Design and Analysis of High Pressure Feedwater Heater

By

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NIRMA UNIVERSITY

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Design and Analysis of High Pressure Feedwater Heater

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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MAY 2015

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 work

I, **Vatsal N. Desai, Roll. No.13MMED01**, give undertaking that the Major Project entitled "**Design and Analysis of High Pressure Feedwater Heater**" 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 Report, entitled "**Design and Analysis of High Pressure Feedwater Heater**", submitted by **Mr. Vatsal N. Desai (Roll No. 13MMED01)**, towards the partial fulfillment of the requirements for the award of Degree of **Master of Technology** in Mechanical Engineering **(Design Engineering)** of Institute of Technology; Nirma University, is the record of work carried out by him under our supervision and guidance. The work submitted has, in our opinion reached a level, required for being accepted for examination. The results embodied in this major project work, to the best of our knowledge, have not been submitted to any other University or Institution for award of any degree or diploma.

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Abstract

Heat exchanger is an essential part of any engineering system which works on thermal gradient principle. High Pressure Feedwater heater is one of the important component for supplying feedwater to the steam generator. It reduces temperature difference, avoid thermal shock to boiler, reduces fuel consumption, improves thermodynamic gain of the regenerative steam cycle by means of utilizing bled-off steam.

It is highly necessary to critically design the different components of feedwater heater as it is subjected to varying pressure conditions (i.e. 8 MPa to 34 MPa) and varying temperature from ambient temperature to 189^{o}C. These variation make it more critical as different zones namely desuperheating, condensation and subcooling are formed in the same shell which creates even high thermal gradients. Various International Codes such as ASME, TEMA, HEI are available for designing such heat exchangers.Commonly heaters are designed to sustain around 150 bars pressure.Hence designing of a special purpose,187 bar, heater as a whole from basic fundamental is highly essential.

As it is coming under non-conventional equipment category, it is difficult to predict its behaviour in the working condition.The analysis of shell to nozzle, tube to tubesheet is critical due to discontinuity stresses and local stresses. This analysis becomes a requirement for such heaters as thermal shock and pressure differences make the joints even more vulnerable to failure.

Mechanical Design Calculations are carried out in accordance with ASME Section VIII Div.1. Further, analysis of shell to nozzle junction(steam inlet/outlet and feedwater inlet/outlet), tubesheet using FEA tool is carried out and results, conforming to ASME standards, is within the allowable limits. Design of leg support has been carried out for horizontal Channel Shell.

Keywords: Feedwater heater, ASME Section VIII Div. 1 and 2, Shell to nozzle junction analysis,tubesheet analysis

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Nomenclature

MAWP	:	Maximum Allowable Working Pressure
D	:	Diameter
CA	:	Corrosion Allowances
R	:	Inside Radius
E	:	Joint Efficiency
MDMT	:	Minimum Design Metal Temperature
t	:	Nominal Thickness
t_r	:	Required Thickness
t_{rn}	:	Required Thickness of nozzle
t_n	:	Thickness of nozzle
t_{rh}	:	Required Thickness of head
S_v	:	Allowable Stress of vessel
S_n	:	Allowable Stress of nozzle
S_p	:	Allowable Stress of reinforcement pad
P	:	Design Pressure
h	:	Tubesheet thickness
G_o	:	thickness of hub at small end
G_i	:	thickness of hub at back of flange

Chapter 1

Introduction

1.1 Preliminary Remarks

All modern large steam power plants use a process of regenerative feedwater heating to increase the overall cycle efficiency of plant and to minimize induced thermal stresses in the boiler. In large power plants feedwater heating generally takes place in multiple stages. Mostly feedwater heaters are shell and tube type heat exchangers.Also, majorly U-tubes are employed against straight tubes because U-tubes are more tolerant to thermal expansion during operation. A beneficial by-product of the energy extracted by the heaters is the reduced rate of rejection of energy to the environment.

1.2 Motivation

Feedwater heaters with higher side pressure value ranging from 100-150 bars are usually made in the industry. So the heater with higher pressure value of 187 bar pose bigger challenges in terms of design and manufacturing. For this reason, design of heater from basics is required which widens the scope of learning. FEA tools must be employed for analysis of critical junctions like shell to nozzle, tubesheet.As it is coming under non-conventional equipment category, it is difficult to predict its behaviour in the working condition.This analysis becomes a requirement for such heaters as thermal shock and pressure differences make the joints even more vulnerable to failure. Analysing various components and interpreting the results increase exposure to wide range of problems faced by industry in day to day life.

1.3 Aim and Scope of Work

1. The present work aims at designing the High Pressure feedwater heater in accordance with the requirements defined by the customer to the host industry. The design is carried out in compliance with ASME Section VIII Division I.

2. Nonstandard leg support on the channel side is to be provided as per customer requirement. This method of providing support doesnot comply with the standard industry practice. So detailed Finite Element Analysis of the support structure and it's joint to respective shell is to be carried out.

3. Analysis of all major nozzles(viz, steam inlet, steam outlet, feedwater inlet, feedwater outlet) to shell junction is to be performed. Also the analysis of tubesheet is carried out and validated by comparison with the calculated values.

1.4 Layout of Thesis

Chapter 2 Gives an overview of existing literature.

Chapter 3 Design calculations are carried out for various components of Heater.

Chapter 4 Design calculations and analysis is done leg support .

Chapter 5 Analysis for various components of HP Heater is carried out.

Chapter 6 The project work done is concluded and scope for future work is laid.

Chapter 7 Analysis for various components of HP Heater is carried out.

Bibiliography

Chapter 2

Literature Review

2.1 Classification Feedwater Heater

1. Open feedwater heater-

   In open type construction the extracted steam directly mixes with the feedwater to be heated. As the steam mixes with feedwater open heaters are generally referred as deaerators. Its major disadvantage is that it requires much larger space as compared to closed heaters.

2. Closed feedwater heater-

   In closed construction steam and feedwater donot come in direct contact. The closed heaters are generally referred to as feedwater heaters. Its construction is of shell and tube heat exchanger. Most modern plants with high pressure steam generators employ one deaerator and other heaters with closed construction.

2.2 Classification of Closed feedwater heaters

2.2.1 Classification based on pressure

1. Low Pressure Heater:[5]

   A heater located (with regard to feedwater flow) between the condensate pump and either the boiler feed pump or, if present, an intermediate pressure (booster) pump. It normally extracts steam from the low pressure turbine.

2. Intermediate Pressure Heater:

   A heater located between the booster pump and the boiler feed pump. Usually the tubeside pressure is within 200-300 psi of the low pressure heaters, and the steam is extracted from an intermediate pressure turbine.

3. High Pressure Heater:

   A heater located downstream of the boiler feed pump. Typically, the tubeside design pressure is at least 1500 psig, and the steam source is the high pressure turbine.

2.2.2 Classification based on orientation

1. Horizontal:

   Most heaters are of this configuration. These are the most stable in regard to level control, although they occupy more floor space. Disassembly is by means of either shell or bundle removal. Most are floor mounted, although some are mounted in the condenser exhaust neck.

2. Vertical, Channel Down:

   Although these conserve floor space, the amount of control area available for liquid level fluctuation is less. Disassembly is by shell removal. Installation and removal may be more difficult than for horizontal heaters.

3. Vertical, Channel Up:

   These are the least frequently used. Disassembly is by means of bundle removal. If a subcooling zone is present, it must extend the full length of the bundle, since the water must enter the bottom and exit at the top end of the heater.

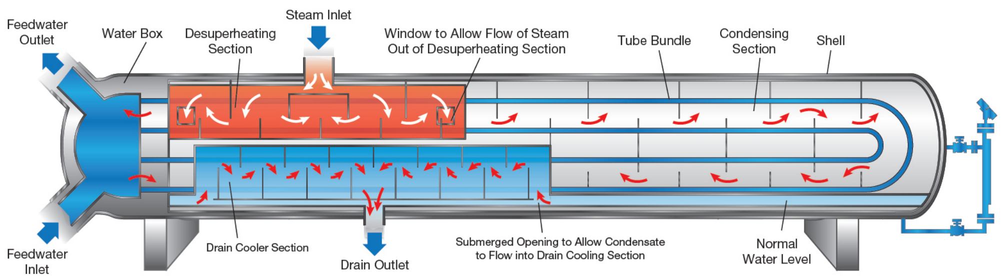


Figure 2.1: Horizontal HP Heater

[5]

2.3 Classification of Heat Exchanger

(a) Classification of TEMA class heat exchanger[1]-

   i. Class R- The TEMA Mechanical Standards for Class R heat exchangers specify design and fabrication of unfired shell and tube heat exchangers for the generally severe requirements of petroleum and related processing applications.

   ii. Class C- The TEMA Mechanical Standards for Class C heat exchangers specify design and fabrication of unfired shell and tube heat exchangers for the generally moderate requirements of commercial and general process applications.

   iii. Class B- The TEMA Mechanical Standards for Class B at exchangers specify design and fabrication of unfired shell and tube heat exchangers for chemical process service.

(b) Nomenclature of TEMA heat exchangers- The current HP Heater is DFU type.

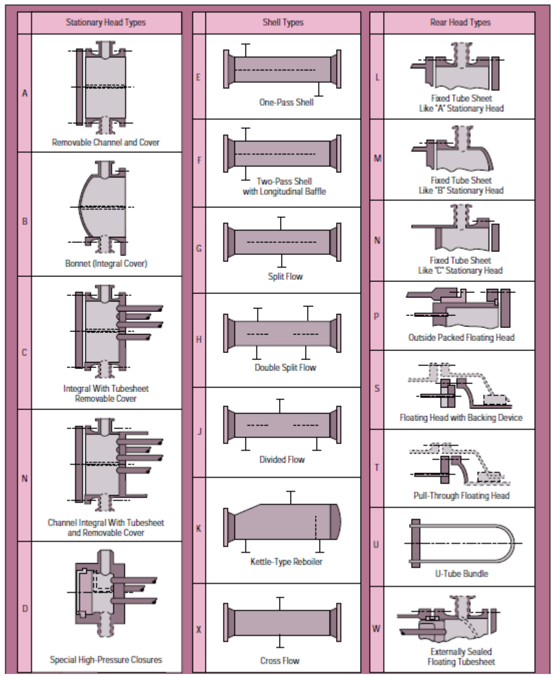


Figure 2.2: Nomenclature of TEMA heat exchangers

[1]

2.4 Major Components of Heat Exchanger

2.4.1 Tubes

Tubes are mainly straight tubes and U-tubes. Several conditions need to be considered while deciding the tube outside diameter, tube pitch, tube layout. Some of which are operating temperature and pressure, corrosive nature of the fluids and thermal stresses due to differential thermal expansion.

Tube layout must be selected in such a manner so as to achieve maximum possible heat transfer area. Four standard tube layout configurations are triangular

30^o, rotated triangular 60^o, square 90^o and rotated square 45^o. Triangular layouts assist in stronger tubesheet for given shell side area, compact arrangement and better shell side heat transfer coefficient. Square type layouts are used when major consideration is mechanical cleaning of tubes.

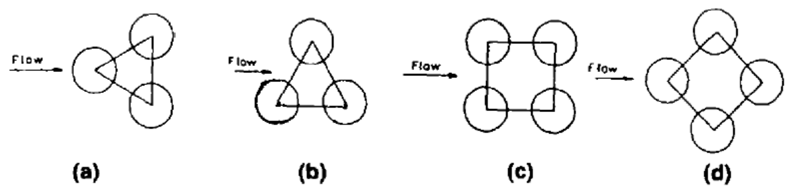


Figure 2.3: Tube patterns (a) 30^o (b) 60^o (c) 90^o (d) 45^o
[7]

2.4.2 Shells

Shells contain the fluid used to carry out heat exchange. Shells can be manufactured for different standard and non standard sizes. If the diameter of shell is small then it is fabricated from standard pipes available otherwise shells are manufactured by rolling of plates. As compared to cost of tubes the cost of shell is much higher so economically it better to have longer shells with smaller diameter such that it satisfies the permitted plant layout factors.

Shells are generally cylindrical in shape but according to the requirement of the design and to incorporate the number of tubes and tube layout structure the shape of shell may vary as in the case of nuclear reactors.

2.4.3 Nozzles

Nozzles serve the purpose as entryway and exitway for shell side and tube side fluid. The size and orientation of the nozzles depend on the amount of fluid flow and flow velocity required.

When the lateral displacement δ of openings in axial orientation with respect to vessel axis is zero the nozzle is defined as radial nozzle.

When the lateral displacement of nozzle axial orientation with respect to vessel axis is $0 \langle \delta \langle$ R then the nozzle is defined as hill side nozzle. And when $\delta \rangle$ R then the nozzle due to its orientation is defined as tangential nozzle.

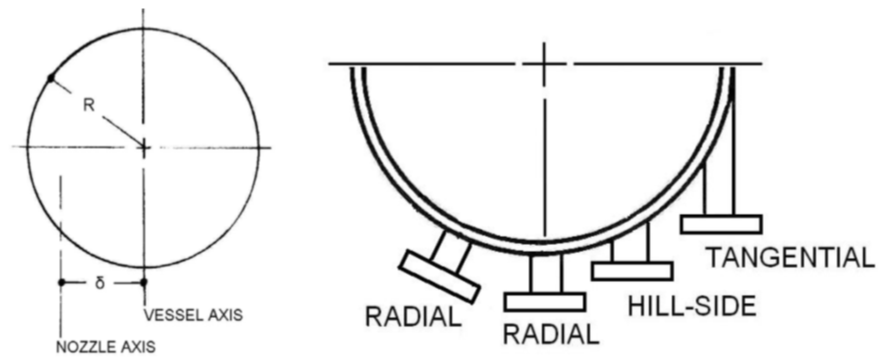


Figure 2.4: Types of Nozzles

2.4.4 Front and Rear end heads

The front and rear end heads are shown in TEMA Chart. These serve the function of entering and leaving of tube fluid. The front end head is stationary but the rear end head can be stationary or floating as per the condition of stresses between shell and tube. The deciding parameters for selection of front end head are the hazards due to mixing of shell and tube fluid, leakage to ambient and operating pressure, cost, maintenance and inspection. Similarly for selection of rear end head allowance for thermal stresses, sealing any leakage path for the shell fluid to ambient, prevention of mixing of tube and shell fluids, and provision to remove the tube bundle for cleaning the shell side.

2.4.5 Internal Shroud

Internal shrouds are used when the phase change of the shell side fluid governs the heat transfer efficiency. For example, the high pressure feedwater heater is divided into three zones namely, desuperheating , condensing and subcooling. If these zones are not distinctly separated then steam may enter the subcooling zone and thermal efficiency of the steam cycle will come down.

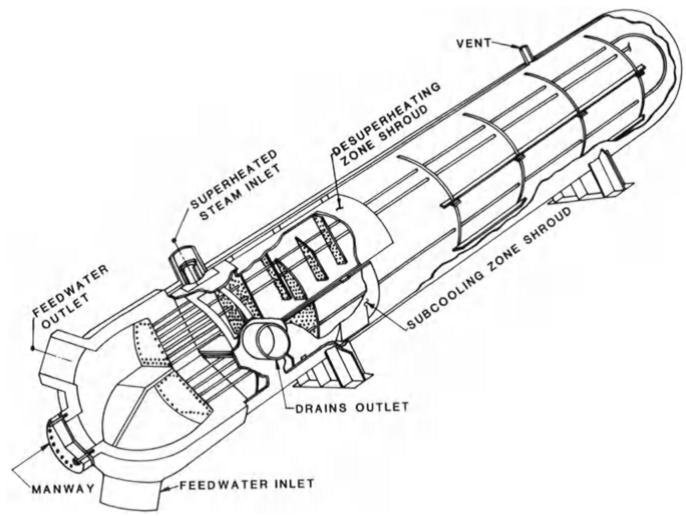


Figure 2.5: Internal Shroud

2.4.6 Baffles

Baffles are the obstructions provided in the path of shell side fluid. Baffles serve the purpose of supporting the tube structure, reducing tube vibration. Baffles also contribute in increasing heat transfer efficiency as it increases heat transfer area. Baffles can be broadly classified into two types namely Transverse and longitudinal.

Transverse baffles direct the shell side fluid through the tubes. Every shell and tube type heat exchanger has transverse baffles except X and K shells. Transverse baffles can be further classified into three types,

(a) Segmental baffle- Single and double segmental baffles are mainly employed because of their ability to help increase the heat transfer for a given pressure drop and at the same time occupying minimum space.

(b) Disk and doughnut baffles- These baffles have small perforations between tube holes so that a combination of cross-flow and longitudinal flow for lower shell side pressure drop which results in slightly higher heat transfer coefficient as compared to pure longitudinal flow.

(c) Orifice baffles- These baffles are rarely used. There is very large clearance between baffle hole and tube which after fouling cannot be cleaned. These baffles donot provide support to the tubes.

Longitudinal baffles should be used only when it can be welded to the shell and tubesheet. Longitudinal baffles divide the shell into different parts so that multiple passes on shell side is possible.

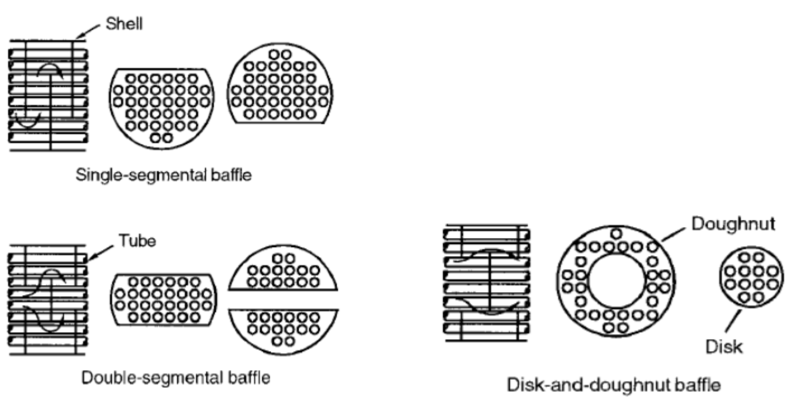


Figure 2.6: Types of Baffles

[11]

2.4.7 Tubesheets

Tubesheets are circular metal plates with perforated holes so as to match the geometric pattern of the tubes. Tubesheet is one of the most important part of the heat exchanger as it separates the shell side fluid from the tube side fluid.

Tubesheet can be connected to a fixed type heat exchanger in two ways, both side integral connection and shell side integral and tube side bolted connection.

Whereas the connection of tubesheet to shell and channel for floating head and U-tube configuration can be of three types, both side integral, both side gasketed and one side integral and one side gasketed. Tubes are attached to Tubesheet

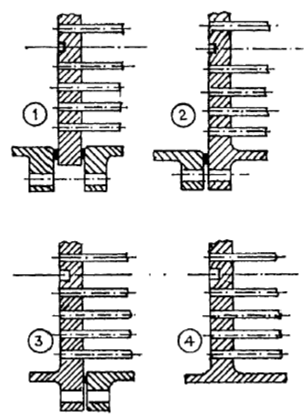


Figure 2.7: Various Tubesheet Connections

[7]

by; rolling, welding, rolling and welding, explosive welding and brazing. Tube to Tubesheet joints are very critical and need special care while designing and production.

2.5 Review of Published Study

Ming-Hsien Lu, Jiun-Shya Yu, Jien-Jong Chen[8] have considered three different 2-d axisymmetric models of shell to nozzle junction and a 3-d model of the same. The vessel radius considered for the three 2-d axisymmetric models are 1, 1.5, 2 times the actual vessel radius. The research paper concludes that for replacing 3-d model by 2-d model the actual vessel radius can be employed. From this research paper the methodology for 3-d analysis of shell to nozzle junction can be utilized.

A.I.SOLER and K.P.SINGH[12] present a method for elastic plastic analysis of integral tubesheet of U-Tube heat exchanger. The paper presents a non-incremental solution technique which yields a conservative result. The paper concludes that use of edge restraints reduces the required thickness of tubesheet. This paper serves as basis for the tubesheet analysis of the present project.

Kotcherla Sriharsha, Venkata Ramesh Mamilla and M.V. Mallikarjun[13] has done research on Strength Analysis of Tube to Tube Sheet Joint in Shell and Tube Heat Exchanger. Researchers have performed contact analysis of gap between tube to tubesheet element. The paper concludes that to obtain a good and leak proof joint the loading pressure, Clearance for tube thickness, Stress-Strain data of materials used in assembly should be monitored closely. This paper helps in investigation of the loading conditions to be applied to analyse the present tube to tubesheet joint.

Shafique M.A. Khan[6] analyses the stress distribution in both horizontal pressure vessel and saddle support. The results provide details of stress distribution in different parts of the saddle. The researcher also investigates the effect of changing the load and various geometric parameters. Some ratio of the distance of support from the end of the vessel to the length of the vessel and ratio of the length of the vessel to the radius of the vessel for minimum stresses both in the pressure vessel and the saddle structure have been recommended. The paper serves as a guideline for analysis of present vessel and saddle structure.

Alwyn S. Tootha, John S.T. Cheungb, Heong W. Ngb, Lin S. Ongb, Chithranjan Nadarajahc[14] present a way to to dispense with the sliding support and to provide saddle designs which, although fixed to the platform or foundation, do not result in the pressure vessel. This paper provides general recommendations for the most appropriate saddle geometries,and details the way in which design-by-analysis and fatigue-life- assessments may be carried out using the stresses that arise from these designs being overstressed when thermal loading occurs.

V. G. Ukadgaonker, P. A. Kale, Mrs. N. A. Agnihotri and R Shanmuga Babu [15] paper is to show the different techniques developed by various researchers in the analysis of tube sheets. A thorough literature review is undertaken and the different techniques such as analytical, experimental and numerical are dealt with separately. The results obtained by various researchers are compared and the author's work is also mentioned.

Wolf D. Reinhardt[10] This paper examines the yielding of a perforated tubesheet and suggests how the behavior could be represented by a solid with equivalent properties. The observed yielding behavior of a perforated element suggests that an anisotropic (orthotropic) yield criterion is needed to reflect the yield behaviour of a perforated plate. The main advantage of the approach is its adaptibility which derives from the equivalencing with an elastic-plastic ligament model.

2.6 Review of Books and Standards Referred

ASME Boiler and Pressure Vessel Code Section VIII Division I and II[3], provide guidelines for design of all types of unfired pressure vessel and heat exchangers. The rules of the code have been strictly followed for design of all components using Division I and analysis of critical components is carried out using Division II.

Standards of the Tubular Exchanger Manufacturers Association (TEMA) [1] These standards supplement and define the ASME Boiler and Pressure Vessel Code Section VIII Division I for heat exchanger applications. The design of various heat exchanger components have been checked with the guidelines put up by TEMA.

Standards for Closed Feedwater Heaters by HEAT EXCHANGE INSTITUTE[5] serve as check points while designing feedwater heaters. Design and manufacturing aspects have been covered in the standard.

Shah R K and Dusan P. Sekulic[11] in his book Fundamentals of Heat Exchanger Design has in depth explanation of various components of different types of heat exchangers, considerations for thermal design and heat exchanger design procedures.

Kuppan T. [7] in his book Heat Exchanger Design Handbook has dedicated several chapters on heat exchanger classification, thermal design and mechanical design of mostly all type of heat exchangers. The book is a fitting companion for designing heat exchanger.

Moss Dennis R.[9] Pressure Vessel Design Manual is one of the most important books for designing components for pressure vessel which are not prescribed in International Codes (like ASME etc.). The book is specially useful for designing the supports of pressure vessels.

Bednar H H[2] Pressure Vessel Design Handbook discusses in depth on Membrane analysis of shells, discontinuity of stresses, local stress analysis of attachments to the pressure vessel. The book has detailed theoretical calculations explaining many fundamentals which are directly used as reference.

Brownell L. L. and Young E.H.[4] Process Equipment Design provides a solid foundation by detailed calculation of fundamental formulas and concepts directly used in many International Codes.

Chapter 3

Mechanical Design Using ASME Section VIII Division-I

3.1 Selection of Material of Construction

The selection of material[2] needs to be done for all the components namely main shell, channel shell, head, tubes, tubesheet, bolts. The material selection mainly depends on following three factors-

1. Corrosion- Corrosion is the deterioration of material due to chemical reactions taking place which makes it one of the most important factor. The corrosive nature of the chosen material may change significantly if the operating fluid or operating conditions change.

2. Strength- The strength level of a material has significant influence on its selection for a given application. This is especially true at elevated temperatures where the yield and ultimate strength are relatively low and creep and rupture behaviour may control allowable stress value.

3. Cost- Evaluation of material cost should be done versus factors such as corrosion, expected life of equipment, availability of material, replacement cost and

code restrictions on fabrication and repairs.

. The following chart provides the guidelines for selection of material.

	Design Temperature, °F	Material	Plate	Pipe	Forgings	Fittings	Bolting
Cryogenic	−425 to −321	Stainless steel	SA-240-304, 304L, 347, 316, 316L	SA-312-304, 304L, 347, 316, 316L	SA-182-304, 304L, 347, 316, 316L	SA-403-304, 304L, 347, 316, 316L	SA-320-B8 with SA-194-8
	−320 to −151	9 nickel	SA-353	SA-333-8	SA-522-1	SA-420-WPL8	
Low temperature	−150 to −76	3½ nickel	SA-203-D	SA-333-3	SA-350-LF63	SA-420-WPL3	SA-320-L7 with SA-194-4
	−75 to −51	2½ nickel	SA-203-A				
	−50 to −21	Carbon steel	SA-516-55, 60 to SA-20	SA-333-6	SA-350-LF2	SA-420-WPL6	
	−20 to 4		SA-516-All	SA-333-1 or 6			SA-193-B7 with SA-194-2H
	5 to 32		SA-285-C	SA-53-B SA-106-B	SA-105 SA-181-60,70	SA-234-WPB	
Intermediate	33 to 60 61 to 775		SA-516-All SA-515-All SA-455-II				
Elevated Temperature	776 to 875	C-½Mo	SA-204-B	SA-335-P1	SA-182-F1	SA-234-WP1	
	876 to 1000	1Cr-½Mo	SA-387-12-1	SA-335-P12	SA-182-F12	SA-234-WP12	
		1Cr-½Mo	SA-387-11-2	SA-335-P11	SA-182-F11	SA-234-WP11	
	1001 to 1100	2¼Cr-1Mo	SA-387-22-1	SA-335-P22	SA-182-F22	SA-234-WP22	with SA-193-B5 SA-194-3
	1101 to 1500	Stainless steel	SA-240-347H	SA-312-347H	SA-182-347H	SA-403-347H	SA-193-BB with SA-194-B
		Incoloy	SB-424	SB-423	SB-425	SB-366	
	Above 1500	Inconel	SB-443	SB-444	SB-446	SB-366	

Figure 3.1: Guidelines for Material Selection

[2] Adhering to above guidelines the materials selected are as follows:

1. Shells and Head- SA-516 Gr 70. Because it has maximum allowable stress from the given materials.
2. Tubesheet- SA-350 Gr LF2.
3. Tubes- SA-213 TP 304.
4. Flange- SA- 105.
5. Stud/ Nut- SA-193 B7/ SA-194-2H.

3.2 Predefined Data Inputs

Sr no.	Inputs	Unit	Shell Side	Tube Side
1	Design Code		ASME Sec. VIII Div. I	
2	Operating Fluid	-	Steam	High Pressure Feedwater
3	Design Pressure(P) (int.)	bar(g)	14.51	184
4	Design Temperature	^oC	360	200
5	Radiography/Joint Efficiency(E)	-	Full/1.0	Full/1.0
6	Corrosion allowance	mm	3.2	3.2(other than tubes)
7	Inner Diameter (D)	mm	808	724
8	Position of Vessel		Horizontal	
9	MDMT	^oC	-20	-20

Table 3.1: Predefined Inputs

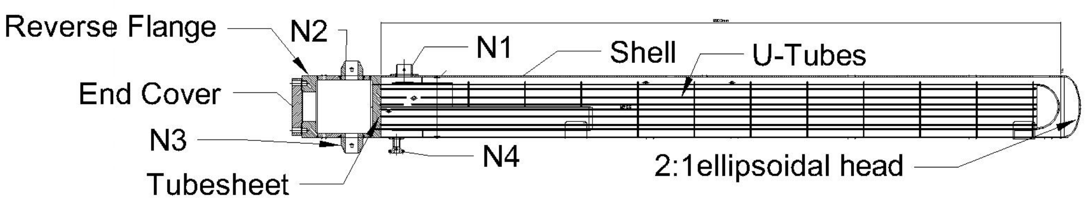


Figure 3.2: HP Heater model

3.3 Internal Pressure Calculations

3.3.1 Thickness of Cylindrical Shells as per UG-27

The minimum thickness required by a cylindrical shell will be the **maximum** value calculated by considering Circumferential stress, Longitudinal stress and thickness arrived at by adding a factor to the corrosion allowance.

Main Shell

Using predefined inputs of design pressure, internal radius, joint efficiency and allowable stress of material at that temperature.

1. For Circumferential Stress

$$t_r = \frac{P * R}{S * E - 0.6 * P} + C.A. = 7.89mm \tag{3.1}$$

The above equation is applicable only when $t_s \leq (0.5*R)$ and $P \leq (0.385*S*E)$

2. For Longitudinal Stress

$$t_r = \frac{P * R}{2S * E + 0.4 * P} + C.A. = 5.44mm \tag{3.2}$$

The above equation is applicable only when $t_s \leq (0.5*R)$ and $P \leq (1.25*S*E)$

3. Minimum thickness according UG 16 (b) (4)

$$t_r = 2.5 + C.A. = 5.7mm \tag{3.3}$$

Channel Shell

Using predefined inputs of design pressure, internal radius, joint efficiency and allowable stress of material at that temperature.

1. For Circumferential Stress

$$t_r = \frac{P * R}{S * E - 0.6 * P} + C.A. = 55.98mm \tag{3.4}$$

The above equation is applicable only when $t_s \leq (0.5*R)$ and $P \leq (0.385*S*E)$

2. For Longitudinal Stress

$$t_r = \frac{P * R}{2S * E + 0.4 * P} + C.A. = 25.51mm \tag{3.5}$$

The above equation is applicable only when $t_s \leq (0.5*R)$ and $P \leq (1.25*S*E)$

3. Minimum thickness according UG 16 (b) (4)

$$t_r = 2.5 + C.A. = 5.7mm \tag{3.6}$$

3.3.2 Thickness Calculation For Right End Head as per UG-32

For 2:1 semiellipsoidal dished end, the minimum required thickness of head computed by following formula,

Required thickness for head,

$$t_{rh} = \frac{P * D}{2 * S * E - 0.2 * P} + C.A. = 7.75mm \tag{3.7}$$

Sr no.	Shell and head	Minimum Required Thickness(mm)	Nominal Thickness(mm)
1	Main Shell	7.89	10
2	Channel Shell	55.98	58
3	Right End Head	7.75	10

Table 3.2: Thickness values for Shell and Head

3.3.3 Calculation for Nozzles using UG-37

For designing the shell/head as discussed above it is considered that there are no cutout sections over the whole length. When nozzle is to be placed on the shell/head a circular portion needs to be cut out. This area needs to be replaced so that the strength is maintained. Firstly, it needs to be confirmed by calculation whether replacement of this area is required or not. This area replacement is done either by providing a reinforcement pad or increasing thickness of the shell. Many times the thickness of the shell is provided more than the required thickness, for manufacturing comfort. [3]

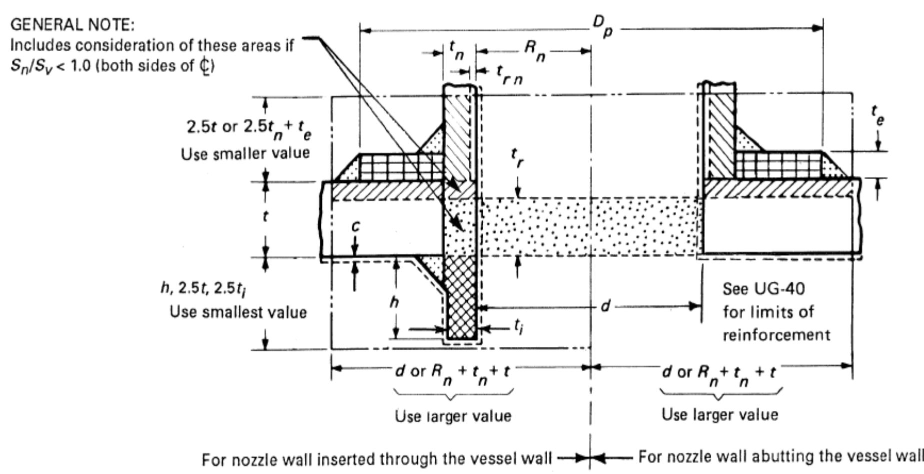


Figure 3.3: Details for required area and available area.

Required thickness of Cylindrical Shell(t_r):

$$t_r = \frac{P * (R + C.A)}{S * E - 0.6 * P} + C.A. = 7.89mm \tag{3.8}$$

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

$$t_{rn} = \frac{P * R_n}{S * E + 0.4 * P}. \tag{3.9}$$

Description	Symbol	Unit	N1	N2	N3	N4
Nozzle	-	-	DN200	-	-	DN80
Nozzle OD	D_{on}	mm	219	293	293	88.9
Nozzle thickness	t_n	mm	12.7	60	60	11.18
Size of corroded openings	d	mm	200	179.4	179.4	72.94

Table 3.3: Calculation of Nozzle Neck Thickness

The total cross sectional area of reinforcement A required

$$A = d * t_r * F + 2 * t_n * t_r * F * (1 - f_{r1}) \tag{3.10}$$

where,

d = Finished diameter of circular opening or finished dimension

t_r = Required thickness of a shell based on the circumferential stress

F = 1, Correction factor

t_n = Nozzle wall thickness

$f_{r1} = S_n/S_v$ for nozzle wall inserted through the vessel wall

Area in excess thickness in the vessel wall available for reinforcement, use larger value

$$A_1 = d * (E_1 * t - F * t_r) - 2 * t_n * (E_1 * t - F * t_r)(1 - f_{r1}) \tag{3.11}$$

$$A_1 = 2 * (t + t_n)(E_1 * t - F * t_r) - 2 * t_n * (E_1 * t - F * t_r)(1 - f_{r1}) \tag{3.12}$$

where,

E_1 = 1, when an opening is in the solid plate or in a Category B butt joint

t = Specified vessel wall thickness

Area available in nozzle projecting outward, use smaller value

$$A_2 = 5 * (t_n - t_{rn}) * f_{r2} * t \tag{3.13}$$

$$A_2 = 5 * (t_n - t_{rn}) * f_{r2} * t_n \tag{3.14}$$

where,

t_{rn} = Required thickness of a seamless nozzle wall

$f_{r2} = S_n/S_v$

Area available in nozzle projecting inward, use smaller value

$$A_3 = 5 * t * t_i * f_{r2} \tag{3.15}$$

$$A_3 = 5 * t_i * t_i * f_{r2} \tag{3.16}$$

$$A_3 = 5 * h * t_i * f_{r2} \tag{3.17}$$

where,

h = Distance nozzle projects beyond the inner surface of the vessel wall

t_i = Nominal thickness of internal projection of nozzle wall

Area available due to outward and inward weld

Area due to outward nozzle weld

$$A_{41} = (leg)^2 * f_{r2} \tag{3.18}$$

Area due to inward nozzle weld

$$A_{43} = (leg)^2 * f_{r2} \tag{3.19}$$

Checking for reinforcement requirement.

Nozzle no.	A_1	A_2	A_3	A_{41}	A_{43}	A_{av}	A(mm^2)	Reinforcement
N1	493.2	548.8	0	95.3	95.3	1232.6	1512.8	Required
N2	433.1	2250.3	0	95.3	95.3	2873.9	1397.4	Not Required
N3	433.1	2250.3	0	95.3	95.3	2873.9	1397.4	Not Required
N4	178.3	512.2	0	95.3	95.3	881.1	556.4	Not Required

Table 3.4: Reinforcement verification for Nozzles.

It is evident from the above table that the available area for replacement of the cut-out section is **less than** required area to maintain the strength. Most common practice is to provide an adequate reinforcement pad around the nozzle.

The shape and size of reinforcement pad is decided such that it is readily available and easy to attach. Thickness of reinforcement pad is taken equal to thickness of shell so that material is readily available and it will be easy fabricate. The outer diameter of reinforcement pad is varied till the area available is not more than area required.

Area available from the reinforcement pad,

$$A_5 = (D_p - d - 2 * t_n) * t_e * f_{r4} \tag{3.20}$$

where

D_p=outer diameter of reinforcement pad=300 (by trial and error)

$f_{r4} = S_n/S_v$ for nozzle wall inserted through the vessel.

Nozzle no.	A_5	A_{av}	A (mm^2)	Reinforcement
N1	710.938	1943.5	1512.8	Reinforced

3.3.4 Left Hand Side End Cover Flange

Pressure vessels require flanged joints to permit their disassembly, inspection and cleaning. Bolted joints are also utilized to alleviate stresses at sections where sharp temperature changes occur, such as the joint between tubeside and shellside chambers in a heat exchanger. The design of flanges is done in accordance with Mandatory Appendix 2.

Gasket Contact Width(N):-

$$N = \frac{G_o - G_i}{2} = 25mm$$

Basic Gasket Width(b_o):-

$$b_o = \frac{N}{2} = 12.5mm$$

Effective Gasket Width(b):-

$$b = C_b * \sqrt{b_o} = 8.9mm$$

Gasket Reaction Diameter(G):-

$$G = G_o - 2 * b = 432.18mm$$

Basic Flange and Bolt Loads

Hydrostatic End load due to pressure[H]:

$$H = 0785 * G^2 * P = 27418304.4kgf$$

Contact Load on Gasket Surfaces [H_p]:-

$$H_p = 2 * b * \pi * G * m * P = 0kgf$$

Operating Bolt Load [W_{m1}]:-

$$W_{m1} = H + H_p = 27418304.4kgf$$

Gasket Seating Bolt load [W_{m2}]:-

$$W_{m2} = \pi * G * b * y = 0kgf$$

Required Bolt Area [A_m]:-

$$A_m = MAX\left(\frac{W_{m1}}{S_b}, \frac{W_{m2}}{S_a}\right) = 15599.14cm^2$$

Minimum Gasket Contact Width [N_{min}]:-

$$N_{min} = \frac{A_b * S_a}{y * \pi * (G_o + G_i)} = 0$$

Flange Design Bolt Load for Gasket Seating [W]:-

$$W = S_a * \left(\frac{A_m + A_b}{2}\right) = 13969339.81kgf$$

Gasket Seating Force [H_G]:-

$$H_G = W_{m1} = 27418304.4kgf$$

Distance to Gasket Load Reaction [h_g]:-

$$h_g = \frac{(C - G)}{2} = 103.90mm$$

Tangential Flange Stress, Flat Head, Operating [S_{to}]:-

$$S_{to} = \frac{1.9 * W_{m1} * h_g * B_{cor}}{t^2 * G} + \frac{C * Z * P * G^2}{t^2} = 469.11kgf/cm^2$$

Tangential Flange Stress, Flat Head, Seating [S_{ta}]:-

$$S_{ta} = \frac{1.9 * W_{m1} * h_g * B_{cor}}{t^2 * G} = 259.44kgf/cm^2$$

3.3.5 Reverse Flange

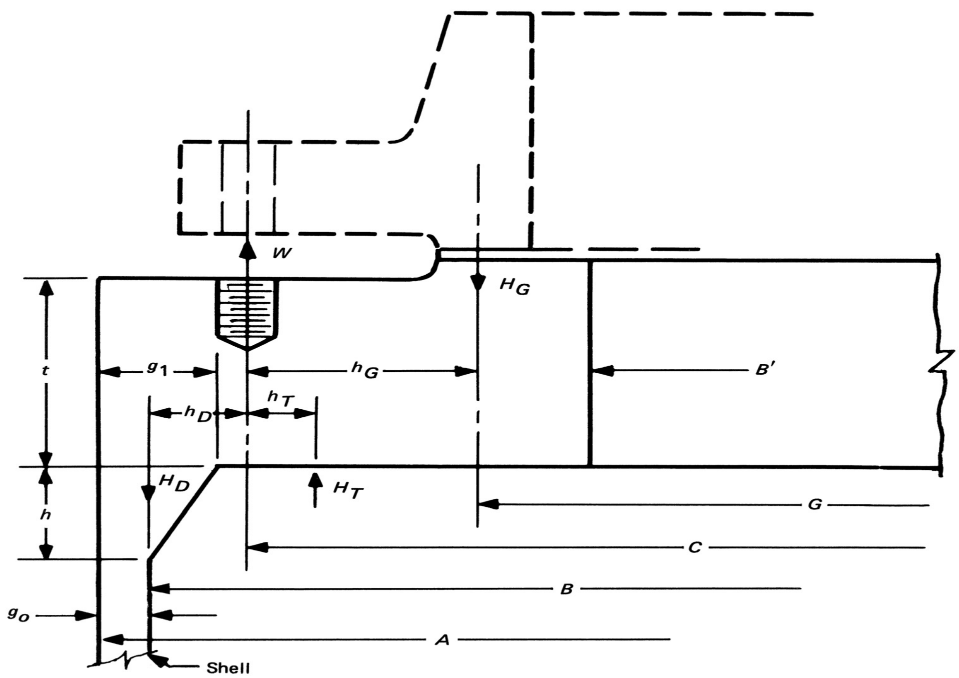


Figure 3.4: Reverse Flange.

[3]

Hub Small End Required Thickness due to Internal Pressure:

$$= \frac{(P*(D/2 + C.A))}{(S*E - 0.6*P)} = 32.57mm$$

Corroded Flange Thickness, t_c=$T - C.A = 156.8mm$

Corroded Flange ID, $B'_{cor} = B' + 2*C.A = 406.400mm$

Corroded Shell ID, $B_{cor} = A + 2(C.A - g_o - c_e) = 730.400mm$

Corroded Large Hub, $g_{1cor} = g_1 - C.A = 54.800mm$

Corroded Small Hub, $g_{0cor} = g_1 - C.A = 54.800mm$

Code R Dimension, $R = (\frac{C-B_{cor}}{2}) - g_{1cor} = 62.000mm$

Gasket Contact Width(N):-

$$N = \frac{G_o - G_i}{2} = 25mm$$

Basic Gasket Width(b_o):-

$$b_o = \frac{N}{2} = 12.5mm$$

Effective Gasket Width(b):-

$$b = C_b * \sqrt{b_o} = 8.909mm$$

Gasket Reaction Diameter(G):-

$$G = G_o - 2 * b = 432.181mm$$

Basic Flange and Bolt Loads

Hydrostatic End load due to pressure[H]:

$$H = 0785 * G^2 * P = 274323.6$$

Contact Load on Gasket Surfaces [H_p]:-

$$H_p = 2 * b * \pi * G * m * P = 90481.312kg$$

Hydrostatic End Load at Flange ID [H_d]:-

$$H_d = \frac{P * B_{cor} * P}{4} = 783524.250kg$$

Pressure Force on Flange Face [H_t]:-

$$H_t = H - H_d = -509200.63kg$$

Operating Bolt Load [W_{m1}]:-

$$W_{m1} = max(H + H_p, 0) = 364804.906kg$$

Gasket Seating Bolt Load [W_{m2}]:-

$$W_{m2} = y * b * \pi * G = 21261.65kg$$

Required Bolt Area [A_m]:-

$$A_m = MAX\left(\frac{W_{m1}}{S_a}, \frac{W_{m2}}{S_a}\right) = 207.55cm^2$$

Flange Design Bolt Load, Gasket Seating [W]:-

$$W = Sa * \frac{A_m + A_b}{2} = 442588.94kg$$

Gasket Load for the Operating Condition [H_G]:-

$$H_G = W_{m1} - H = 90481.31kg$$

Distance to Gasket Load Reaction [h_g]:-

$$h_g = \frac{C - G}{2} = 103.9mm$$

Distance to Face Pressure Reaction [h_t]:-

$$h_t = \frac{C - (B_{cor} + G)/2}{2} = 29.35mm$$

Distance to End Pressure Reaction [h_d]:-

$$h_d = \frac{C + G_{1cor} - 2 * G_{0cor} - B_{cor}}{2} = -72.6mm$$

End Pressure Moment (M_d):-

$$M_d = H_d * h_d = -56884kg - m$$

Face Pressure Moment (M_t):-

$$M_t = H_t * h_t = -14947.kg - m$$

Gasket Load Moment (M_g):-

$$M_g = H_g * h_g = 9402.kg - m$$

Gasket Seating Moment (M_a):-

$$M_a = W * h_g = 45989.kg - m$$

Total Moment for Operation ($M_o p$):-

$$M_{op} = M_d + M_t + M_g = -62429.kg - m$$

Effective Hub Length, [h_o]:-

$$h_o = \sqrt{B_{cor} * g_{ocor}} = 214.551mm$$

Hub Ratio,

$$\frac{h}{h_o} = \frac{H_L}{H} = 00.093$$

Thickness Ratio,

$$\frac{g_1}{g_0} = \frac{g_{1cor}}{g_{ocor}} = 1.000$$

Flange Factors for Reverse Flange:

As per Appendix 2-7.2: F =0.9 Factor V per 2-7.3: V = 0.458 Factor f per 2-7.6: f = 1.262, α =1.664, β =1.886, γ =1.909, δ =2.621, λ =4.530

Longitudinal Hub Stress, Operating [S_{Ho}]:-

$$S_{Ho} = \frac{f * M_{op}/B_{cor}}{L * g_1^2} = 110.73MPa$$

Longitudinal Hub Stress, Seating [S_{Ha}]:-

$$S_{Ha} = \frac{f * M_a/B_{cor}}{L * g_1^2} = 81.57MPa$$

Radial Flange Stress, Operating [S_{Ro}]:-

$$S_{Ro} = \frac{\beta * M_{op}/B_{cor}}{L * t^2} = 25.50MPa$$

Radial Flange Stress, Seating [S_{Ra}]:-

$$S_{Ro} = \frac{\beta * M_a/B_{cor}}{L * t^2} = 18.78MPa$$

Tangential Flange Stress, Operating [S_{to}]:-

$$S_{to} = \frac{Y * M_o}{t^2 * B_{cor}} - Z * S_{Ro} = 38.53MPa$$

Tangential Flange Stress, Seating [S_{Ta}]:-

$$S_{Ta} = \frac{Y * M_a}{t^2 * B_{cor}} - Z * S_{Ra} = 28.38MPa$$

Average Flange Stress, Operating [S_{Ao}]:-

$$S_{Ao} = \frac{S_{Ho} + max(S_{Ro}, S_{To})}{2} = 74.63MPa$$

Average Flange Stress, Seating [S_{Aa}]:-

$$S_{Aa} = \frac{S_{Ha} + max(S_{Ra}, S_{Ta})}{2} = 54.98MPa$$

Bolt Stress, Operating [B_{So}]:-

$$B_{So} = \frac{W_{m1}}{A_b} = 120.83MPa$$

Bolt Stress, Operating [B_{Sa}]:-

$$B_{Sa} = \frac{W_{m2}}{A_b} = 7.04MPa$$

Reverse Weld Neck Flange Stress, Operating [S_{Vo}]:-

$$S_{Vo} = \left(\frac{M_o}{t^2 * B_{cor}}\right) - \left[Y - \frac{(2 * K^2 * (1 + 2/3 * t * e))}{(K^2 - 1) * L}\right] = 122.59MPa$$

Reverse Weld Neck Flange Stress, Seating [S_{Va}]:-

$$S_{Va} = \left(\frac{M_a}{t^2 * B_{cor}}\right) - \left[Y - \frac{(2 * K^2 * (1 + 2/3 * t * e))}{(K^2 - 1) * L}\right] = 90.31MPa$$

3.3.6 Tubesheet Calculation as per UHX-12

Tube Required Thickness under Internal Pressure (Tubeside pressure) :t_r

$$t_r = \frac{P*(D/2 - C.A}{S*E + 0.4*P} = 1.4186mm \tag{3.21}$$

Tube Required Thickness under External Pressure (Shellside pressure) :EMAP

$$EMAP = \frac{4*B}{3*\frac{D}{t}} = 1.451MPa \tag{3.22}$$

Results for ASME U-tube Tubesheet Calculations for Configuration a,

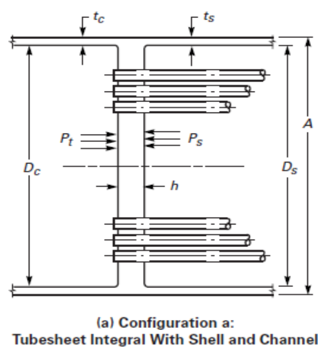


Figure 3.5: U-tube Tubesheet Configuration a

UHX-12.5.1 Step 1:

Compute the Equivalent Outer Tube Limit Circle Diameter $[D_o]$:-

$$D_o = 2*r_o + d_t = 663.5mm$$

Determine the Basic Ligament Efficiency for Shear $[\mu]$:

$$\mu = \frac{p - d_t}{p} = 0.286$$

Original Thickness : Minimum Required Thickness for Shear [H_{reqS}]:

$$H_{reqS} = 1/(4*\mu)*(D_o/(0.8*S))*|P_s - P_t| + C.A + C.A = 88.87mm \tag{3.23}$$

UHX-12.5.2 Step 2 :

Compute the Ratio [ρ_s]:-

$$\rho_s = \frac{D_s}{D_o} = 1.2178$$

Compute the Ratio [ρ_c]:-

$$\rho_c = \frac{D_c}{D_o} = 1.0912$$

Moment on Tubesheet due to Pressures (Ps, Pt) [M_{ts}]:-

$$M_{ts} = \frac{D_o^2}{16}*[(\rho_s - 1)*(\rho_s^2+1)*P_s - (\rho_c - 1)*(\rho_c^2+1)*P_t] = -807552.31kgf/cm^2*mm^2 \tag{3.24}$$

UHX-12.5.2 Step 3, Determination of Effective Elastic Properties :

Compute the Ratio [ρ]:-

$$\rho = \frac{l_{tx}}{h} = 0.9750(mustbe0 \leq \rho \leq 1)$$

Compute the Effective Tube Hole Diameter [d^*]:

$$d^* = max(d_t - 2t_t*(Et/E)(St/S)(\rho), d_t - 2t_t) = 13.74mm$$

Compute the Effective Tube Pitch [p^*]:-

$$p^* = \frac{p}{\sqrt{(1 - \frac{4*min[(A_L),(4*D_{op})]}{\pi*D_o^2}}} = 23.21mm$$

Compute the Effective Ligament Efficiency for Bending [μ^*]:-

$$\mu^* = \frac{(p^* - d^*)}{p^*} = 0.40798$$

E^*/E and ν^* for Triangular pattern from Fig. UHX-11.3.

$$E^*/E = 0.423911, \nu^* = 0.309068, E^* = 826526.kgf/cm^2$$

UHX-12.5.4 Step 4:

Compute Shell Coefficient [β_S]:

$$\beta_S = \frac{\sqrt[4]{12(1-v_s^2)}}{\sqrt{(D_s+t_s)*t_s}} = 0.0183mm^{-1}$$

Determine Shell Coefficient [k_s]:-

$$k_s = \beta_S * \frac{E_s * t_s^3}{6(1-\nu_s^2)} = 10406354.0kgf/cm*mm$$

Determine Shell Coefficient [Λ_s]:-

$$\lambda_s = \frac{6D_s}{h^3} * k_s * \left(1 + h*\beta_S + \frac{h^2*\beta_S^2}{2}\right) = 163992.56kgf/cm^2$$

Determine Shell Coefficient [δ_S]:-

$$\delta_S = \frac{D_s^2}{4*E_s*t_s} * \left(1 - \frac{\nu_s}{2}\right) = 0.0064433385mm/kgf/cm^2$$

Calculate Parameter [ω_S]:-

$$\omega_S = \rho_s * \beta_S * k_s * \delta_S * (1 + h*\beta_S) = 4786.97mm^2$$

Compute Channel Coefficient [β_C]:

$$\beta_C = \frac{\sqrt[4]{12(1-v_s^2)}}{\sqrt{(D_s+t_s)*t_s}} = 0.0087mm^{-1}$$

Determine Channel Coefficient [k_C]:-

$$k_C = \beta_C * \frac{E_c * t_c^3}{6(1-\nu_c^2)} = 549399680.0kgf/cm*mm$$

Determine Channel Coefficient [Λ_c]:-

$$\lambda_c = \frac{6D_c}{h^3} * k_c * \left(1 + h*\beta_C + \frac{h^2*\beta_C^2}{2}\right) = 3574989.0kgf/cm^2$$

Determine Channel Coefficient [δ_C]:-

$$\delta_C = \frac{D_c^2}{4*E_c*t_c} * \left(1 - \frac{\nu_c}{2}\right) = 0.0010128507mm/kgf/cm^2$$

Calculate Parameter [ω_C]:-

$$\omega_C = \rho_c * \beta_C * k_c * \delta_C * (1 + h*\beta_C) = 10793.75mm^2$$

UHX-12.5.5 Step 5:

Diameter ratio [K]:-

$$K = \frac{A}{D_o} = 1.266$$

Determine Coefficient [F]:-

$$F = \frac{1-\nu^*}{E^*} * (\lambda_s * \lambda_c * E * \ln K) = 3.51$$

UHX-12.5.6 Step 6:

Moment Acting on Unperforated Tubesheet Rim [M^*]:-

$$M^* = M_{ts} + \omega_C * P_t - \omega_S * P_s = 1140032.5 kgf/cm^2 * mm^2$$

UHX-12.5.7 Step 7:

Maximum Bending Moment acting on Periphery of Tubesheet [M_p]:-

$$M_p = \frac{M^* - \frac{D_o^2}{32} F(P_s - P_t)}{1+F} = 2096509.12 kgf/cm^2 * mm^2$$

Maximum Bending Moment acting on Center of Tubesheet [M_o]:-

$$M_o = M_p + \frac{D_o^2}{64} * (3+\nu^*) * (P_s - P_t) = -1823093.25 kgf/cm^2 * mm^2$$

Maximum Bending Moment acting on Tubesheet [M]:

$$M = max(|M_p|, |M_o|) = 2096509.12 kgf/cm^2 * mm^2$$

UHX-12.5.8 Results for Step 8:

Tubesheet Bending Stress at Original Thickness[σ]:-

$$\sigma = \frac{6 * M}{\mu^*(h - h'_g)^2} = 2141.1731 kgf/cm^2$$

The Allowable Tubesheet Bending Stress [σ_{All}]:-

$$\sigma_{All} = 2 * S = 2812.20 kgf/cm$$

Tubesheet Bending Stress at Final Thickness [σ]:-

$$\sigma = \frac{6 * M}{\mu^*(h - h'_g)^2} = 2810.82 kgf/cm^2$$

Required Tubesheet Thickness, for Bending Stress [H_{reqB}]:-

$$H_{reqB} = H + C.A + C.A = 105.14mm$$

Required Tubesheet Thickness for Given Loadings (includes CA) [H_{req}]:-

$$H_{req} = max(105.14, 88.87) = 105.14mm$$

UHX-12.5.9 Step 9:

$$|P_s - P_t| = |14.80 - 187.00| = 172.2 kgf/cm^2$$

Shear Stress check [τ_{limit}]:-

$$\tau_{limit} = 3 * S * \mu * h/D_o = 232.51 kgf/cm^2$$

Average Shear Stress at the Outer Edge of Perforated Region [τ]:-

$$\tau = \frac{1}{4 * \mu} * \frac{D_o}{h} * |P_s - P_t| = 833.11 kgf/cm$$

UHX-12.5.10 Results for Step 10:

Axial Shell Membrane Stress [$\sigma_{s,m}$]:-

$$\sigma_{s,m} = \frac{D_s^2}{4 * t_s * (D_s + t_s)} * P_s = 245.48 kgf/cm^2$$

Axial Shell Bending Stress [$\sigma_{s,b}$]:-

$$\sigma_{s,b} = \frac{6}{t_s^2} * k_s * \left[\beta_S * \delta_S * P_s + 6\frac{1 - \nu^*}{E^*}\frac{D_o}{h^3}\left(1 + \frac{h * \beta_s}{2}\right) * \left(M_p + \frac{D_o^2}{32} * (P_s - P_t)\right)\right]$$

$\sigma_{s,m} = 279.986 kgf/cm^2$

Shell Membrane + Bending Stress [σ_s]:-

$$\sigma_s = |\sigma_{s,m}| + |\sigma_{s,b}| = 525.47 kgf/cm^2$$

$\sigma_s \leq$ 1.5*Shell stress allowable at temperature.525.473 $\leq 1938.44 kgf/cm^2$

Hence it is safe.

Chapter 4

Support Design for Channel Shell

For horizontal vessels generally saddle supports are provided. For the present High Pressure Feedwater also saddle supports are provided for the Main Shell. But the case is, the saddles are furnished with rollers so that the shell can be removed for tube cleaning.

So for the channel shell also one saddle support can be provided. But the position of nozzle is such that the saddle cannot be provided over the length of the channel shell. Hence leg support needs to be designed.

Nonstandard leg support on the channel side is to be provided as per customer requirement. This method of providing support doesnot comply with the standard industry practice. So detailed Finite Element Analysis of the support structure and it's joint to respective shell is to be carried out.

4.1 Design calculation for leg support

4.1.1 Design Input

Description	Symbol	Unit	Value
O.D. of the vessel	Do	mm	840
Shell thickness	ts	mm	58
Width of Base plate	b	mm	310
Length of Base plate	l	mm	310
Thickness of base plate	tb	mm	36
No. of Anchor bolts per leg	Nb	nos.	2
Size of Anchor bolts	-	-	M20
Root Area of one bolt	Ab	mm^2	217.06
P.C.D. of Anchor bolts	PCDb	mm	790
MOC for Anchor bolt	-	-	SA516 Gr70
MOC for legs	-	-	SA516 Gr70
Thickness of pad	tp	mm	20
Length of Support	L	mm	900
No. of Leg supports	N	nos.	2

4.1.2 Leg properties

Description	Symbol	Unit	Value
Leg section	-	-	ISA 150x150x16
Leg Section Length (Along Y-axis)	A	mm	150
Leg Section Width (Along X-Axis)	B	mm	150
C/S Area	a	mm^2	4544
C.G. Distance	Cx	mm	43.38
C.G. Distance	Cy	mm	43.38
M.I @ X-X Axis	Ixx	mm^4	9629056.9
M.I @ Y-Y Axis	Iyy	mm^4	9629056.9
Sectional Modulus @ X-X	Zxx	mm^3	221932.4
Sectional Modulus @ Y-Y	Zyy	mm^3	221932.4
Radius of Gyration @ X-X	rxx	mm	46
Radius of Gyration @ Y-Y	ryy	mm	46
Slenderness ratio = 0.7*L/(min of rxx, ryy)	λ	-	13.69
Elastic Critical stress = $\pi^2 * E/\lambda^2$	fcc	Mpa	10585
A factor	n	-	1.4
Allowable axial comp. stress=0.6*fcc*fy/$[(fcc)^n + (fy)^n]^{1/n}$	Fac	Mpa	155.38
Allowable Bending Stress = 0.66*fy	Fbc	Mpa	171.6
Moment of Inertia @ X-X Axis = Ixx	Ixx’	mm^4	9629056.9
Total M.I @ axis perpendicular to F= 2*Iyy+2*Ixx’	ΣI	mm^4	38516227.6
Weight acting on Channel Shell	Ws	kg	5464

4.1.3 Wind Load Calculation

Wind Load calculations have been carried out according to IS 875 Part 3.

Probability Factor, K1 = 1

Size Factor, K2 = 2

Topography Factor, K3 = 1

Basic Wind Speed, Vb = 70 m/sec.

Design Wind speed, Vz = K1*K2*K3*Vb=1*1*1*70

hence Vz= 70 m/sec.

Wind Pressure, Pz= 0.6*Vz^2 = 2940 N/m^2

Wind resisting diameter, Do' = 1.2 * Do = 1.008 m

Wind resisting height, H = 1.25 m.

Hence wind resisting area, Af= 1.26mm^2

Force coefficient, Cf = 0.6.

Wind force at base, Fw = Pz*Cf*Af= 2222 N.

Wind moment at base, Mw = Fw * H/2= 1389 N-m.

4.1.4 Seismic Load calculation

Seismic Load calculation has been done according to IS 1893 part 4.

Seismic Zone Factor, Z= 0.36

Spectral Acceleration = Sa/g = 2.5

Damping = 2%

Multiplying factor from table 3 part 1, M_F = 1.4

Response reduction factor from table 9, R = 2

Importance factor from table 2, I = 2

Horizontal zone coefficient, Ah= (Z/2)* (Sa/g) *(R/I) = 0.63

Height of vessel above ground level = 1.25 m

Weight load = 1857* 9.81= 53582 N

Seismic force at base, Fe = Ah*W = 33757 N

Seismic moment at base, Me = Fe * H/2 = 21098 N-m.

4.1.5 Support Design

Maximum force, F = max of Fe,Fw = 33757 N

Maximum moment, M = max of Me, Mw = 21098 N-m

PCD of legs, PCD = Do+2*tp+2*(B-Cy) = 1.093 m

Eccentricity, e = ts/2 + tp + (B-Cy) = 0.155 m

Weight, W = 53582 N

Axial load per leg due to weight, Fv = W/N= 26791 N

Axial load at support A due to Moment, R1= 4*M/(N*PCD) = 38605 N

Axial load at support B due to Moment, R2 = 4*M/(N*PCD) = 38065 N

Base shear at support A, FsA= F*Iyy/ΣI = 8439 N

Base shear at support B, FsB= F*Ixx'/ΣI = 8439 N

Moment @ minor axis at base of support A,

$$MbA = (Fv + R1) * e + FsA * (3/4) * L = 18047N - m$$

4.1.6 Stress at the base of leg without Seismic/Wind load

Axial stress at support A,fac A = Fv/a = 5.89 MPa

Bending stress at support B,fbc B= Fv*e/Zyy = 18.7 MPa

Ratio, SRa A= fac A/Fac = 0.038

Ratio, SRb A= fbc A/Fbc = 0.109

A coefficient,Cm = 0.85

Combined stress ratio,

$$SRcA = SRaA + Cm * SRbB/(1 - facA/(0.6 * fcc)) = 0.012 + 0.0314 = 0.13056$$

.

Allowable Value = 1

Result- As $0.13056 < 1$**, Safe.**

4.1.7 Stress at the base of leg with Seismic/Wind load

Axial stress at support A,fac A = (Fv+R1)/a = 14.39 MPa

Bending stress at support B,fbc B= Mb A/Zyy = 81.31 MPa

Ratio, SRa A= fac A/Fac = 0.092

Ratio, SRb A= fbc A/Fbc = 0.47

A coefficient,Cm = 0.85

Combined stress ratio,

$$SRcA = SRaA + Cm * SRbB/(1 - facA/(0.6 * fcc)) = 0.0277 + 0.11 = 0.4914$$

.

Allowable Value = 1.33

Result- As $0.4914 < 1.33$**, Safe.**

4.1.8 Design of Base plate

Maximum compressive force in one leg,

$$Pcomp = W/N + 4 * M/(N * PCDb) = 9109 + 1455 = 80203N$$

Top shear in one leg, Ts = F/N = 16878.5 N

Bending moment at the base of leg, BM = Ts*L = 21098 N-m

Maximum bearing pressure below base plate,

$$BPmax = Pcomp/(b * l) + 6 * BM/b^3 = 5.07MPa$$

Allowable bearing pressure for concrete, fbp = 5.17 MPa

Projection of base plate beyond leg support,

$$b' = ((b^2 + l^2)^{0.5} - (A^2 + B^2)^{0.5})/2 = 113mm$$

Moment at face of leg due to bearing pressure,

$$Mbp = BPmax * b'^2 * b/2 = 10034518N - mm$$

Section modulus of Base plate, Zbp = b*tb^2/6 = 66960 mm^3

Bending stress in base plate,sb= Mbp/Zbp = 149.8 MPa

Allowable bending stress,σba = 0.66*fy = 171.6 MPa

Result- As $149.8 < 171.6$**, Safe.**

4.1.9 Design of Anchor Bolts

Tension in one leg due to seismic load, Te = 4*Me/(N*PCDb)-W/N = 26621 N

Tension in one leg due to wind load, Tw = 4*Mw/(N*PCDb)-W/N = -23274 N

Maximum tension in one leg, T = Max of Te and Tw = 26621 N

Maximum tensile force in one bolt, Tb = T/Nb = 13310.5 N

Shear force in one bolt, SHb =F/(N*Nb) = 8439.25 N

Required root area of one bolt

For tension, Ar = Tb/f = 5410/117 = 113.76mm^2

For shear, Ab = SHb/fs = 5735/87.75 = 96.17mm^2

Root Area of Bolt provided = 217.06mm^2, Hence Safe.

4.2 Finite Element Analysis of leg support for Channel Shell

4.2.1 Material of Construction

1. Shell : SA 516 Gr 70
2. Legs : SA 516 Gr 70
3. Pad : SA 516 Gr 70
4. Nozzle : SA 105

All the above listed materials are carbon steel with carbon content C≤0.3%. Hence the material properties as stated in ASME Section II Part D are as follows,

Modulus of Elasticity(E)= 198x10^3 MPa.

Poisson's Ratio(ν) = 0.3

Density(ρ) = 7750 kg/m^3.

4.2.2 Finite Element Model

A 3D CAD model,, of Channel Shell supported on legs is prepared. The model consists of Channel Shell, Nozzles, Legs, Base plate and Side plate. The dimensions of the model considered are as follows,

I.D. of Shell = 724 mm

Thickness of Shell = 58 mm

Length of Legs = 1200 mm

Thickness of Base Plate = 36 mm

Thickness of Side Plate = 20 mm

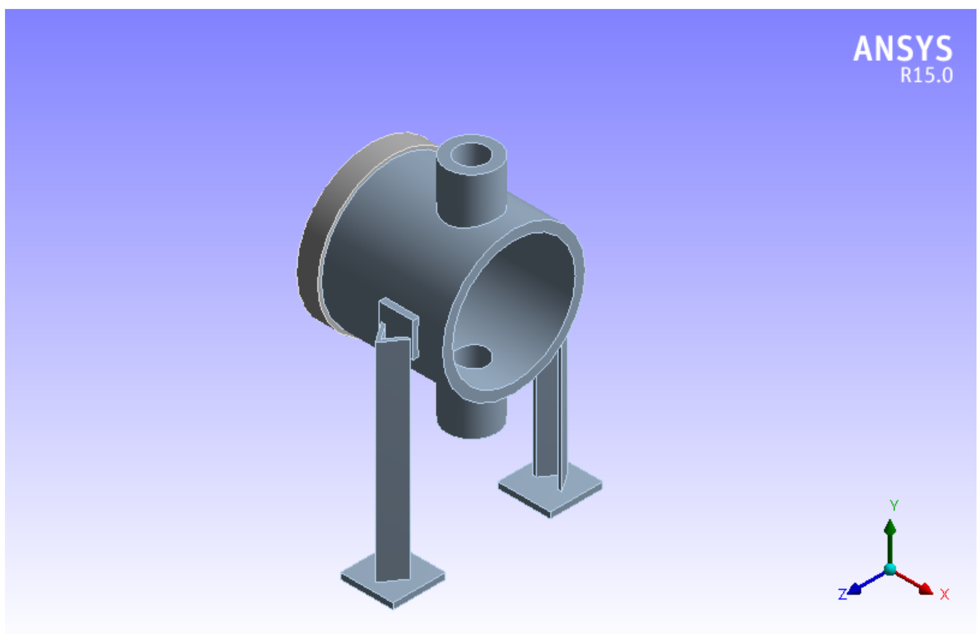


Figure 4.1: Finite Element Model

The CAD model is prepared such that weight of different components such as end cover, nozzles etc is considered. Tubesheet and tubes are not modelled but weight is considered during applied loading conditions.

4.2.3 Meshed Model in Ansys

Fig.4.2 shows the meshed model of the nozzle to the shell junction. Type of element employed for meshing are SOLID186, SOLID187.

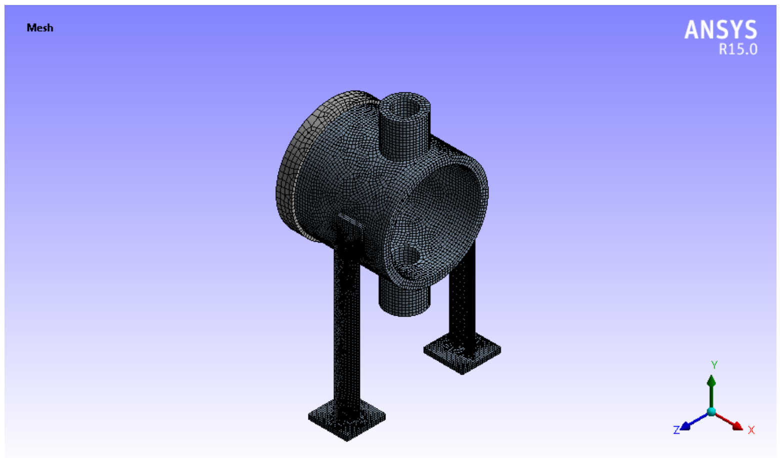


Figure 4.2: Meshed Model for Channel Support

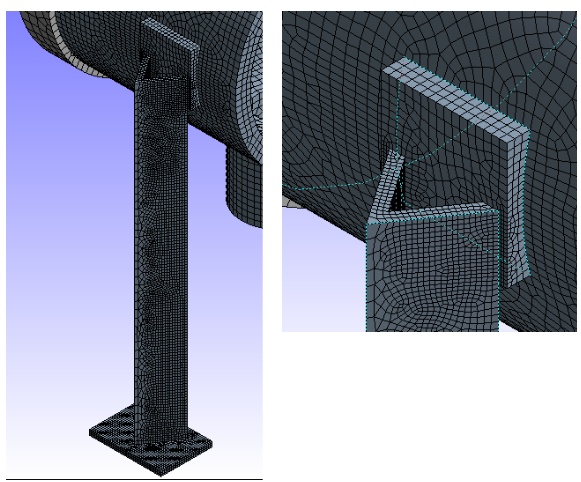
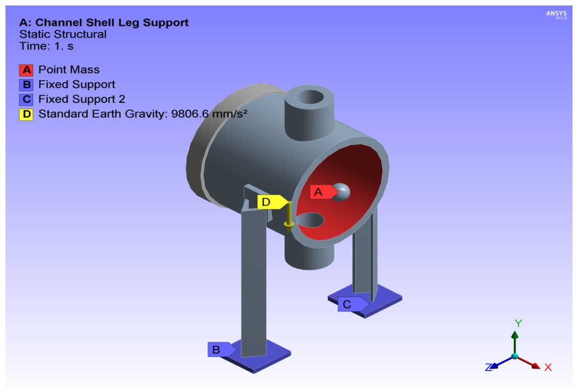


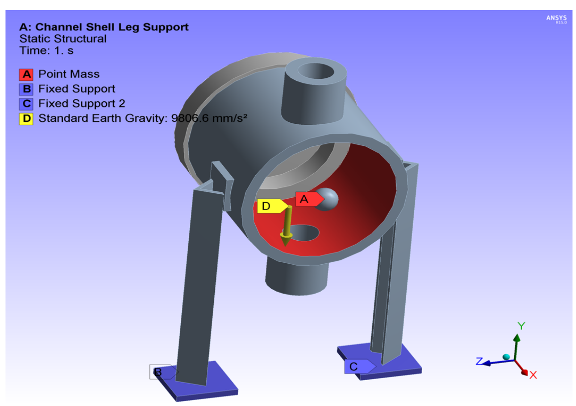
Figure 4.3: Meshed Model for Channel Support

4.2.4 Loads and Constraints Applied to the model

To replicate the loading condition encountered by the leg support; the bottom face of base plate is fixed, mass of tubesheet and tubes is calculated and equivalent point mass is applied to the model. For consideration of it's self weight, condition of Standard Earth Gravity is applied.



(a) label 1

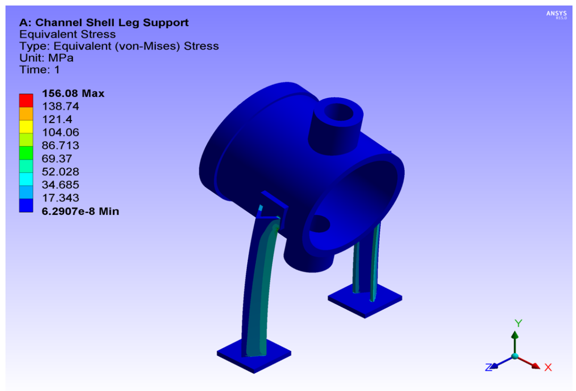


(b) label 2

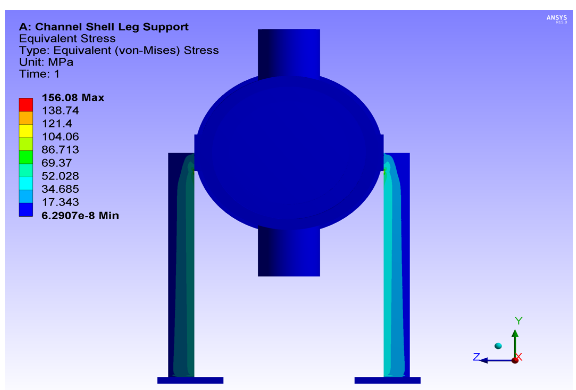
Figure 4.4: Loads and Constraints Applied to the model

4.2.5 Stresses generated

Fig 4.5 shows the distribution of stresses. Fig 4.5 shows the isometric view of the junction where maximum stress is generated. Fig. 4.5 shows the enlarged view of the same area maximum stress generated is 136.55 MPa.



(a) label 1



(b) label 2

Figure 4.5: Distribution of stresses for Channel Support

Chapter 5

Analysis of various components of HP Heater

5.1 Introduction ASME Section VIII Div.2 Part 5

ASME Section VIII Div. 2 Part 5 provides guidelines for design by analysis of components. Requirements for design by analysis are classified, as follows, based on protection against various failure modes.

1. Protection Against Plastic Collapse - It is the fundamental requirement for any component which is designed by analysis.

2. Protection Against Local Failure - It is required to satisfy the local failure criterion in addition to protection against plastic collapse. This requirement doesnot apply to the components for which the thickness, weld detail etc. has been determined by ASME Section VIII Div. 2 Part 4.

3. Protection Against Collapse from Buckling - This requirement is to be considered when the applied loads result in a compressive stress field.

4. Protection Against Failure from Cyclic Loading -This requirement is to be considered when the applied loads are cyclic.

Methods for evaluating Protection against Plastic Collapse will be discussed henceforth.

Protection Against Plastic Collapse:-

For protection against plastic collapse three alternate methodologies are available for analysis.Each of the method is discussed briefly,

1. Elastic Stress Analysis Method - Stresses are calculated by performing elastic analysis.The resulting stresses are categorized and compared with the allowable values. The allowable values have been established such that a plastic collapse will not occur.

2. Limit Load Analysis Method - A lower bound value for the limit load of a component is determined by calculation. Various design factors are applied to the limit load obtained, such that the onset of gross plastic deformation will not occur, and allowable load is established.

3. Elastic-Plastic Stress Analysis Method - Elastic plastic analysis is carried out considering applied load and deformation characteristics and collapse load is derived. Design factors are applied to this collapse load value and allowable load values are obtained.

5.2 Categorization of Stress

Stresses can be broadly categorized into following three types,

1. Primary Stress

2. Secondary Stress

3. Peak Stress

1. **Primary Stress** -

Primary stresses are caused by externally applied mechanical loads and are required to be in equilibrium with the external loads. These stresses are developed in load cases such as, if a mechanical force is applied on a pressure part such that the part reaches or crosses it's yield point and will continue to yield to failure if stress distribution doesnot take place. Hence Primary stresses are not self limiting.

Primary stresses can be further classified into following types,

Primary Membrane Stress (P_m)- The component of primary stress that is obtained by averaging the stress distribution through the thickness of the pressure vessel is referred to as the primary membrane stress. It is the most significant stress type .Design codes limit its value to the specified allowable material stress. An essential attribute of a primary membrane stress is that if the yield point is reached,causing the material to yield, a redistribution of stresses in the structure does not occur.

Primary Bending Stress (P_b)- Primary bending stress is produced in certain regions of a pressure vessel to resist externally applied loads. Some examples of Primary bending stress are bending stress in the crown of torispherical head due to internal pressure, bending stresses in tubesheet, flat cover bolted to a flange with a raised face gasket etc.

Local Membrane Stress (P_L) - Local membrane stress can be caused due to geometric discontinuity in the structure or locally applied external load on the structure or load discontinuity in the structure. Following rules should be pondered upon before qualifying the stress as local membrane stress,

(a) The stressed region will be considered local if the distance over which the membrane stress intensity exceeds 1.1 Sm, does not exceed $(rt)^{0.5}$, where r is the minimum mid-surface radius of curvature and t is the minimum thickness in the region considered.

(b) Regions of local primary stress intensity involving axisymmetric membrane stress distributions which exceed 1.1 Sm shall not be closer in the meridional direction than $2.5(rt)^{0.5}$, where r is defined as average of r1 and r2 and t is defined as average of t1 and t2 ; t1 and t2 are the minimum thicknesses of each of the regions considered, and rl and r2 are the minimum mid-surface radii of curvature of these two regions where the membrane stress intensity exceeds 1.1 Sm.

2. **Secondary Stress(Q)** -

Secondary stress is a normal or shear stress arising because of the constraint of adjacent material or by self constraint of the structure. The basic characteristic of a secondary stress is that it is self-limiting. Some examples of secondary stress are,

(a) Bending stress in a shell where it is connected to a head or to a flange.

(b) Bending stress in a shell or a head due to nozzle loads.

(c) Bending stress in the knuckle or shell at a head to shell joint.

(d) Bending stresses in a nozzle at the nozzle-shell junction due to internal pressure.

3. **Peak Stress (F)** -

   Peak stress is the highest stress in a region produced by a concentration (such as a notch or weld discontinuity) or by certain thermal stresses. Peak stresses do not cause significant distortion but may cause fatigue failure.

   Some examples of peak stress are,

   (a) Additional stresses developed at the fillet between a nozzle to shell junction due to internal pressure or external loads.

   (b) Thermal stress in the cladding of a tubesheet.

   (c) The non-linear stress at a local structure discontinuity, such as small nozzles and couplings attached to thin shells, caused by thermal mismatches.

5.3 Selection of Stress Classification Line

There are sections of structural discontinuity in pressure vessels where there are abrupt or sudden change in geometry, material or loading conditions applied. Generally highest stress in a component is obtained near such regions. Stress Classification Lines are located mainly along such regions of gross or local discontinuity. Following points should be checked after SCL is drawn and stresses along SCL are obtained.

1. Orientation of SCL should be normal to line of stress component of highest stress magnitude.

2. Hoop stress and meridional stress components should be monotonically increasing or decreasing across the SCL except for stress concentration.

3. Distribution of stress through thickness should be monotonically increasing or decreasing.

4. The shear stress distribution should be parabolic and/or the stress should be low relative to the hoop and meridional stresses.

5.4 Allowable limits for stresses

Allowable value for different stresses and also different combination of stresses is given in ASME Section VIII Div.2 Part 5. Following figure from the same clearly states the stress combinations to be considered and allowable values for those cases.

Nomenclature:-

P_m = Primary Membrane Stress

P_b = Primary Bending Stress

P_L = Local Primary Membrane Stress

Q = Secondary Stress

F = Peak Stress

S = Allowable stress value of material from ASME Section II Part D.

S_{PS} = Allowable limit on the primary plus secondary stress range

S_{PL} = Allowable limit on the local primary membrane and local primary membrane plus bending stress

S_a = Alternating stress obtained from a fatigue curve for the specified number of operating cycles.

S_y = Yield strength of the material.

where,

S_{PL} = 1.5 x S.

S_{PS} = 3 x S or 2 x S_y

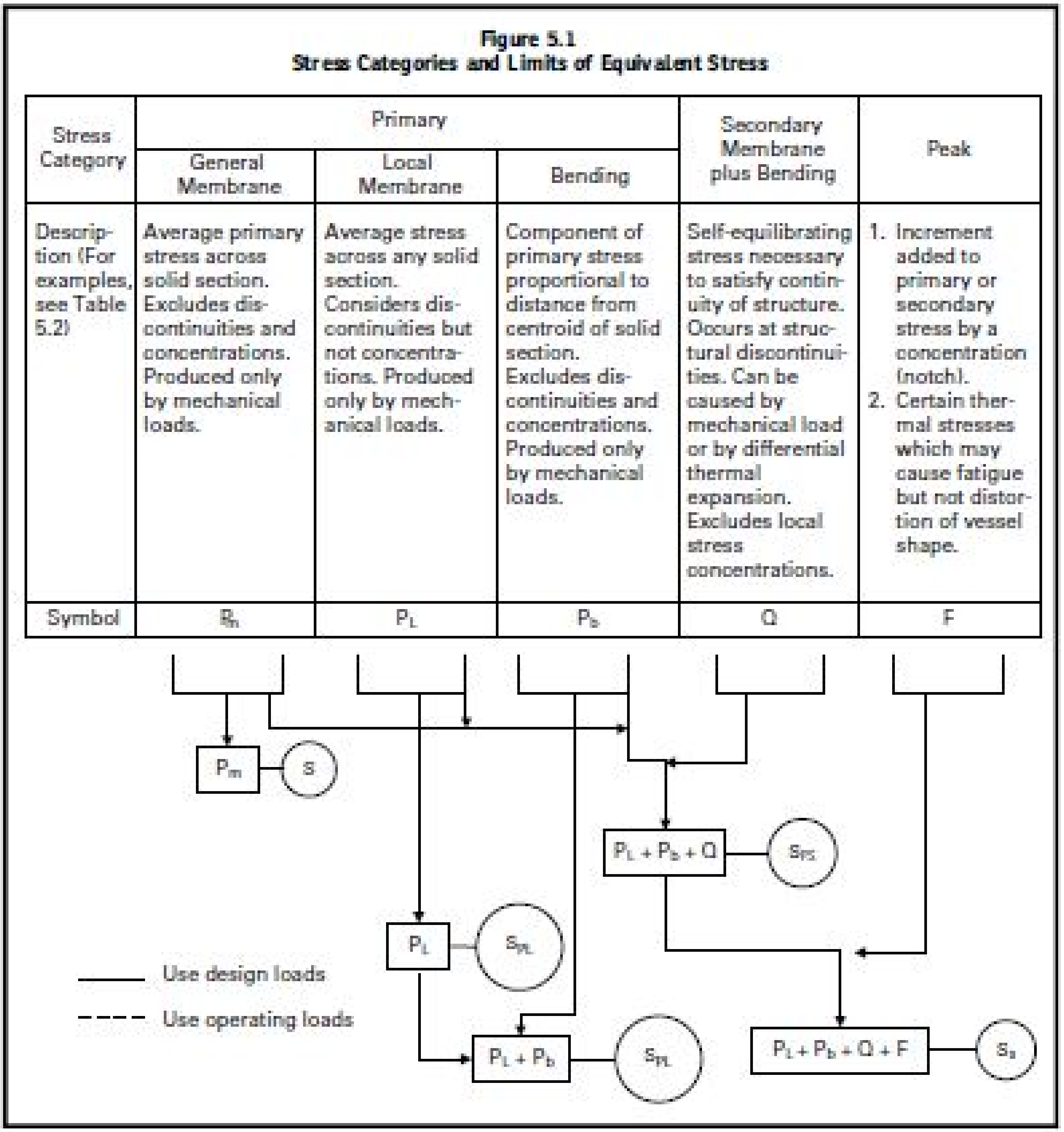


Figure 5.1
Stress Categories and Limits of Equivalent Stress

Stress Category	Primary: General Membrane	Primary: Local Membrane	Primary: Bending	Secondary Membrane plus Bending	Peak
Description (For examples, see Table 5.2)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentrations.	1. Increment added to primary or secondary stress by a concentration (notch). 2. Certain thermal stresses which may cause fatigue but not distortion of vessel shape.
Symbol	P_m	P_L	P_b	Q	F

Figure 5.1: Allowable Stress Values

5.5 Main Shell to Nozzle Junction

5.5.1 Design Input and Material of Construction

Material of construction

Shell: SA 516 GR.70

Nozzle: SA 105

Material properties: The physical properties used for various materials are listed below as per the material specification. The following temperature dependent properties of materials as given in ASME Section II, part D.

Elastic Modulus (E) = $2.05 * 10^5$ MPa

Poison's ratio (ν) = 0.3

Density (ρ) = 7850 kg/m^3

Allowable Stress:

$$\sigma_a = \frac{\sigma_{ultimate}}{Factor of Safety} = \frac{485}{3.5} = 138MPa$$

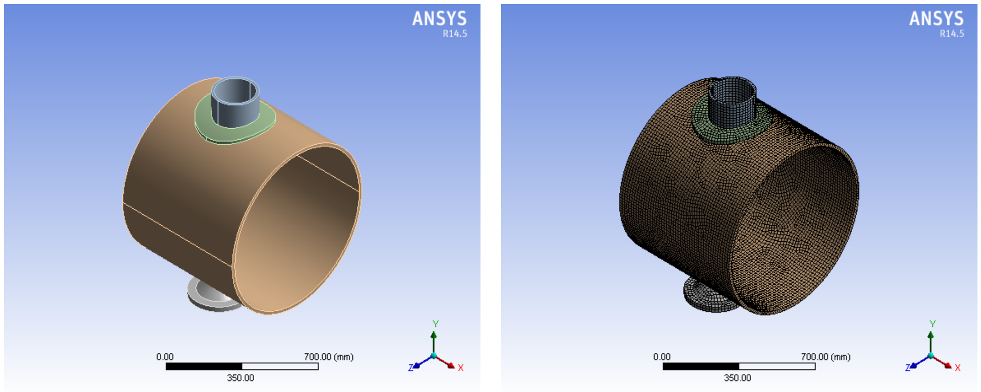
5.5.2 Finite Element Model

A 3D model,figure **??**, of Main Shell to nozzle is prepared. This model comprises shell made of SA-516 Gr. 70 material. and nozzle made of SA-105.The dimensions for model are as follows.

ID of shell= 808 mm

Thickness of shell = 10 mm

Length = 9200 mm.



(a) FE Model (b) Mesh Model

Figure 5.2: Meshed Model in Ansys

To properly simulate the result of stresses around shell to nozzle junctions the length of shell around the nozzle should be greater than $2.5 * \sqrt{R * t}$, where

R=average inside radius

t=thickness of shell.

Hence length of shell considered=700 mm.

Fig.5.6 shows the meshed model of the nozzle to the shell junction. Type of element selected is Brick 8 node 185.

5.5.3 Loads and Constraints Applied to the model

For evaluating the stresses due to structural loads,the side walls of the shell are fixed as it will be welded on both side in real condition as shown in Fig.5.3 Internal pressure is applied on the internal surfaces of the shell and nozzle. Following table shows the value of loads and moments to the nozzle.

Internal Pressure= 1.45 MPa

Nozzle	F_x	F_y	F_z	M_x	M_y	M_Z
N1	7340	5500	7340	4400	2930	3820
N4	2750	2060	2750	620	410	530

Table 5.1: Nozzle Loads

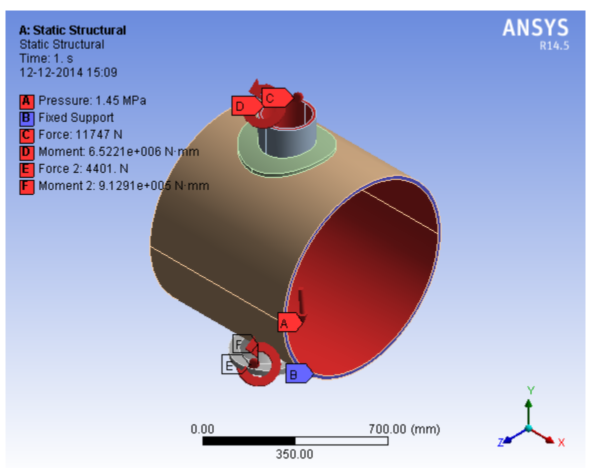
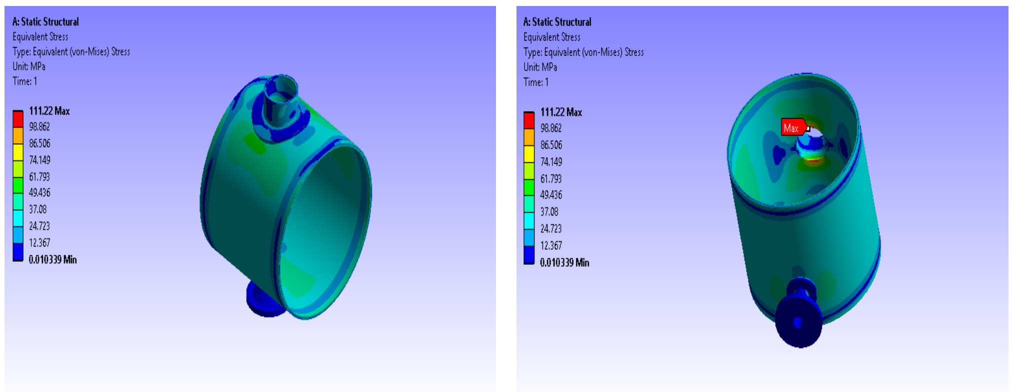


Figure 5.3: Nozzle Loads and pressure

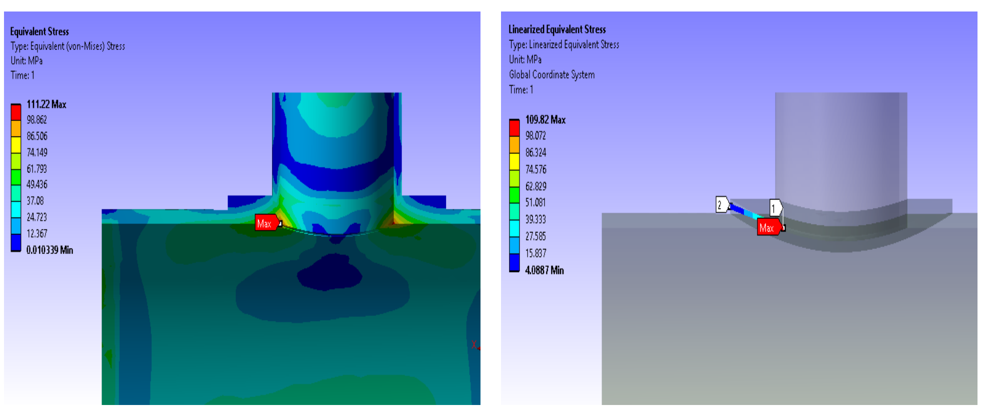
5.5.4 Stresses generated

Fig 5.4 shows the distribution of stresses. Fig 5.4 shows the isometric view of the junction where maximum stress is generated. Fig. 5.4 shows the enlarged view of the same area maximum stress generated is 111.32 MPa.



(a) (b)

Figure 5.4: Distribution of stresses



(a) Maximum Equivalent Stress (b) Stress Distribution along SCL

Figure 5.5: Distribution of stresses along SCL

Sr.No.	Length [mm]	Membrane [MPa]	Bending [MPa]	Membrane+Bending [MPa]	Peak [MPa]	Total [MPa]
1	0	40.12	49.715	89.711	24.34	109.82
2	1.92	40.12	47.644	87.642	21.202	105
3	3.84	40.12	45.572	85.573	18.236	100.43
4	5.76	40.12	43.501	83.504	15.434	96.106
5	7.68	40.12	41.429	81.435	12.569	91.667
6	9.6	40.12	39.358	79.367	9.2942	86.565
7	11.52	40.12	37.286	77.298	6.0783	81.291
8	13.44	40.12	35.215	75.23	1.5978	76.177
9	15.36	40.12	33.143	73.162	1.7266	72.943
10	17.28	40.12	31.072	71.094	2.9573	69.793
11	19.2	40.12	29	69.026	4.328	66.719
12	21.12	40.12	26.929	66.959	5.66	63.719
13	23.04	40.12	24.858	64.892	6.9244	60.788
14	24.96	40.12	22.786	62.825	8.1201	57.923
15	26.88	40.12	20.715	60.758	8.7055	55.398
16	28.8	40.12	18.643	58.692	8.7955	53.12
17	30.72	40.12	16.572	56.626	8.8627	50.843
18	32.64	40.12	14.5	54.561	8.9081	48.565
19	34.56	40.12	12.429	52.496	8.9337	46.287
20	36.48	40.12	10.357	50.431	8.942	44.006
21	38.4	40.12	8.2858	48.367	8.9366	41.724
22	40.32	40.12	6.2144	46.304	8.7464	39.654
23	42.24	40.12	4.1429	44.242	8.3838	37.798
24	44.16	40.12	2.0715	42.18	8.0091	35.962
25	46.08	40.12	3.54E-14	40.12	7.6217	34.146
26	48	40.12	2.0715	38.061	7.2214	32.35
27	49.92	40.12	4.1429	36.003	6.8083	30.576
28	51.84	40.12	6.2144	33.947	6.3827	28.824
29	53.76	40.12	8.2858	31.892	5.9198	27.095
30	55.68	40.12	10.357	29.84	5.2538	25.388

Sr.No.	Length [mm]	Membrane [MPa]	Bending [MPa]	Membrane+Bending [MPa]	Peak [MPa]	Total [MPa]
31	57.6	40.12	12.429	27.791	4.6129	23.681
32	59.52	40.12	14.5	25.746	4.0233	21.976
33	61.44	40.12	16.572	23.705	3.5269	20.275
34	63.36	40.12	18.643	21.67	2.4116	19.391
35	65.28	40.12	20.715	19.643	2.1377	17.955
36	67.2	40.12	22.786	17.625	2.3147	16.566
37	69.12	40.12	24.858	15.622	2.811	15.225
38	71.04	40.12	26.929	13.64	3.4578	13.934
39	72.96	40.12	29	11.688	4.1607	12.695
40	74.88	40.12	31.072	9.7858	4.8778	11.513
41	76.8	40.12	33.143	7.9683	5.386	10.267
42	78.72	40.12	35.215	6.3094	5.4711	9.0972
43	80.64	40.12	37.286	4.9703	6.0166	8.3102
44	82.56	40.12	39.358	4.264	6.7358	7.5109
45	84.48	40.12	41.429	4.4991	7.5658	6.7138
46	86.4	40.12	43.501	5.5572	8.4638	5.9437
47	88.32	40.12	45.572	7.0786	9.403	5.243
48	90.24	40.12	47.644	8.8268	10.435	4.6152
49	92.16	40.12	49.715	10.691	11.757	4.0887

The above table shows the variation of stresses along the SCL. All the stresses required to satisfy the requirement of ASME Section VIII Div. 2 can be met.

From the table above the maximum value of Membrane + Bending stress = 89.71MPa

Also, S_{PL} = 1.5 x S = 1.5 x 138 = 207 MPa.

Hence the Shell to nozzle junction is safe.

5.6 Channel Shell to Nozzle Junction

5.6.1 Design Input and Material of Construction

Material of construction Shell: SA 516 GR.70

Nozzle: SA 105

Material properties: The physical properties used for various materials are listed below as per the material specification. The following temperature dependent properties of materials as given in ASME Section II, part D.

Elastic Modulus (E) = $2.05 * 10^5$ MPa

Poison's ratio (ν) = 0.3

Density (ρ) = 7850 kg/m^3

Allowable Stress:

$$\sigma_a = \frac{\sigma_{ultimate}}{Factor of Safety} = \frac{485}{3.5} = 138 MPa$$

Internal Pressure= 18.34 MPa

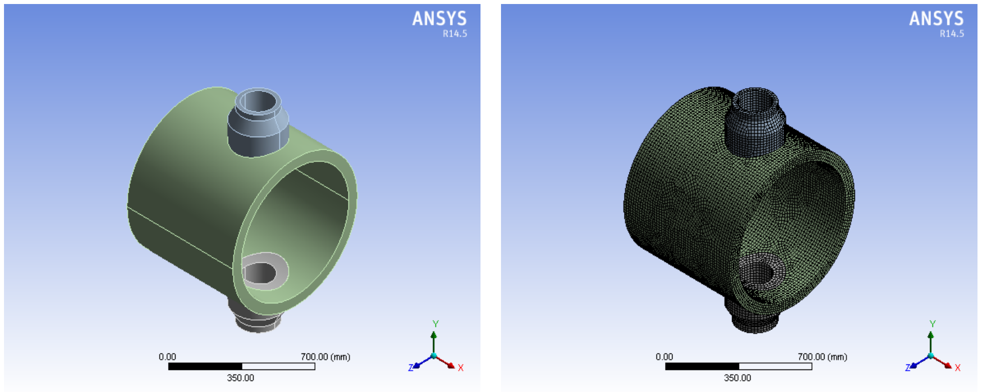
5.6.2 Finite Element Model

A 3D model,figure 5.6, of Channel Shell to nozzle is prepared. This model comprises shell made of SA-516 Gr. 70 material. and nozzle made of SA-105.The dimensions for model are as follows.

ID of shell= 724 mm

Thickness of shell = 58 mm, Length = 713 mm.

Fig.5.6 shows the meshed model of the nozzle to the shell junction. Type of element selected is Brick 8 node 185.



(a) FE Model

(b) Mesh Model

Figure 5.6: Meshed Model in Ansys

5.6.3 Loads and Constraints Applied to the model

For evaluating the stresses due to structural loads,the side walls of the shell are fixed as it will be welded on both side in real condition as shown in Fig.5.7 Internal pressure is applied on the internal surfaces of the shell and nozzle. Following table shows the value of loads and moments to the nozzle.

Nozzle	F_x	F_y	F_z	M_x	M_y	M_Z
N2	7340	5500	7340	4400	2930	3820
N3	7340	5500	7340	4400	2930	3820

Table 5.2: Nozzle Loads for channel shell

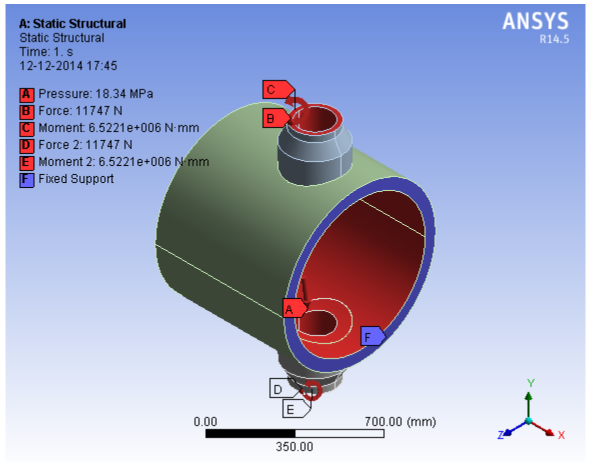
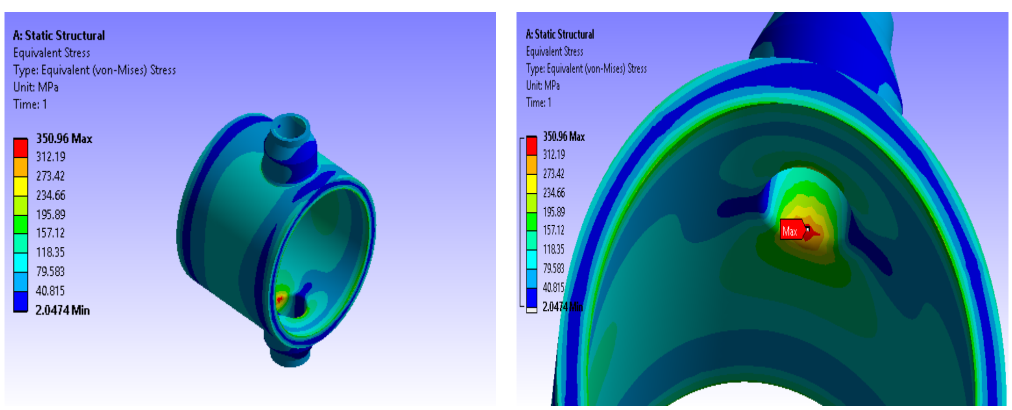


Figure 5.7: Nozzle Loads and pressure of channel side

5.6.4 Stresses generated

Fig 5.8 shows the distribution of stresses. Fig 5.8 shows the isometric view of the junction where maximum stress is generated. Fig. 5.8 also shows the enlarged view of the same area maximum stress generated is 350.96 MPa.



(a) (b)

Figure 5.8: Distribution of stresses in Channel shell

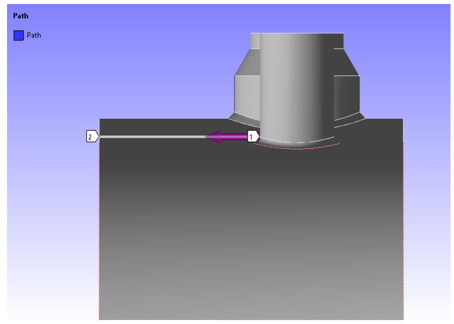
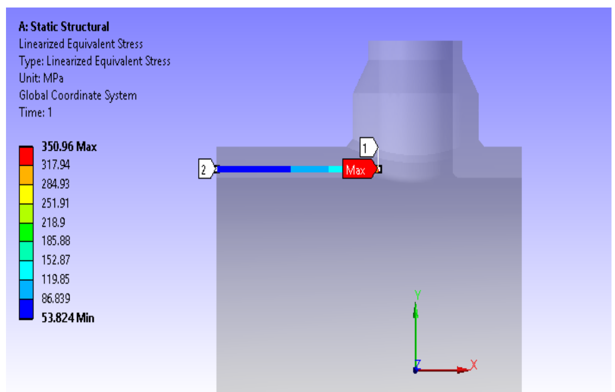
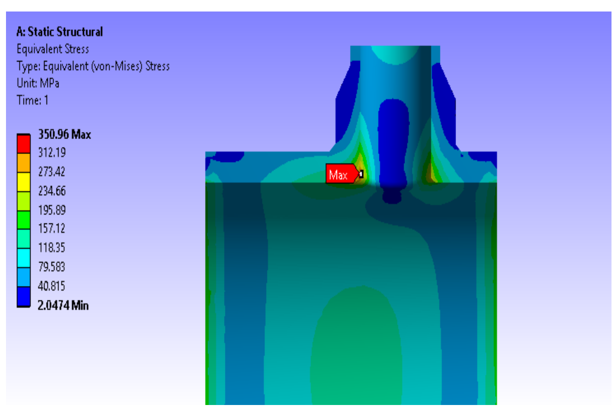


Figure 5.9: Stress Classification Line



(a) Maximum Equivalent Stress



(b) Stress Distribution along SCL

Figure 5.10: Distribution of stresses along SCL for channle shell

0	Length [mm]	Membrane [MPa]	Bending [MPa]	Membrane+Bending [MPa]	Peak [MPa]	Total [MPa]
1	0	97.72	113.4	203.44	154.95	350.96
2	7.8638	97.72	108.68	198.86	74.925	269.15
3	15.728	97.72	103.95	194.3	41.368	232.23
4	23.591	97.72	99.227	189.74	20.656	203.76
5	31.455	97.72	94.502	185.19	18.041	182.17
6	39.319	97.72	89.776	180.65	19.594	171.91
7	47.182	97.72	85.051	176.12	23.406	161.61
8	55.046	97.72	80.326	171.59	27.667	152.17
9	62.91	97.72	75.601	167.08	27.441	146.78
10	70.774	97.72	70.876	162.59	26.756	141.61
11	78.638	97.72	66.151	158.1	26.119	136.62
12	86.501	97.72	61.426	153.63	24.837	132.54
13	94.365	97.72	56.701	149.18	23.304	128.91
14	102.23	97.72	51.976	144.74	21.336	125.91
15	110.09	97.72	47.251	140.32	19.303	123
16	117.96	97.72	42.526	135.92	17.421	120.07
17	125.82	97.72	37.801	131.55	15.659	117.34
18	133.68	97.72	33.076	127.19	14.345	114.45
19	141.55	97.72	28.35	122.87	13.289	111.7
20	149.41	97.72	23.625	118.58	12.378	109.03
21	157.28	97.72	18.9	114.32	11.878	106.37
22	165.14	97.72	14.175	110.1	11.875	103.69
23	173	97.72	9.4502	105.92	12.366	100.74
24	180.87	97.72	4.7251	101.8	13.187	97.82
25	188.73	97.72	1.67E-14	97.72	14.14	94.69
26	196.59	97.72	4.7251	93.706	14.925	91.26
27	204.46	97.72	9.4502	89.761	15.619	88.084
28	212.32	97.72	14.175	85.894	16.315	84.934
29	220.19	97.72	18.9	82.118	16.839	81.524
30	228.05	97.72	23.625	78.445	17.196	77.842

0	Length [mm]	Membrane [MPa]	Bending [MPa]	Membrane+Bending [MPa]	Peak [MPa]	Total [MPa]
31	235.91	97.72	28.35	74.889	17.383	74.147
32	243.78	97.72	33.076	71.469	16.902	70.575
33	251.64	97.72	37.801	68.206	16.566	67.185
34	259.5	97.72	42.526	65.121	15.65	63.748
35	267.37	97.72	47.251	62.244	14.982	60.421
36	275.23	97.72	51.976	59.602	13.173	57.616
37	283.1	97.72	56.701	57.229	11.23	55.368
38	290.96	97.72	61.426	55.159	8.7108	54.069
39	298.82	97.72	66.151	53.428	6.2654	53.824
40	306.69	97.72	70.876	52.07	5.1467	53.829
41	314.55	97.72	75.601	51.114	5.6308	55.812
42	322.41	97.72	80.326	50.583	8.5735	58.88
43	330.28	97.72	85.051	50.491	13.585	63.973
44	338.14	97.72	89.776	50.839	19.372	70.168
45	346.01	97.72	94.502	51.62	24.254	75.758
46	353.87	97.72	99.227	52.813	28.637	81.134
47	361.73	97.72	103.95	54.392	29.113	82.783
48	369.6	97.72	108.68	56.324	19.453	72.737
49	377.46	97.72	113.4	58.575	14.682	62.171

The above table shows the variation of stresses along the SCL. All the stresses required to satisfy the requirement of ASME Section VIII Div. 2 can be met.

From the table above the maximum value of Membrane + Bending stress = 203.44MPa

Also, S_{PL} = 1.5 x S = 1.5 x 138 = 207 MPa.

Hence the Shell to nozzle junction is safe.

5.7 Finite Element Analysis of Tubesheet

5.7.1 Material of Construction

1. Tubesheet : SA 350 LF2

2. Tubes : SA 213 TP 304

All the above listed materials are carbon steel with carbon content C$\leq$0.3%. Hence the material properties as stated in ASME Section II Part D are as follows,

Modulus of Elasticity(E)= 198x10^3 MPa.

Poisson's Ratio(ν) = 0.3

Density(ρ) = 7750 kg/m^3.

5.7.2 Finite Element Model

An axisymmetric quarter model of tubesheet is prepared. As the configuration of tubesheet is such that the tube arrangement is symmetric about two axis.The dimensions of the model considered are as follows,

O.D. of Tubesheet = 840 mm

Thickness of Tubesheet = 110 mm

O.D. of Tubes = 15.875 mm

Thickness of Tube = 1.65 mm

No. of tubes = 628. As the model is quarter no. of tubes = 157

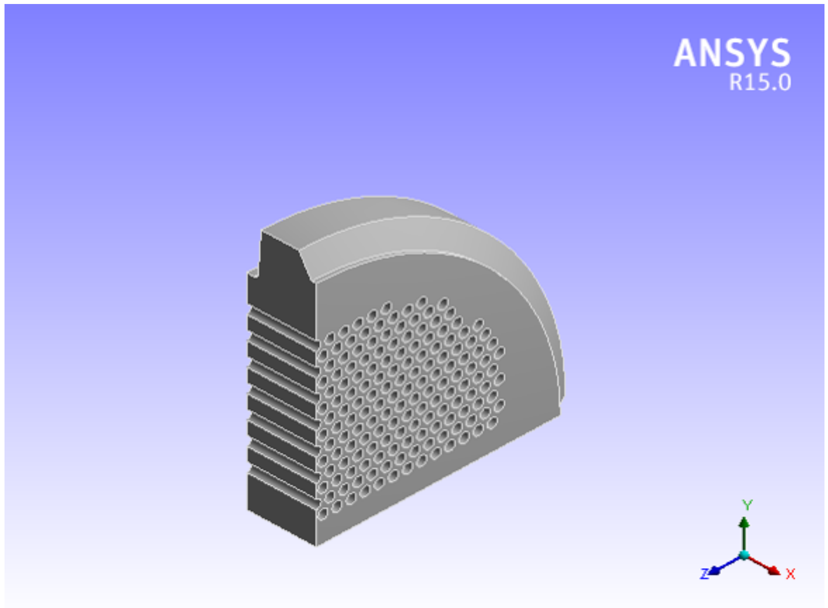


Figure 5.11: Finite Element Model

5.7.3 Meshed Model in Ansys

Fig.5.12 shows the meshed model of the Tubesheet. Type of element employed for meshing are SOLID186, SOLID187.

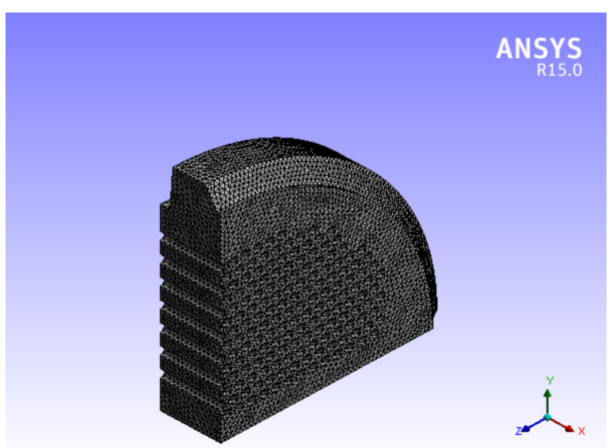
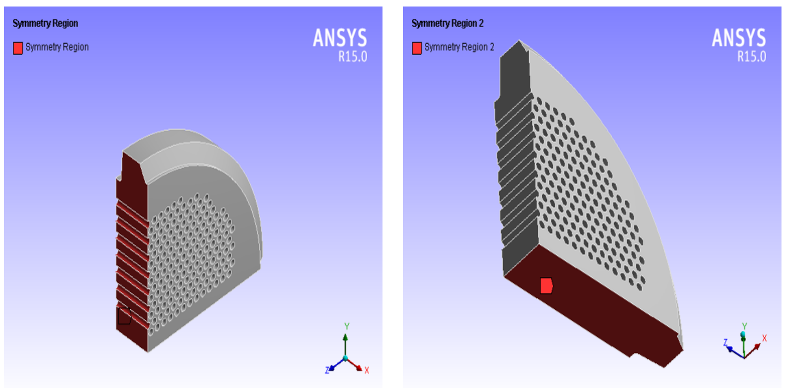


Figure 5.12: Meshed Model for Tubesheet

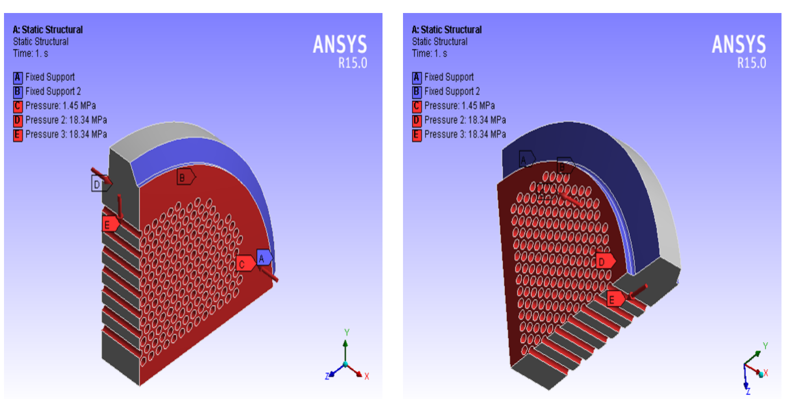
5.7.4 Loads and Constraints Applied to the model

To replicate the loading condition encountered by the tubesheet ; the bottom face and side face of tubesheet is given symmetric condition. Actual shape cut on the tubesheet for shell welding is also modelled, these faces are fixed as these faces are welded in real case scenario. Pressure is applied on respective faces where shell and tube side pressures are acting. Also the tube holes are pressurized as is the actual case.



(a) label 1 (b) label 2

Figure 5.13: Symmetry of the model

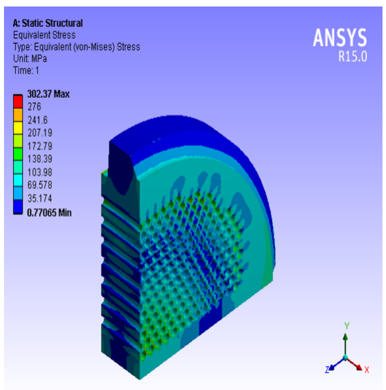


(a) label 1 (b) label 2

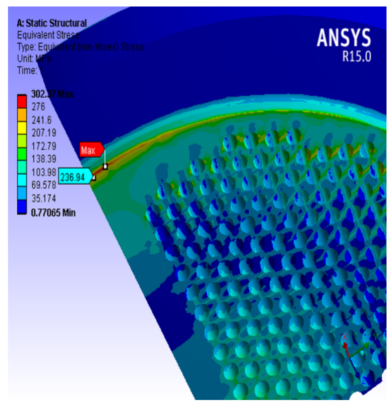
Figure 5.14: Loads and Constraints Applied to the model

5.7.5 Stresses generated

Fig 5.15 shows the distribution of stresses. Fig 5.15 shows the isometric view of the junction where maximum stress is generated. Fig. 5.15 shows the enlarged view of the same area where maximum stress generated is 302.37 MPa.Except for the stress concentration at the edge the resulting stress value is less than 276 MPa which is the limit for checking stresses in bending for tubesheet.



(a) label 1



(b) label 2

Figure 5.15: Distribution of stresses for Tubesheet

Chapter 6

Conclusion and Future Scope

6.1 Conclusion

Detail study of various type of Feedwaters and Heat Exchangers is done.For Mechanical Design of Heat Exchangers ASME and TEMA codes have been thoroughly studied.

The thickness for components like shell, head, channel shell,calculation of reinforcement of nozzle,design of tubesheet has been done according to ASME code.

The solid model of shell to nozzle junction has been prepared in CAD software. FE analysis for calculating the stresses at shell to nozzle junction has been carried out for all major nozzles.An axisymmetric model of Tubesheet is prepared and FE Analysis of the tubesheet with all the relevant boundary and load conditions is carried out. The stress values are below allowable limits of the material.

Leg support design is carried out for horizontal channel shell and the results are compared with the result of FE Analysis.

6.2 Future Scope

1. Limit Load analysis can be performed for the same heat exchanger so that, if possible, more intensity of load can be applied on it.

2. Thermal analysis of U-bend of tubes can be carried out as they are subjected to varying temperature conditions.

3. Detailed analysis of roller saddles and its guideways can be carried out.

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