S_w} = 1$$

Minimum Weld Length (a_r):

$$a_r = 2 \times \left[\sqrt{(0.75 \times D_t)^2 + 1.07 \times t_t \times (D_t - t_t) \times F_w \times F_d} - 0.75 \times D_t \right] = 2.4666 \quad mm$$

- **UHX-13.5.10 Step 10:**

The Shell Membrane Stress Due to Joint Interaction ($\sigma_{s,m}$):

$$\sigma_{s,m} = \frac{{a_0}^2}{t_s \times (2 \times a_s + t_s)} \times \left[P_e + \left({\rho_s}^2 - 1\right) \times (P_s - P_t)\right] + \frac{{a_s}^2}{t_s \times (2 \times a_s + t_s)} \times P_t$$

$$= 398.9319 \quad kg/cm^2$$

The Shell Bending Stress Due to Joint Interaction ($\sigma_{s,b}$):

$$A = \frac{6}{{t_s}^2} \times K_s$$

$$B = \beta_s \times \left[\delta_s \times P_s - V_s \times \frac{a_s}{E_s} \times \sigma_{s,m} \right]$$

$$C = \frac{6 \times (1 - V^{*2})}{E^*} \times \left(\frac{{a_0}^3}{h^3} \right) \times \left(1 + \frac{h \times \beta_s}{2} \right) \times \left[P_e \times (Z_v + Z_m \times Q_1) + \frac{2}{{a_0}^2} \times Z_m \times Q_2 \right]$$

$$\sigma_{s,b} = A \times (B + C) = 117.558 \quad kg/cm^2$$

3.2.8 PV Elite Model

Client gives the specification (i.e. design data) of the shell and tube type fixed tube sheet heat exchanger to the company as shown in fig3.1

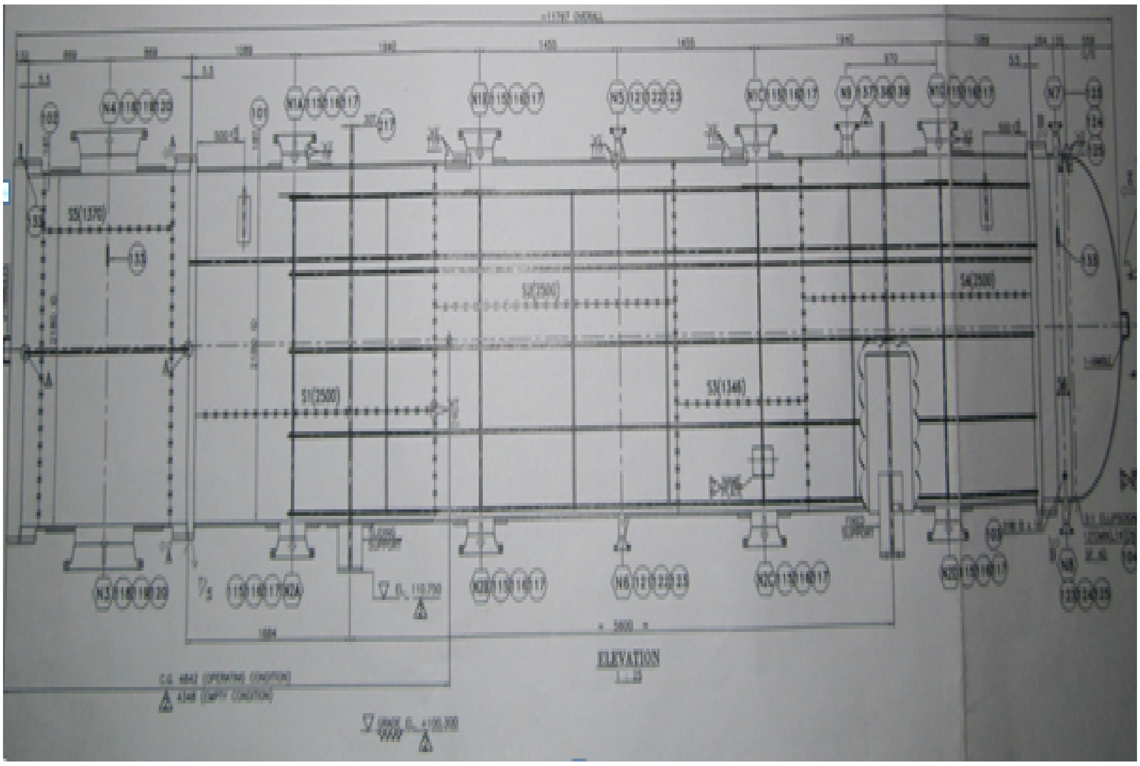

Figure 3.1: Data Sheet

According to the specification and with the help of ASME code and TEMA standard, i prepared the PV Elite model of the shell and tube type fixed tube sheet heat exchanger as shown in fig 3.2.

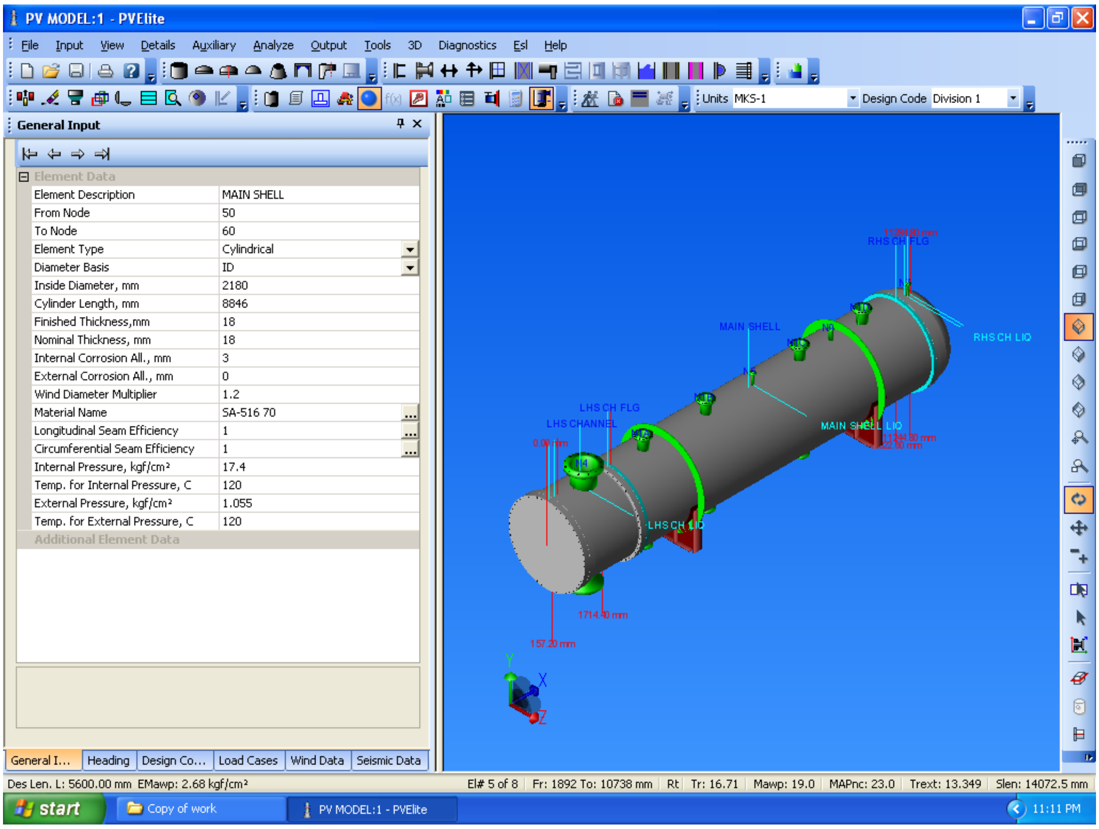

Figure 3.2: PV Elite model

PV Elite Calculations

For fixed tube sheet heat exchanger,internal pressure calculations, nozzle calculation, flange calculation and tubesheet calculations are shown in respectively table 3.1 to 3.4.

Table 3.1: Internal Pressure Calculation

	Thickness mm	MAWP kg/cm^2	MAPNC kg/cm^2	Stress kg/cm^2
Tube side-LHS/RHS channel	11.4472	18.923	22.993	794.699
Tube side-RHS dish end	11.4147	11.352	15.463	1314.764
Shell side	16.7052	19.040	22.993	1285.670

Table 3.2: Nozzle Calculation

	Thickness-cylindrical shell,mm	Thickness-nozzle wall,mm
Nozzle	8.2769	2.2547

Table 3.3: Flange calculation

	Tangential stress, operating, kg/cm^2	Tangential stress, seating, kg/cm^2	Longitudinal hub stress, operating, kg/cm^2	Longitudinal hub stress, seating, kg/cm^2	Radial flange stress, operating, kg/cm^2	Radial flange stress, seating, kg/cm^2
Integral weld neck flange	644.67	413.86	1382.35	887.44	37.33	23.96
Blind flange	776.44	58.11	-	-	-	-

Table 3.4: Tube sheet Calculation

	Shell membrane stress kg/cm^2	Shell bending stress kg/cm^2
Tube sheet	399.0044	123.3082

Chapter 4

Results and Discussion

4.1 Comparison of PV Elite and Theoretical Calculation

The calculation of the thickness, pressure, stresses etc. are done by using PV Elite software and ASME code. The comparisons are shown in the following tables 4.1 to 4.7.

Comparing the manual calculations with the PV Elite calculations, the results are shown in the table. The variations of thickness, pressure and stress of the internal pressure, nozzle, flange and tubesheet are much lower and it is within the limit. There isn't any part that is fail due to that.

Table 4.1: Comparison of Internal pressure-(Tube side-LHS/RHS channel)

	PV Elite Calculation	Theoretical Calculation	Variation,(%)
Thickness (mm)	11.4472	11.4471	0.0008735
MAWP (kg/cm^2)	18.923	18.9228	0.001056
MAPNC (kg/cm^2)	22.993	22.9928	0.000869
Stress (kg/cm^2)	794.699	794.6889	0.0012709

Table 4.2: Comparison of Internal pressure-(Tube side-RHS dish end)

	PV Elite Calculation	Theoretical Calculation	Variation,(%)
Thickness (mm)	11.4147	11.4146	0.000876
MAWP (kg/cm^2)	11.352	11.3519	0.0008809
MAPNC (kg/cm^2)	15.463	15.4634	0.0002586
Stress (kg/cm^2)	1314.764	1314.7463	0.001346

Table 4.3: Comparison of Internal pressure-shell side

	PV Elite Calculation	Theoretical Calculation	Variation,(%)
Thickness (mm)	16.7052	16.7052	0
MAWP (kg/cm^2)	19.040	19.0398	0.00105
MAPNC (kg/cm^2)	22.993	22.9928	0.0008698
Stress (kg/cm^2)	1285.670	1285.6667	0.0002566

Table 4.4: Comparison of Nozzle

	PV Elite Calculation	Theoretical Calculation	Variation (%)
Thickness-cylindrical shell (mm)	8.2769	8.2769	0
Thickness-nozzle wall (mm)	2.2547	2.2547	0

Table 4.5: Comparison of Blind Flange

	PV Elite Calculation	Theoretical Calculation	Variation (%)
Tangential Stress, operating (kg/cm^2)	776.44	705.8952	9.086
Tangential Stress, seating (kg/cm^2)	58.11	57.4122	1.20082

Table 4.6: Comparison of Integral Weld Neck Flange

	PV Elite Calculation	Theoretical Calculation	Variation (%)
Tangential Stress, operating (kg/cm^2)	644.67	631.3529	2.06572
Tangential Stress, seating (kg/cm^2)	413.86	408.972	1.18107
Longitudinal Hub Stress, operating (kg/cm^2)	1382.35	1352.7071	2.14438
Longitudinal Hub Stress,seating (kg/cm^2)	877.44	876.2441	1.2615
Radial Flange Stress, operating (kg/cm^2)	37.33	36.5367	2.1251
Radial Flange Stress,seating (kg/cm^2)	23.96	23.6674	1.2212

Table 4.7: Comparison of Tube sheet

	PV Elite Calculation	Theoretical Calculation	Variation (%)
Shell Membrane Stress (kg/cm^2)	399.0044	398.9319	0.01821
Shell Bending Stress (kg/cm^2)	123.3082	117.558	4.6632

4.2 Finite Element Analysis

4.2.1 Finite Element Model

- **Computer Program**

The modelling is performed in PRO-E version 5,and analysis is performed using finite element analysis computer program ANSYS version 11.

- **Design Data**

Table 4.8: Design Data

Design Data	ASME SEC.VIII DIV-1 EDITION 2007 ADD 2009 TEMA (NINTH EDITION) CLASS R	
	SHELL SIDE	CHANNEL SIDE
Process Fluid	Propane Mixture	Water
Design Pressure Internal /MAWP (kg/cm^2)	17.4/19.038	10.6/10.656
Design Temperature ^0C	120	65

- **Material of Construction**

Shell, RF Pad, Nozzle Neck : SA 516 GR 70

Flange : SA 350 GR LF2 CL1

- **Material Properties**

Table 4.9: Material Properties

Material	Young's Modulus, MPa	Poisson's Ratio	Density, Kg/m3	Tensile Ultimate Strength,MPa	Max.Allowable Stress, MPa
SA 516 GR 70 SA 350 GR LF2 CL1	195000	0.3	7850	485	138

- **Geometric Model**

A 3D model of head to nozzle junction is considered with weld.The model comprises intersections shell, head and nozzle.A geometric model is shown in fig4.1.The dimensions are as follows:

ID of shell = 2180 mm

Thickness of shell = 18 mm

ID of nozzle = 573.6 mm

Thickness of nozzle = 18 mm

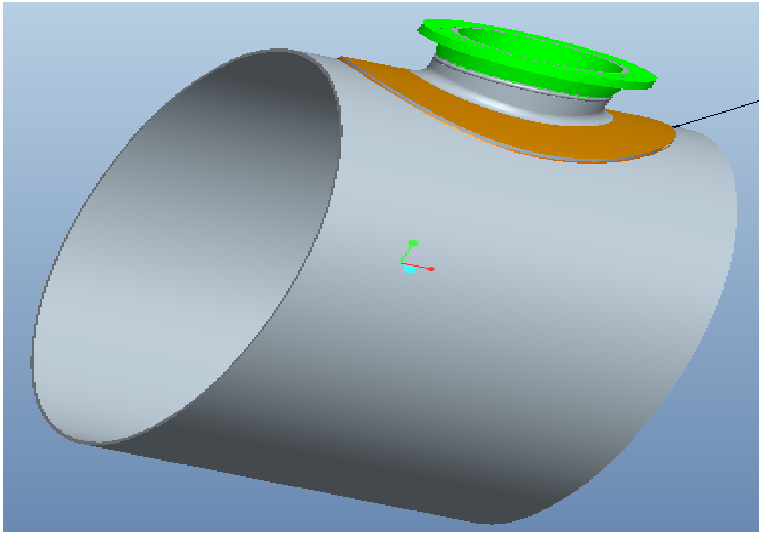

Figure 4.1: Geometric Model

- **Finite Element Mesh**

The characteristics of the elements used for finite element analysis as follows: SOLID187 element is a higher order 3-D, 10-node element. SOLID187 has a quadratic displacement behavior and is well suited to modeling irregular meshes (such as those produced from various CAD/CAM systems). The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions. The element has plasticity, hyperelasticity, creep, stress stiffening, large deflection, and large strain capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elastoplastic materials, and fully incompressible hyperelastic materials.

Type of element : 3-D 10-Node Tetrahedral Structural Solid (Solid 187)

Total no. of elements in the model :38173

Total no.of nodes in the model :74737

Finite Element Meshing fig4.2 is as follows:

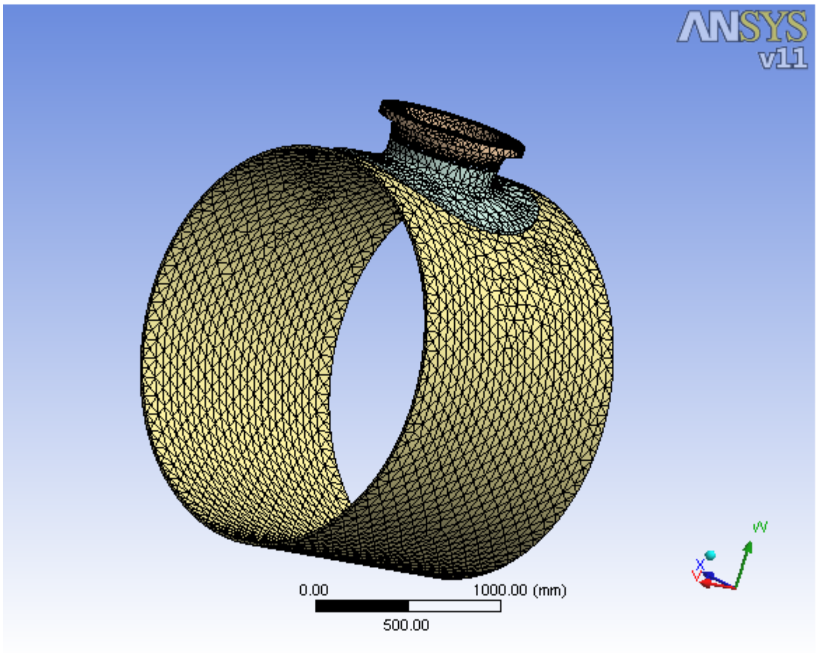

Figure 4.2: Finite Element Meshing

- **Mechanical Boundary Condition**

For evaluating the stresses due to the structural loads, the right and left periphery of the shell are fixed and $10.6 kg/cm^2$ pressure is given inside and it is normal to the shell inner diameter.

- **Static Structural Analysis**

A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those caused by time-varying loads. A static analysis can, however, include steady inertia loads (such as gravity and rotational velocity), and time-varying loads that can be approximated as static equivalent loads (such as the static equivalent wind and seismic loads commonly defined in many building codes). Static analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time. The types of loading that can be applied in a static analysis include:

1.Externally applied forces and pressures

2.Steady-state inertial forces (such as gravity or rotational velocity)

3.Imposed (nonzero) displacements

4.Temperatures (for thermal strain)

According to that stresses are generated, von-mises stress fig4.3, maximum principal stress fig4.4 and normal stress fig4.5 are as follows:

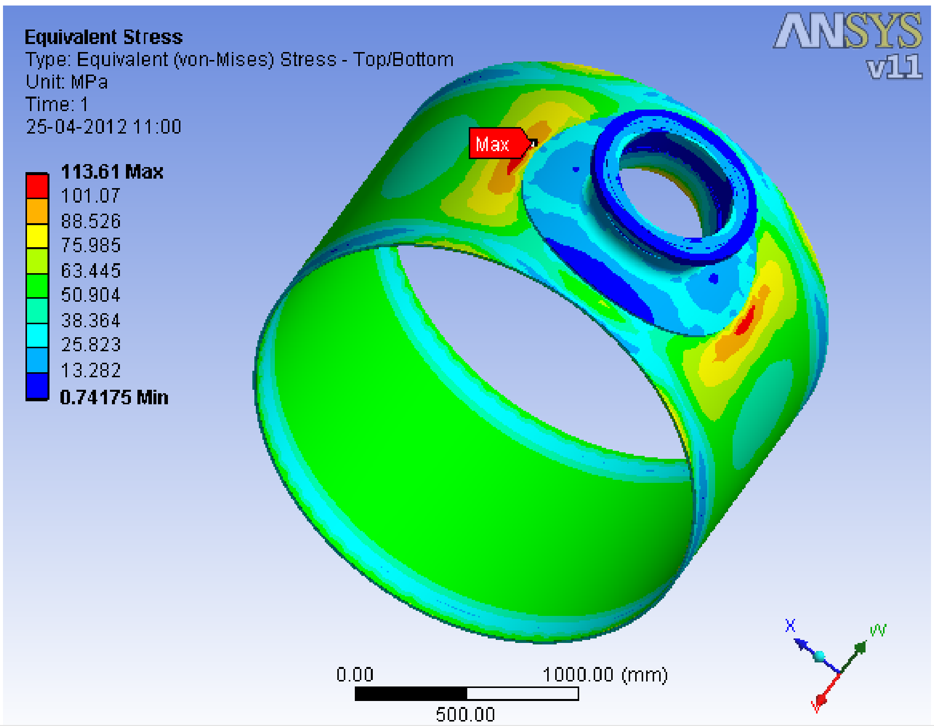

Figure 4.3: Von Mises Stress

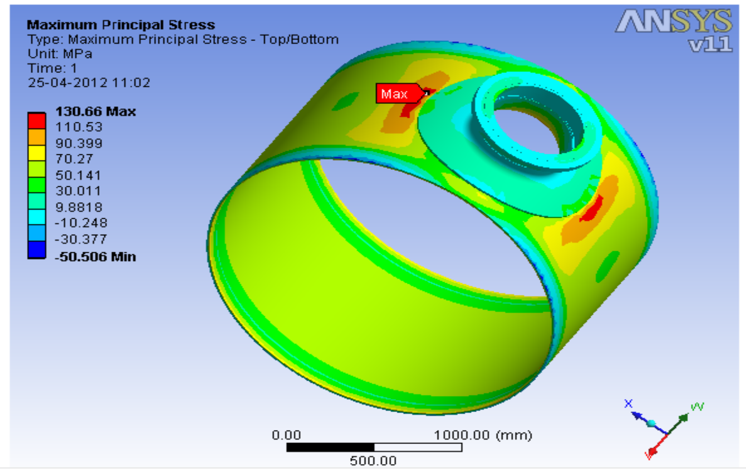

Figure 4.4: Maximum Principal Stress

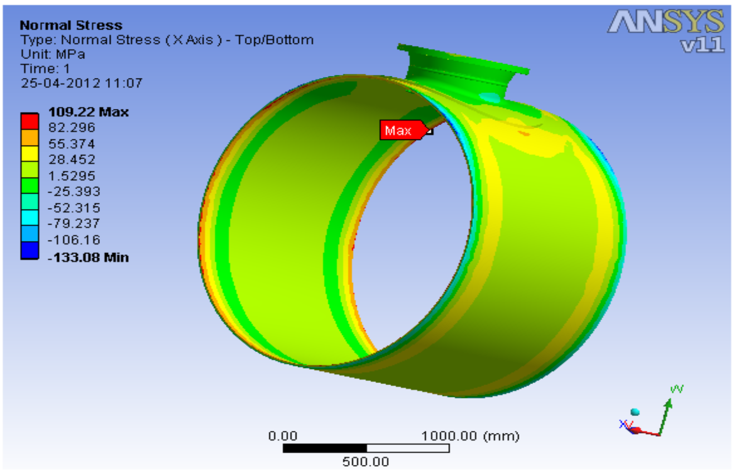

Figure 4.5: Normal Stress

Results obtained from finite element analysis are shown in the following table4.10

Table 4.10: FEA results

Stresses	Obtained MPa	Permissible MPa
Von-Mises	113.61	138
Max.Principal	130.66	138
Normal	109.22	138

Stresses obtained from the static structural analysis are quite less than the permissible limit.So the design and analysis of shell to nozzle junction of shell and tube type fixed tube sheet heat exchanger is safe for the given working conditions.Also it is verified that adopted procedure gives satisfactory results within the safe working limits.

Chapter 5

Conclusion and Future Scopes

5.1 Conclusion

- The calculation of the thickness,stresses etc. has been done by using ASME code and PV Elite software. After comparing theoretical calculation and PV Elite calculation, the variation of the thickness and stresses are much lower and it is within the limit.

- The solid model of the shell to nozzle junction is prepared by considering weld at the junction using PRO-E. The weak section of shell and tube type fixed tube sheet heat exchanger i.e noozzle to shell junction is analyzed with finite element analysis method using Ansys.

- Results obtained from theoretical calculations and analysis performed is within the permissible limits and variations are not exceeding more than five percentage.

5.2 Future Scopes

- Present work highlights about the complete static structural analysis of shell to nozzle junction but the other junctions like shell to saddle and shell to head can be analyzed.
- Work can also be extend to design and analysis of heat exchanger under external loading conditions.
- Zick analysis of supported saddle can be done.

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