Computer Aided Design and Analysis of Counter Flow Heat Exchanger at Shell and Nozzle Junction

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Abstract

Shell and nozzle junction analysis of counter-flow heat exchanger which is manufactured at BHPV Ltd, Visakhapatnam is considered for the present work. The analysis is done by comparing the conventional design calculation from first principles with the results obtained from FEA package ANSYS. This paper presents structural analysis of the critical shell and nozzle junction by plotting different stresses developed. Design of the model is also obtained from the pressure-stress and thickness-stress graphs for the allowable working stress. Hence the final remedy to sustain these structural stresses at the critical junction is recommended by designing and providing necessary nozzle reinforcement pads. The work involves the knowledge of pure design principles from ASME and TEMA codes, conformability in mechanical engineering design, thorough knowledge and application of advanced FEA software ANSYS.

Keywords: Nozzle junction, heat exchanger, structural analysis, TEMA, ASME, FEA, ANSYS

1. Introduction

The design of a heat exchanger for any industrial purpose provides ample opportunity for collaborative effort among various engineers from diverse fields such as Design, Thermal, Fabrication and Quality Control. The design proposal is to be carefully analyzed by these engineers and the final specifications must be in accordance with standard codes applicable. The specifications for the heat exchanger design are often specified by the customers as per commercial & industrial requirements. The design often includes selection of an appropriate heat exchanger and its thermal, mechanical design specifications.

2. Design Criteria and Theoretical Analysis

2.1 Codes and Standards

A number of experienced heat exchanger manufacturers got together and formulated general rules popularly known as the TEMA standards. The first edition was issued in 1941. These have been since then expanded and issued at regular intervals.

ASME code section VIII DIV.1 is taken as the design and construction code along with its supplements. The heat exchanger codes viz. , TEMA and ASME DIV. 1 are applicable for design principles and constructional practices for vessels/heat exchanger up to 210 kg/cm^2 (3000psi).

2.2 Design Aspects

The mechanical design of a heat exchanger is carried out in accordance with the rules of codes and standards that are widely followed. The choice of a code or standard for the design and manufacture is normally made by the purchaser and his process licensor, guided by their experience on similar plants and this is strongly influenced by the country in which this experience has been gained. The consistent and painstaking work in the area done by ASME & TEMA had diffused into several countries of the world and as such these codes & standards are widely accepted. One of the benefits of the ASME code & TEMA Standards has been that several important terms are effectively and clearly defined and the rules & requirements are so orderly arranged which help in the design & more specially , in communication between the buyer, designer, fabricator & inspector.

2.3 Theoretical Analysis

Mechanical Design

The following parameters and values are used for required mechanical designing.

Material - Carbon Steel

Design Pressure, P - 2.27 N/mm^2

Allowable Stress, S - 104.1 N/mm^2

Shell radius (inside), R - 559 mm

Nozzle radius - 136.525 mm

Joint Efficiency, E - 1 (as per ASME VW-50)

Corrosion Allowance - 1.6 mm

Poisson's Ratio - 1/m= 0.3

Young's modulus, E - 2.1 e 05

Shell length considered - 750 mm

Nozzle length considered - 300 mm

Allowable tensile stress - f

Longitudinal Stress - f_0

Thickness of Cylindrical Shell under Internal

Pressure: Ug 27(ASME)

Formulae $t = P.R_0 / (SE-0.6\ P)$ (for $P \leq 0.385$ SE)

$P = 2.27$ N/mm^2

$R = 559$ mm

$E = 1.0$

$S = 104.1$ N/mm^2

$$t = \frac{2.27*559}{104.1*1-0.6*2.27} = 12.19 \text{ mm}$$

Considering corrosion allowance for water as fluid inside.

t = thickness + corrosion allowance

= 12.19 + 1.6 = 13.79 mm.

Considering Hydraulic test conditions

Pressure h = 1.5 * working pressure

= 1.5 * 2.27 = 3.405

$S_h = 0.9$ yield stress

$= 202.32$

$t = 3.405 * 559 / (196.76*1 - 0.6*3.405)$

$= 9.77$ mm

Required thickness=Greater (1) and (2)

$t = 13.69$ mm

Standard thickness = 14 mm

Stresses:

Hoop stress = $f_E = P.R/ t + 0.6 P$
$f = 2.27 * 559 /14 + 0.6 * 2.27$

$f = 91.99$ N/mm^2

Longitudinal Stress

$f_0 = ½ f$ **$= 45.99$ N/mm^2**

Max. Shear Stress at any point in the thickness of the material

$= \frac{f+p}{2}$ **$= 47.13$ N/mm^2** Von misses Stress =**225 N/mm^2** (as per ASME material properties)

Strains:

(Considered only for hoop stress as it is maximum)

Hoop Strain

Circumferential Displacement

$= I/E (f - f_0 * 1/m)*D$

$= 1/210000 (91.99 – 45.99 *0.03)* 1118$

= <u>0.416 mm.</u>

Strain = Stress/ E $= 91.99 /210000$

$= \mathbf{0.438 * 10^{-3}}$

Longitudinal Strain

Longitudinal displacement

$= I/E (f-f_0 * 1/m)$

$= I/E (45.99 - 91.99*0.3)*750$

= 0.0656 mm.

Strain = Stress/ E

$= 45.99 / 210000$

$= \mathbf{0.2196 * 10^{-3}}$

Thickness of Nozzle on Shell Under Internal Pressure:

Formulae $t = P.R_0 /(SE + 0.4P)$

$P = 2.27$ N/mm^2

R = 136.525 (From Appendix -C)

E = 1.0

S = 104.1 N/mm^2

t = $\frac{2.27*136.25}{104.1*1+0.4*2.27}$ = 2.95 mm.

Thickness after corrosion allowance

= t + C.A.

= 2.95 + 1.6

= 4.55 mm.

Standard Wall thickness = 9.55 mm.

Wall thickness required

= 0.875 * Sw +C.A.

= 0.875 * 9.5 +1.6

= 9.9125 (1)

But for such 60 min recommended

thickness = 12.7 (2)

Greater of (1) & (2) **t = 12.7 mm.**

Stresses:

Hoop Stress = f = **23.495 N/mm^2**

Longitudinal Stress = ½ f = f_0 = **11.747 N/mm^2**

Shear Stress = (f + p)/2 = **12.882 N/mm^2**

Strains:

Circumferential Displacement

= I/E (f –f_0 * 1/m)*247.65

= 0.0235 mm.

Circumferential Strain

= 220495 / 210000 = 0.118* 10^{-3}

Longitudinal Displacement

= I/E (f_0 – f *1 /m)*300

= 0.006712 mm.

Longitudinal Strain

= 11.747 / 210000

= **0.5593 * 10 $^{-4}$**

Calculation of Reinforcement: Ug 37(ASME)

Reinforcement padding is provided as per the recommendations for stress concentration on the shell and nozzle junction after thorough analysis by theoretical and experimental methods.

A = Reinforcement area required = dt_r F

d = Finished dia of circular opening (inside dia of nozzle)

t_r = Required thickness of a seamless sphere based on circumferential stress or a formal head.

t_n = Nominal thickness of nozzle wall , irrespective of product form , in

f = correction factor

f_{r1} = 1.0 for nozzle wall abutting the vessel wall.

Shell Side Inlet :

d = 10.75” -2(12.7 /25.4)

= 9.75” = 123.825 mm.

t_r = 0.5213”

T_n = 12.7 /25.4 = 0.5”

t_n = T_n – C.A. = 0.5” – 1.6 / 25.4

= 0.437”

A = dt_r = 9.75 * 0.5213

= 3279.13 mm^2

A_1 =Area available in shell (area in excess thickness in the vessel wall available for reinforcement)

= $d(E_1t - Ft_1) = 2(t + t_n)(E_1t - Ft_r)$

Use larger of (1) & (2)

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

= Joint efficiency obtained from table UW -12 when any part of the opening passes through any other welded joint.

t = Nominal thickness of the vessel wall.

F = Correction factor

Shell Side Inlet:

d = 9.75”

t_r = 0.5213”

t_n = 0.437”

E = 1.0

F = 1.0

F_{r1} = 1.0

T = 14/ 25.4 = 0.55118”

A_1 = d $(t-t_r)$ =0.2913 in^2

A_1 = 2 $(t+t_n)(t-t_r)$ =0.61 in^2

Adopted

A_1 = 0.2913 in^2 = 187.93 mm^2

$A_2 =$ Outside of Nozzle area

As total Area = A = $A_1 + A_2$

Hence A_2 = **2191.2 mm^2**

Hence the reinforcement is mostly concentrated on the outer side of Inlet of Shell, which is done as per standards followed.

The final parameters obtained from designing are:

Mechanical Design: Shell

Thickness of shell	=	14mm
Hoop stress developed due to pressure	=	91.99 N/mm^2
Longitudinal stress developed due to pressure	=	45.99 N/mm^2
Max. shear stress in the thickness of material	=	47.13 N/mm^2
Von misses (yield criteria) of the material	=	225 N/mm^2
Circumference displacement	=	0.416 mm
Circumference strain developed	=	0.438 e-3

Mechanical Design: Nozzle

Thickness of shell	= 12.7 mm
Hoop stress developed due to pressure	= 23.495 N/ mm^2
Longitudinal stress developed to pressure	= 11.747 N/ mm^2
Max. shear stress in the thickness of material	= 12.882 N/ mm^2
Von misses (yield criteria) of the material	= 225 N/ mm^2
Circumferential displacement	= 0.0235 mm
Circumferential strain developed	= 0.118 e -3
Longitudinal displacement	= 0.006712 mm
Reinforcement total area	= 0.5593 e -4

3. Modeling and Experimental Analysis

Modeling and experimental analysis is carried out for shell and nozzle junction as per the below provided data:

Material Properties

Material of the shell : SA 516 Gr 60 (ASME)
Material of the nozzle : SA 106 Gr (ASME)
Young's modulus : 2.1 e 05 N/mm^2
Poison's ratio : 0.3

Shell

Applied pressure : 2.27 N/mm^2
Internal radius : 559 mm
Thickness : 14 mm
Length considered : 750 mm

Nozzle

Applied pressure : 2.27 N/mm^2
Internal radius : 123.825 mm
Thickness : 12.7 mm
Length considered : 300 mm

Modeling of the nozzle junction along with considered length of shell as per the design specifications mentioned above is carried out in ANSYS pre-processor. Considering the complex geometry involved 3D solid model created with different boolean and extrude operations as shown in Fig.3.1. The meshed model with 3D tetrahedron element is presented in Fig.3.2. The loading and boundary conditions are shown in Fig.3.3

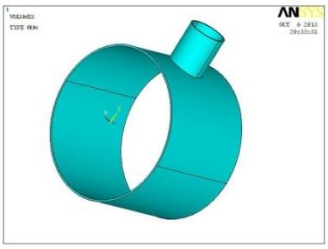

Fig.3.1

Fig.3.2

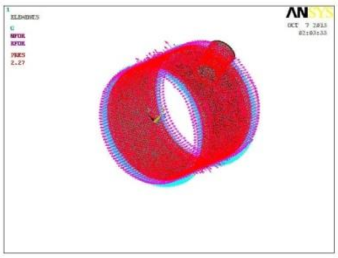

Fig.3.3

4. Discussion on Results

Experimental Analysis

It is observed from theoretical and experimental analysis that the results obtained by ANSYS is greater than the theoretical stress developed by major component shell and is greater by an amount which indicate the complex shell and nozzle combined stress. It can also be seen that the theoretical design criteria of the material is

Allowable working stress = 104.1 N.mm^2
Yield point stress = 225 N/mm^2

The experimental results shows that the maximum stress developed is 97.02 N/mm^2 (Fig.4.1)and Von misses stress (yield criteria) is 219 N/mm^2(Fig.4.2) which is within the safe design limits.

Hence the design is safe.

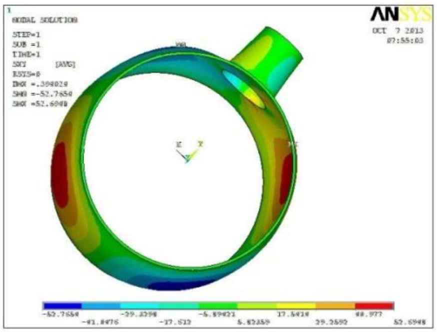

Fig.4.1

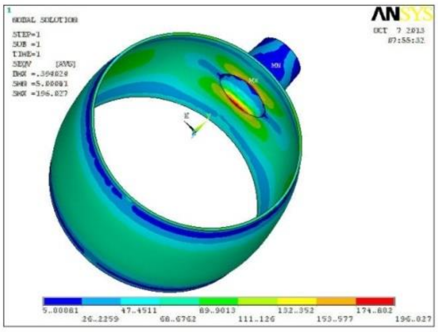

Fig.4.2

From the Pressure vs Stress graph (Fig.4.3 and Table.4.1) it can be seen that as the pressure increases the stress induced also increases till the yield point of 225 N/mm^2 with uniform slope. From graph it is evident that the design is optimal for pressure of 2.27 N/mm^2 for which maximum stress induced is 97.02 N/mm^2 against 129.472 N/mm^2 for P = 3 N/mm^2, which exceeds the allowable working stress of 104.1 N/mm^2 as calculated. The same is applicable for Von misses stress as with the increase of pressure from and onwards 3 N/mm^2 the stress exceeds the design limit of 225 N/mm^2.

It is found from the graph that for the designed allowable stress limit of 104.1 N/mm^2 the thickness is 13.1 mm.

Table 4.1

Pressure In N/Mm^2	**Shear Stress XY-Plane**	**Shear Stress YZ-Plane**	**Shear Stress XZ-Plane**	**Von Misses Stress**	**D$_{Max}$**	**Strain In X Direction**	
						Min	**Max**
2.27	52.6948	48.7021	58.0485	196.027	0.394024	-0.197e-O3	0.956e-03
3.00	69.6522	64.3664	76.3199	260.043	0.520336	-0.258e-03	0.001256
5.00	116.087	107.277	127.2	433.405	0.867226	-0.431e-03	0.002093
6.00	139.304	128.733	152.64	520.086	1.04067	-0.517e-03	0.002511
7.00	163.862	150.189	160.728	549.574	1.20695	-0.634e-03	0.002648

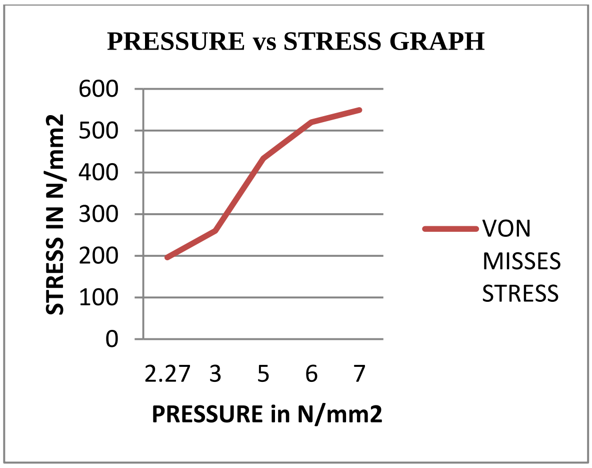

Fig.4.3 Thickness Vs Stress Graph

From the thickness vs stress graph (Fig.4.4 and Table.4.2), it can be seen that as we reduce the thickness of shell and nozzle the stresses developed increases and crosses the allowable working stress even for the next lowest practical size of heat exchanger (i.e. 12mm/ 12.77 mm). Also Von misses stress developed for the next lowest practical size is higher than the yield stress of 225 N/mm^2. Hence the design is optimal for 14 mm /12.7 mm shell and nozzle model.

It is fond from the graph that for the designed allowable stress limit of 104.1 N/mm^2 the pressure is 2.6 N/mm^2.

Table 4.2

Thickness Shell/Nozzle	**Shear Stress XY-plane**	**Shear Stress YZ-plane**	**Shear Stress XZ-plane**	**Von Misses Stress**	**$D_{Max.}$**	**Strain in X direction**	
						Min	**Max**
14/12.7	52.6948	48.7021	58.0485	196.027	0.394024	-0.197e-03	0.956e-03
12/12.7	59.431	61.5244	62.5323	215.977	0.449845	-0.306e-03	0.001044
10/9.27	71.6409	72.861	76.7297	273.703	0.543916	-0.326e-03	0.001271
8/7.8	93.0604	90.4173	103.434	326.915	0.67678	-0.349e-03	0.001538
6/6.35	122.818	117.266	132.224	422.753	0.89984	-0.397e-03	0.002005

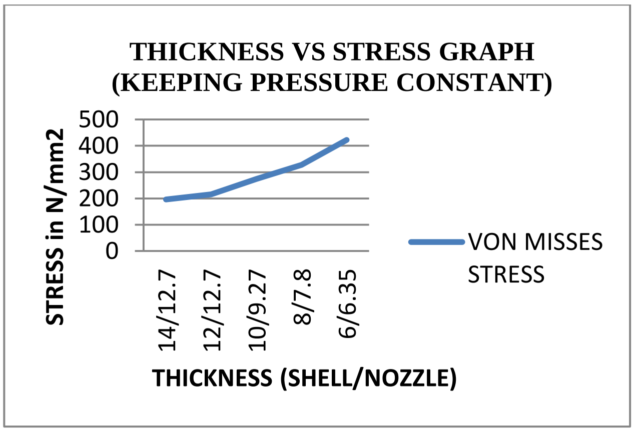

Fig.4.4

Recommendations

From structural analysis, it is seen that the maximum stress is developed at shell and nozzle junction and also from thermal distribution on surfaces of model; the maximum temperature distribution is seen at the same shell and nozzle junction. Hence the total FEM analysis projects the shell and nozzle joining junction as the most critical section which needs extra care to avoid failure due to structural and thermal stress.

In view of the above, it is recommended to provide nozzle reinforcement pads as per ASME Code of UG-37.1(Fig 4.5). The calculations for it have been already calculated under mechanical design (Table.4.3). Further due to flow induced vibration and stresses the heat exchanger is needed have very good supporting system designed as per the end use and the design codes.

Table 4.3
(at pressure 2.27 n/mm^2, thickness of shell/nozzle 14/12.7)

Parameter	Without reinforcement pad	With reinforcement pad
Von misses (N/mm^2)	196.027	163.151
Maximum shear stress in xy plane (N/mm^2)	52.6948	53.0844
Maximum deformation (mm)	0.394024	0.375390

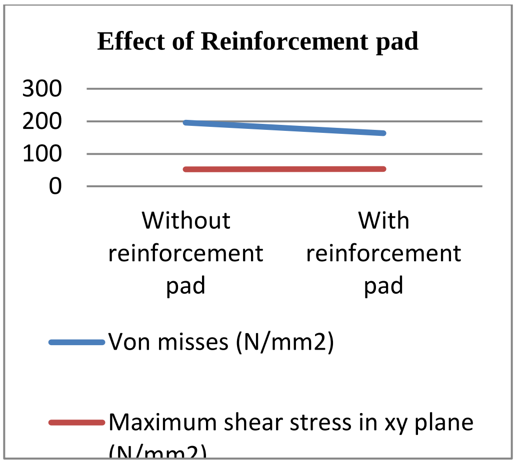

Fig.4.5(a)

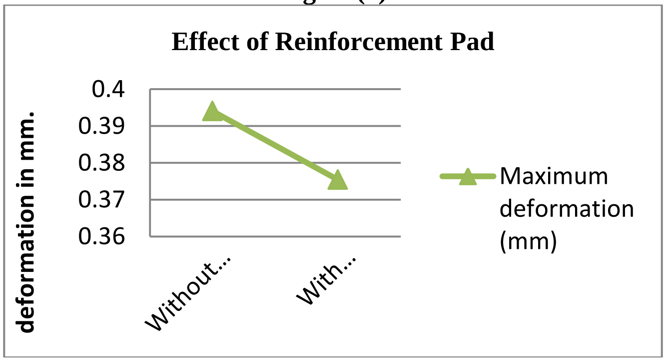

Fig.4.5(b)

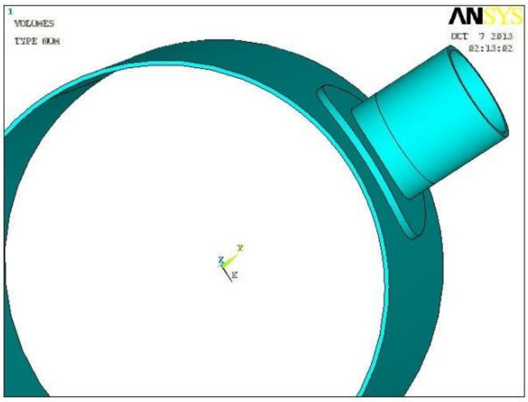

Fig.4.5(c)

5. Conclusions

a. Shell and tube heat exchangers, as already discussed in the earlier chapters are the most widely used kind of exchangers in the present day industry.
b. With ever increasing pressures and temperatures the demand for better performances and higher efficiencies are also increasing. As such shell and tube heat exchangers which are already prone to certain operational problems at lower and medium pressures and temperatures, require suitable modifications in construction to meet the demand.
c. As the project is focused on design and analysis of shell and nozzle junction. It is brought out that there are certain problems which effect the efficiency of shell and tube heat exchange such fouling, flow vibrations and gasket leakages etc.
d. As per the analyzed shell and nozzle of the heat exchanger, the stress values obtained are within and nearer limits. Also graphical shows that the obtained results for allowable working stress limit are pressure = 2.6 N/mm^2 for constant thickness and thickness =13.1 mm for constant pressure conditions. But due to practical norms from codes, the pressure is fixed by user himself for the required purpose and the thickness of plates for shell are manufactured in standard increments.
e. Hence in view of the above criteria , the optimal design obtained is of thickness = 14 mm/12.7 mm for shell and nozzle at given pressure of 2.27 N/mm^2. The results obtained from temperature distribution analysis shows that shell and nozzle junction is most vulnerable part which is to bear both structural and thermal max. stresses. Therefore, the design of heat exchanger is concluded with the design of reinforcement pad at the nozzle as per the TEMA and ASME code.
f. Hence the design and analysis shell and nozzle junction is within acceptable limits to meet the requirements of customer as well as the applicable standard codes.

6. Future Scope of Work

This work can be extended to study the thermal effects using temperature distributions and couple field analysis. The study can be extended to test various materials for optimum material utilization

7. Acknowledgements

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