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Heat Exchanger: Conical shell and Nozzle Reinforcement Effectiveness: Case Study

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ABSTRACT----

Conventionally heat exchangers and pressure vessels are designed using either ASME code or other international standards for the given design conditions. These designs ensure the safety, however the stresses at typical locations, such as a Kettle type conical shell and in the vicinity of nozzle reinforcement pads, are not easy to evaluate. In this project paper we have carried out mechanical design of such a shell and nozzle reinforcement pad of kettle type heat exchanger along with analysis, using finite element analysis (FEA) by using 'ANSYS' software. Due to weakening effect of nozzle and shell junction, reinforcement pad is required. Here we have carried out the analysis of eccentric cone which is welded with the shell on both sides with nozzle. Necessity of pad reinforcement is checked. The stresses induced in the shell and nozzle pad and the effect of pad on the stress intensity at nozzle shell junction is studied through FEA. The effect of various thicknesses of pad on stress intensity is also studied.

Key worlds: Heat Exchanger, reinforcement pads, FEA,

1. INTRODUCTION

Industrial heat exchangers are much less complex although the technological civilization makes them no less vital. Manufactured heat exchangers are found in facet of our life. Automobiles are equipped with heat exchangers called as radiators and all electronic and electrical equipment must be provided with heat exchanger for cooling.

2. ESSENTIAL COMPONENT OF A HEAT EXCHANGER

In shell and tube heat exchanger, the tubes are mechanically attached to tube sheets, which are contained inside a shell with ports for inlet and outlet fluid and gas. They are designed to prevent the liquid flowing inside the tubes from mixing with the fluid outside the tubes. Tube sheet can be fixed to the shell or allowed to expand and contract with thermal stress. In the later design an expansion blows is used or one tube sheet is allowed to float inside the shell. Heat exchanger studied here has fixed tube sheet without bellows and without floating head. The brief description of each component goes as follows:

2.1 Shell:

It is the main body of heat exchanger enclosing tubes, baffles (only for supporting), impingement plate, with inlet and outlet connections for working fluid. The shell is constructed either from pipe up to 24 inch or rolled and welded plate material.

The main shell is welded to tube sheet and channel shell is welded to the integral type flange.

2.2 Headers/Channel dished ends:

It forms the tube side of the shell and contains pass partition plates in multiple pass heat exchangers with inlet and outlet connections for working fluid, and a drain nozzle. The dished end is formed from steel plates, considering the appropriate forming allowances. A straight flange portion is kept to match channel shell while welding.

2.3 Tubes

The number and length of the tubes decide heat transfer area of exchanger. Drawn and seamless stainless tubes are welded to the tube sheet so as to provide for a double pass heat exchanger. The baffles are used for supporting the tubes as per construction code requirement, Radiographic tests, eddy current tests, pneumatic tests are conducted on tubes prior to its fit up.

2.4 Tube Pitch

Tube shall be spaced within the minimum centre to centre distance of 1.25 OD of tube.

Tube patterns:

A) Square pattern:

In removable bundle units when mechanical cleaning of tubes is specified. Tube lanes should be continuous.

B) Triangular pattern:

Triangular or rotated triangular pattern where mechanical cleaning is not specified.

2.5 Tube sheets (TS)

Tube sheets are used to support the tubes of heat exchanger at extreme ends.

TS are made from round plate piece of metal. Holes are drilled for the tube ends at precise locations and pattern relative to one another. Tube sheets are manufactured from the same range of materials as tubes. Tubes are attached to the tube sheets by pneumatic or hydraulic pressure by roller expansion if needed. Tube holes can drilled and reamed. This greatly increases tube joint strength.

The tube sheet is in contact with both fluids. Thus it has corrosion resistance allowances and metallurgical and electrochemical properties appropriate for the fluids and velocities. The tube hole pattern or pitch varies the distance from one tube to another as well as the angle of the tubes relative to each other and direction of flow. This allows the fluid velocities and pressure drop to be manipulated to provide the maximum amount of turbulence and tube surface contact for effective heat transfer.

2.6 Tube bundle

As the name implies it is a bundle of tubes held together by baffles and tie rod spacers.

The tube bundle is assembled and it is inserted in to the shell. At the end, tubes are welded to tube sheet and finally tube sheet to main shell and channel.

3 PROBLEM STATMENT:

To design a heat exchanger to meet the thermal and mechanical requirements. The scope here is limited to study the stress variation at conical shell and nozzle reinforcement pad. Only mechanical design is considered here. Design should be based on ASME standard. Given data is as follows:

3.1 MECHANICAL DESIGN CALCULATIONS:

Internal design Pressure of shell	= 20 kg/cm^2
Internal design Pressure of Tube	= 15.5 kg/cm^2
Design Temperature of Shell	= 90˚C
Design Temperature of Tube	= 160˚C

Static head of the fluid is added to the internal design pressure and design is made for a total pressure of 20.151 kgf/cm^2

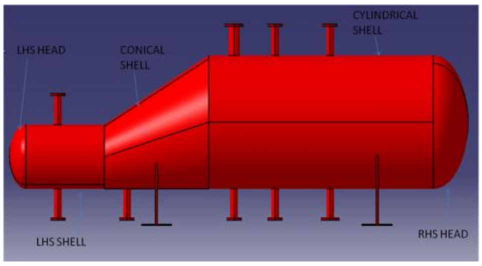

Fig1. Arrangement of the Heat Exchanger

4.DESIGN OF CONICAL SHELL

Material of Channel = SA240 304L
Inside Diameter of Channel = 1040 mm
Allowable Stress at 90^0 (S_o) = 1174.13 kgf/cm^2
Allowable Stress at Ambient (S_a) = 1174.13 kgf/cm^2
Efficiency of Longitudinal Seam = 0.85
Corrosion Allowance (C) = 1mm

Minimum Required Thickness of Conical Shell due to Internal Pressure[t]:

$$= \frac{PD}{(2\cos(\propto)) \times (S \times E - 0.6P)} + C$$

……………….UG 27 (c)(1)

$$= \frac{20.151 \times 1040}{2 \times \cos(30)\ [1174.13 \times 0.85 - 0.6 \times 20.151]} + 1$$

---------- taking angle as 30^o

= 12.7 + 1= 13.7 mm ≈ 14 mm

Maximum Allowable Working Pressure (M.A.W.P.):

$$= \frac{2SEt.\cos(\propto)}{D + 1.2 \times t \times \cos(\propto)}$$

$$= \frac{2 \times 1174.13 \times 0.85 \times 14 \times \cos(30)}{(1040 + 1.2 \times 14 \times \cos(30))}$$

= 22.9 Kgf/cm^2

Actual Stress at given Pressure and Thickness [S_{act}]:

$$= \frac{P(D + 1.2t \times \cos(\propto))}{2E \times t \times \cos(\propto)}$$

$$= \frac{20.151(1040 + 1.2 \times 14 \times \cos(30))}{2 \times 0.85 \times 14 \times \cos(30)}$$

= 1030.5 Kgf/cm^2

Sact < So, and M.A.W.P > Internal Pressure (P)
Therefore at t = 14 mm Design is Safe.
Actual thickness used = 16mm

4. DESIGN OF NOZZLE AND OPENING IN UNFIRED PRESSURE VESSEL/ HEAT EXCHANGER.

Openings are provided in the heat exchanger for functional requirement. They are required for

1. Inlet & outlet opening connections
2. Drain pipe connections
3. Pressure gauge connections
4. Safety device connections

Nozzles are formed or welded around these openings. Openings or hole causes discontinuity in the vessel wall which creates stress concentration in the vicinity of openings.
Higher stresses at the openings can be reduced by providing reinforcement in the vicinity of opening. This can be achieved either by one or more combinations of following methods:

1.Providing additional thickness to the shell wall itself near the nozzle.
2.Use of separate reinforcement pad attached to the heat exchanger wall covering an area surrounding the opening .

3.Providing additional thickness to the nozzle:
Most widely used method for designing reinforcement for nozzle is area for area method of compensation.

4.1 Area for area method of compensation

In this method the area of the material removed is compensated by providing additional area.

1. In the portion of shell as excess thickness.
2. In the portion of the nozzle outside the vessel as excess thickness.
3. In the portion of the nozzle inside the vessel as excess thickness.
4. In reinforcing pad (compensation ring)

Fig. 2 shows the reinforcement boundary limits. The area of the opening to be compensated is dn_c*trs which is the minimum area of the shell that is required to sustain the pressure. Requirement of the reinforcing pad is first established as below.

4.2 Calculation for reinforcement pad:

Let;

d_n= diameter of nozzle= 50 mm.

σ_{all}= allowable tensile stress for shell and nozzle material is =115.4 N/mm^2

ts= thickness of shell
=16 mm

tn =Thickness of nozzle = 8.7 mm

C= corrosion allowance for shell and nozzle
= 1 mm

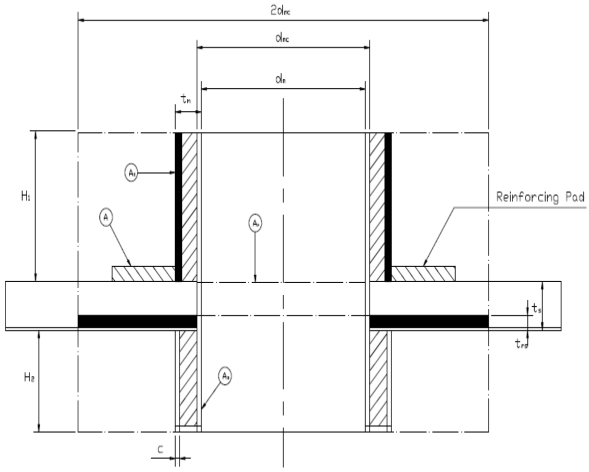

Fig2. Area for area method of compensation

P_i= internal pressure =1.96 N/mm^2

Required thickness of shell = 13 mm

1. Inner diameter of the nozzle in corroded condition

d_{nc}=dn+2c
= 50+2
=52 mm

2. Required thickness nozzle

$$t_{rn} = \frac{p_i.d_n}{2..S.\sigma_{all} - pi} \quad \text{---------UG37(a)}$$

t_{rn}= (1.96×50)/(2×1×115.7-1.96)

= 0.337 mm -----------------I

3. Height of nozzle

H_1=√ dnc(t_n-c)

Or

(Actual length of nozzle outside the heat exchanger)

= √52(8.7-1)

= 20 mm or 225 mm

Selecting smaller value,

Taking H1=20 mm

4. Height of nozzle projecting in side H_2= 0 mm

5. Estimation of compensation:

The addition area required is estimated as follows

1. Area for opening in corroded condition is

$A_r = d_{nc} \times t_{rs} = 52 \times 14$

$= 728\ mm^2$------------------ I

2. Area available for compensation(Aa)

I. The area of excess in shell thickness in the portion of shell ;

$A_1 = d_{nc}(t_s - t_{rs} - C) = 52(16-13-1))$

$= 104\ mm^2$

II. The area of excess in shell thickness in the portion of nozzle projecting outside of shell ;

$A_2 = 2H_1 (t_n - t_{rn} - C) = 2\times20\ (8.73-0.367-1)$

$= 294.5\ mm^2$

III. The area of excess in shell thickness in the portion of nozzle projecting inside of shell ;

$A_3 = 0\ mm^2$

So total area available for compensation

$A_a = A_1 + A_2 + A_3 = 104+294.5+0$

$= 398.5\ mm^2$ -----------------II

Required area for reinforcing pad;

$A = A_r - A_a$

$A = 728-398.5$

$A = 329.5\ mm^2$

Calculating the dimension for pad

$A = (d_{po} - d_{pi})\ t_p$

$329.5 = (104 - 67.4)\ t_p$

$t_p = 9.0 = 10$ mm

So taking thickness of pad = 10 mm

5. FE ANALYSIS PROCEDURE

A programmed FE pre-processor is used in order to standardize the analysis approach and significantly speed up the input process. This can also be used to further refine individual geometry. The post-processor provides a plot of stresses in the nozzle and shell utilizing the highest peak stress intensity indication to locate the line to plot. The post processor may also include the acceptance criteria plotted for reference in position to the indicated stresses. This will allow an immediate visual determination whether the loading and geometry is acceptable. As this procedure can be used for many geometry's and loading conditions, it can also be used to analyze many so-called "standard geometry's "in order to arrive at acceptable nozzle loading criteria for a range of conditions.

Methodology and F.E. Idealization:

5.1 System of Units:

The following system of units is followed for consistency throughout this analysis and results evaluation

Table 1: system units used

S.No	Parameter	Units	Conversion factor used in Analysis.
1.	Length	Millimetres	1.0
2	Force	Newton	1.0
3	Mass	Kg	1.0
4	Moment	N-mm	1.0
5	Pressure, Modulus of elasticity, stress	N/mm^2	1.0

5.2 Ansys Elements Used

The complete assembly is modeled using ANSYS Element Types as follows:

Table 1: Element types used

S.No	Element	ANSYS Element	Parts Modeled
1.	3-D Elastics shell	8 node shell 93	Shell plate, cone plate, nozzle and pad etc.
2	Rigid Constrain	MPC-184	Rigid Element

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5.3 Material Properties

Material: Austenitic Stainless Steel. Isotropic

Young's Modulus = 189860 N / mm ^{2}.

Poisson's Ratio = 0.3

Density: 8000 Kg / m^3.

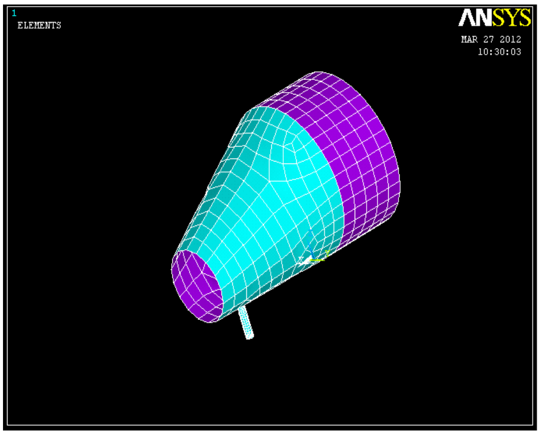

Figure 3: F.E.A. Mesh Model

5.4 Loading (Design Condition):

The Model has been analyzed for combinations of one or more of the following loads

1. Internal Design pressure = 1.96 N / mm ^{2}
2. External Design Pressure = 0.1013 N / mm ^{2}
3. Design Temperature = 90˚C
4. Hydrostatic Pressure = 2.54 N / mm ^{2}
5. Hydrostatic Temperature = Ambient

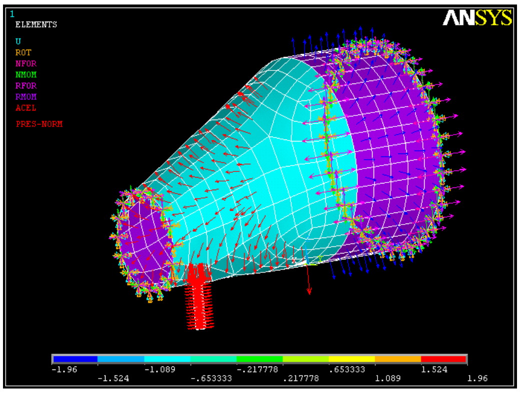

Fig.4Shell at loading condition

The figure 4 and 5 show the conical shell portion and the nozzle subjected to various loading conditions.

Conical shell as well as the nozzle is subjected to internal pressure of 1.96 N/mm^2 and the conical shell is connected to cylindrical channel on both ends. The shell side temperature is 90°C.

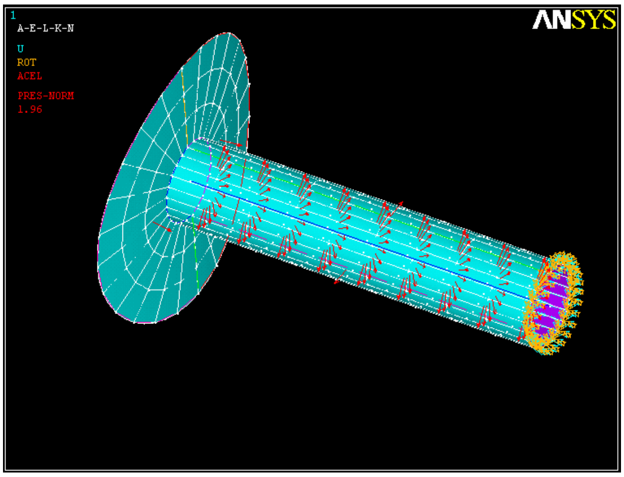

Fig.5 nozzle at loading condition

6. RESULTS INTERPRETATION AND CODE CHECKING:

Figure 6 shows the stress pattern in the conical shell without reinforcement pad for the nozzle. The stresses for the above mentioned load combinations are summarized and compared with code allowable limits for all critical parts in the Table no.3.The stress plots are also enclosed and referred.

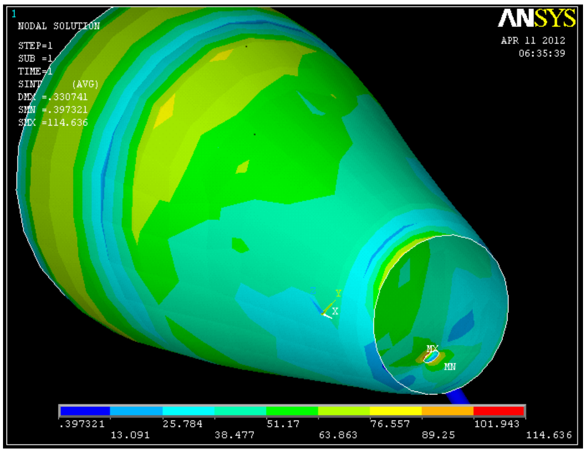

Fig.6 Stresses in shell without nozzle pad

Due to asymmetry in the conical shell, the stresses are not uniform and the stress variation is shown in figures 7 and 8. These stresses are difficult to calculate by hand calculations and therefore the stress variation plot becomes useful information.

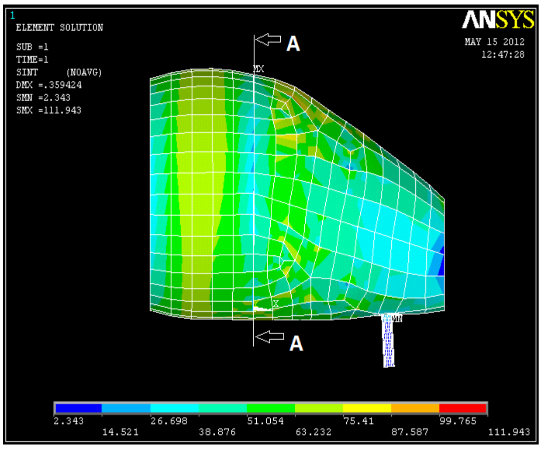

Fig.7 Stresses in shell without nozzle pad

The various locations in the conical shell at section AA are shown in fig. 8 below and the corresponding stresses in the shell are summarised in the table 3. As expected, the stresses at maximum discontinuity at point 1is maximum.

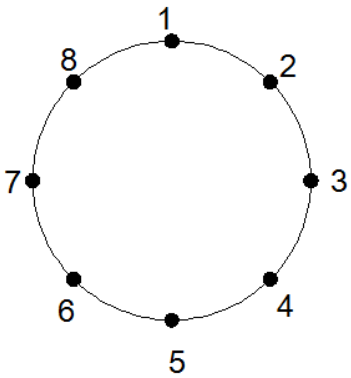

Fig.8 Node points at A-A section

Table 3: Stress at various node points.

Node Point	Stresses (N/mm^2)
1.	82
2	62
3	49
4	45
5	55
6	47
7	49
8	71

The stress pattern can be used to locate a nozzle opening or any other auxiliary opening at lowly stressed zone wherever choice exists.

Stress Pattern at Nozzle opening:

As per hand calculations shown earlier, reinforcement pads were found essential and 10 mm was found as the required thickness. The stress pattern was studied using FEA after applying the reinforcement pads of different thicknesses.

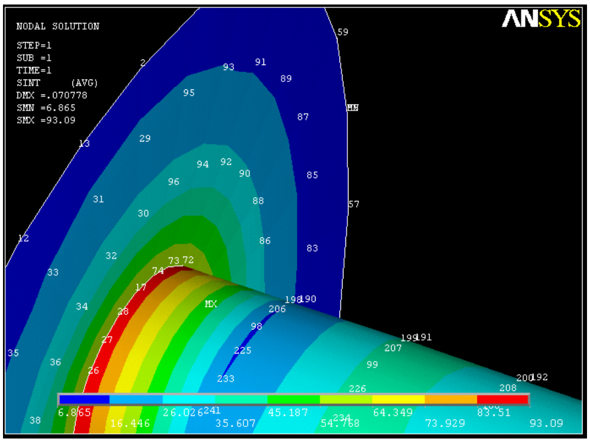

Fig.9 Stresses in nozzle pad and nozzle (at 10 mm)

Four node locations were identified as shown in figure 10 and the stress values were estimated by FEA at all these locations. The exercise was carried out for thicknesses varying from 4 mm to 16 mm keeping the outer diameter same.

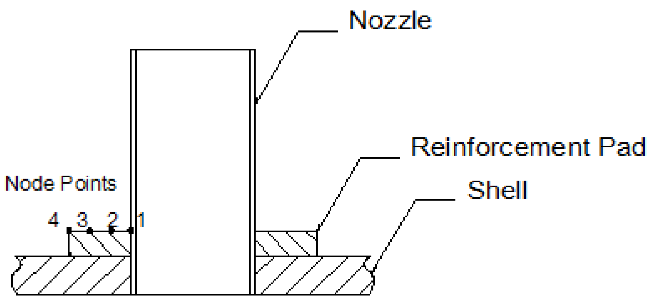

Fig.10 Nozzle Shell Junction

The observations are tabulated in table 4. It may be noted that the maximum stress at node 1 location reflects the stress peak in the nozzle rather than the stresses in the pad due to stress concentration.

Table 4: Stress Analysis Results.

Sr. No.	Thickness (mm)	Stresses at node Points (N/mm^2)			
		1	2	3	4
1.	4	141.5	47	32.6	24.2
2	6	121	40	28	20.9
3	8	105.5	35.5	24	17
4	10	93	31	20.5	15
5	12	83.3	27.5	18	13.5
6	14	75.6	24.4	16.5	12
7	16	69.43	21.8	14.7	10.8

It can be seen that the stresses at node locations of 2, 3, and 4 are significantly low for even smaller values of pad

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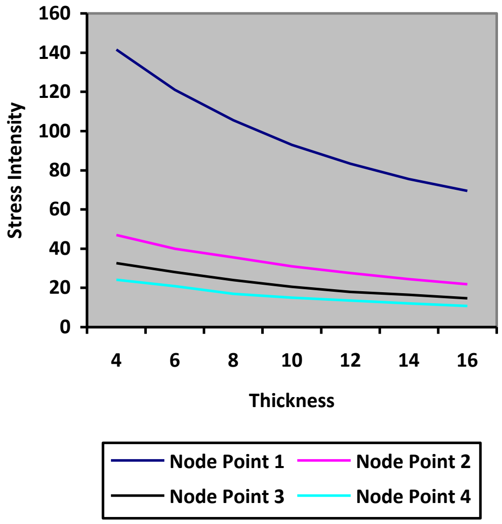
thicknesses such as 4, 6, or 8 mm. Moreover the effect is highly concentrated at nozzle point.



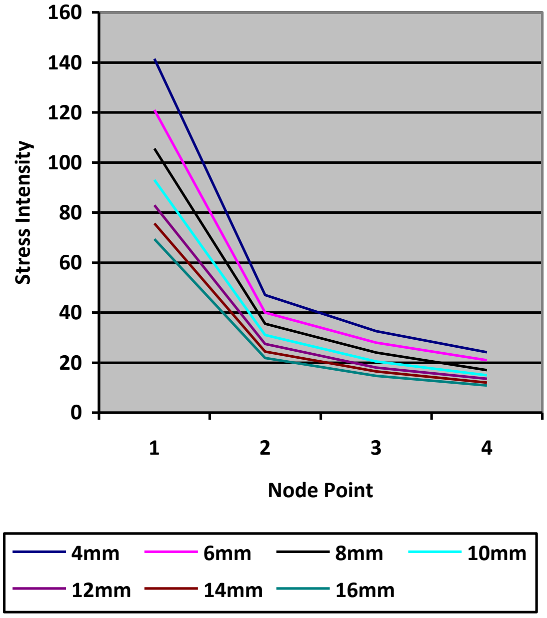


Figure 11: Plot of stress intensity

7. CONCLUSION:

The stress pattern has been studied at conical shell and the nozzle reinforcement locations.

The shell stress variation can be used to advantage by properly locating the openings in the shell at low stress areas.

Stress peak due to nozzle opening is observed mainly in the close vicinity of the nozzle.

8. REFRENCES

Books :

1. Heat Exchanger design by Mr.Young and Mr.Brownell, 1998 editions for integral type of the flange design (Pg.No 242-260)
2. Heat transfer by Donald Kern.
3. Heat transfer by Mr.Rajput,2002 edition for the study of heat transfer area
4. Preassure vessel deign manual by Mr. Dennies R.Moss,1987 edition
5. Heat exchanger design by Mr.Ganeshan and Mahajan ,1996 edition
6. The fundamentals of heat exchanger by Ramesh.K.Shah and Dusan.John 2003 edition
7. ASME SEC. VIII Div.1 Ede 2010

Websites:

1. http://www.tema.org
2. http://www.heatexchangeronline.com
3. http://www.en.wikipedia.org/wiki/heat_exchanger#shell_&_tube_heatexchanger