INTERNATIONAL JOURNAL OF PURE AND APPLIED RESEARCH IN ENGINEERING AND TECHNOLOGY

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STATIC ANALYSIS OF HEAT EXCHANGER UNDER VARIOUS OPERATING CONDITION

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Accepted Date: 15/03/2016; Published Date: 01/05/2016

Abstract: This paper consist of static analysis of the effect of various loading conditions on the heat exchanger. During design, various aspects of the heat exchanger are need to be studied so that the design must able to resist the static as well as the dynamic forces acting on it. We are interested in the type of heat exchanger mostly used in petroleum industries. Thus we have carried out static analysis of a kettle reboiler which is basically Shell and tube type heat exchanger, in ANSYS 14.0 to justify the design.

Keywords: Heat Exchanger, ANSYS 14.0, meshing, operating conditions, pressure, deformation, stresses.

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How to Cite This Article:

Vrushabh R. Rathod, IJPRET, 2016; Volume 4 (9): 41-50

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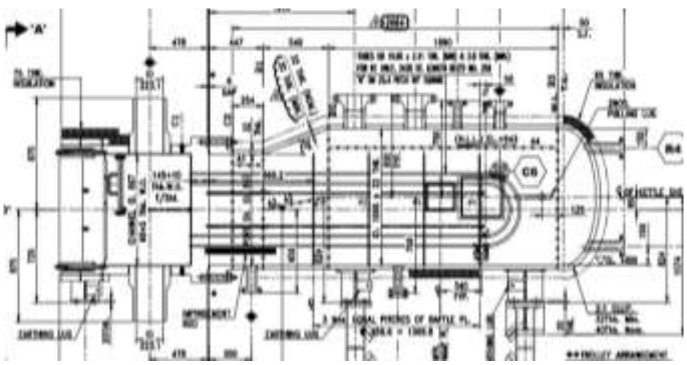
INTRODUCTION

A shell and tube heat exchanger is a class of heat exchanger designs. It is the most common type of heat exchanger in oil refineries and other large chemical processes, and is suited for higher-pressure applications. As its name implies, this type of heat exchanger consists of a shell (a large pressure vessel) with a bundle of tubes inside it. One fluid runs through the tubes, and another fluid flows over the tubes (through the shell) to transfer heat between the two fluids. The set of tubes is called a tube bundle, and may be composed of several types of tubes: plain, longitudinally finned, etc. In this paper we are concerned about the static analysis of shell and tube type heat exchanger under different loading conditions. To do the same we have first obtain a design of a shell and tube type heat exchanger from industry and start analyzing it.

1. **Design:**



2. **Material Properties:**

Mechanical & Thermal properties of Material Used for Various Components						
Component	Material	Design Temp. (° C)	Modulus of Elasticity (Kg/mm²)	Allowable Design Stress at Design Temp. (Kg/mm²)	Thermal Conductivity (kgf-mm)/(mm-s-C)	Coeff. of Thermal Exp. mm/(mm-C
Shell , Dish End, Cone, Port Shell, Saddle Wrapper plate	SA 516 Gr70	286	18931	13.92	4.4781	0.13140 E-04
Shell flanges	SA 266 Gr2	286	18931	13.349	4.4781	0.13140 E-04
Nozzle Neck & Nozzle Flanges (Shell Side)	SA 105	286	18824	13.349	4.4781	0.13140 E-04
Nozzle Neck & Nozzle Flanges (Channel Side)	SA 182 F22 Cl3	418	18824	11.671	4.4781	0.13140 E-04
Channel Barrel	SA 182 F22 Cl3	418	18568	13.64	3.7674	0.13140 E-04
Tubesheet (Equivalent Tubesheet model)	SA 336 F22 Cl3	418	E* = 8755	13.337	3.7674	0.13140 E-04
Saddle support Channel side	SA 387 Gr22 Cl2	418	18568	NA	3.7674	0.13140 E-04
Saddle supports - Shell side	SA 36	286	17021	NA	4.4781	0.17628 E-04

3. Stress Calculation:

Assumptions-

1) In static conditions, a heat exchanger acts as a pressure vessel.

2) Shell and channel of the heat exchanger are the main components to be analyzed.

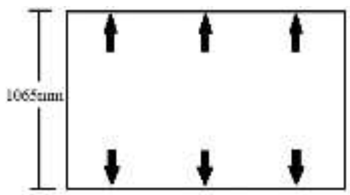
3) As t<0.1D the shell and channel of heat exchanger is considered as thin pressure vessel.

4) In thin cylinders the principle stresses are

a) Circumferential or Tangential stresses

b) Longitudinal stresses.

For shell side:



Material SA516 Gr70

Internal pressure = P_i = 61 Kgf/cm^2

= 5.98MPa

Corrosion Allowance = 3 mm

Thickness = 32-3 = 29 mm

Internal Diameter of Cylinder = 1065 mm

Tangential Stress = $\frac{\text{PiDi}}{2t}$

$= \frac{5.98*1065}{2*29}$

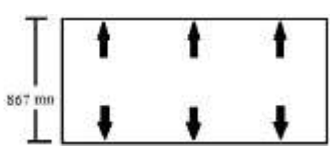
= 109.80 N/mm^2

Longitudinal Stress = $\frac{\text{PiDi}}{4t}$

$= \frac{5.98*1065}{4*29}$

= 54.90 N/mm^2

For Channel side:



Material SA182 F22

Internal pressure = P_i = 169.3 Kgf/cm^2

= 16.60 N/mm^2

Thickness = 68 mm

Internal Diameter of Cylinder = 867 mm

<u>Calculations:</u>

Tangential Stress = $\frac{\text{PiDi}}{2t}$

$= \frac{16.60*867}{2*68}$

= 105.82 N/mm^2

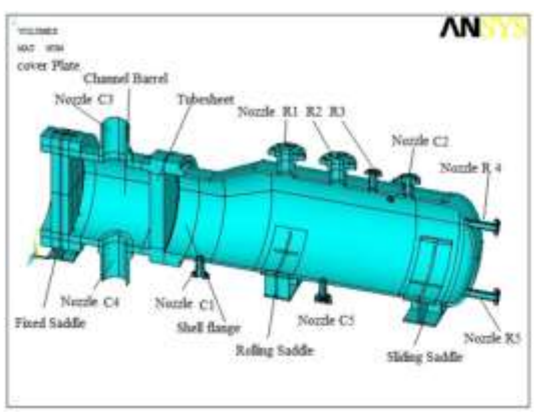
Longitudinal Stress = $\frac{\text{PiDi}}{4t}$

$= \frac{16.60*867}{4*68}$

= 52.91 N/mm^2

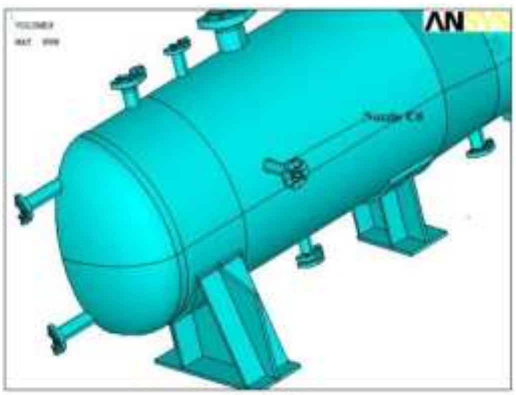
4. Extent of Modelling:

Half sectional symmetric geometry consisting of Shell region, Torricone, Dish End, Shell Side Nozzles, Channel, Solid Tubesheet, Channel Side Nozzles, Fixed, rolling & Sliding saddles are considered for geometric modeling. Tubes (U shape) are not modeled in Analysis as it nullifies the net effect of internal pressure & thermal expansion acting inside it (Geometry is U shape). In other words U shape geometry for tubes doesn't account for any resultant expansion effect. Hence it was not considered for analysis. Solid Tubesheet with Equivalent Modulus of Elasticity (E*) as calculated form ASME Section VIII Div1, UHX was considered during analysis. Corroded model was considered during Analysis.

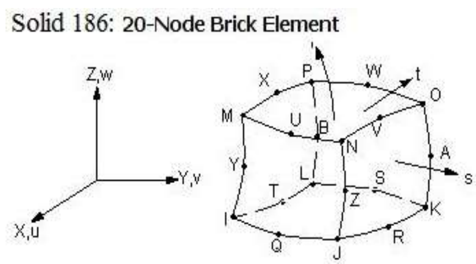


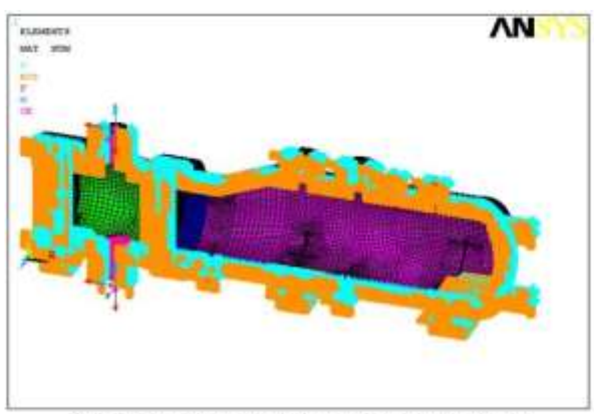
5. Meshing:

FEA uses a complex system of points called nodes which make a grid called a mesh. This mesh is programmed to contain the material and structural properties which define how the structure will react to certain loading conditions. Nodes are assigned at a certain density throughout the material depending on the anticipated stress levels of a particular area. Regions which will receive large amounts of stress usually have a higher node density than those which experience little or no stress. Points of interest may consist of: fracture point of previously tested material, fillets, corners, complex detail, and high stress areas. The mesh acts like a spider web in that from each node, there extends a mesh element to each of the adjacent nodes. This web of vectors is what carries the material properties to the object, creating many elements.



Nozzle C6





Symmetric Boundary condition at symmetric faces of Heat Exchanger

6. Boundary Conditions:

- Design internal pressure on Shell side & Tube side.

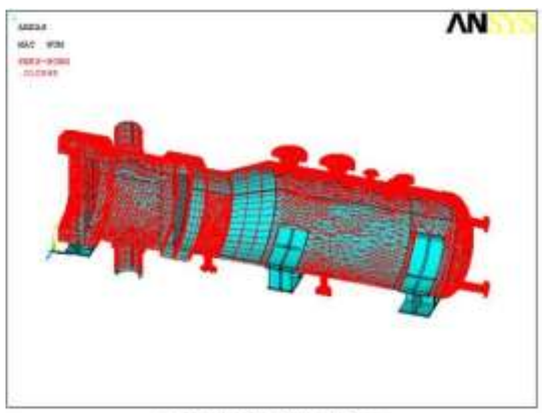
- Fixed saddle is constrained in all Dofs while rolling and sliding are kept axially free. Symmetry boundary condition on symmetry faces.
- Temperature distribution from thermal load case.
- Nozzle loads as per GA drawing

7. Load Cases:

- Vacuum Load case - Analysis with only atmospheric pressures conditions to evaluate deformation & Stresses
- Mechanical Design Load case - Analysis with only Shell & tube side pressures at Design conditions to evaluate deformation & Stresses

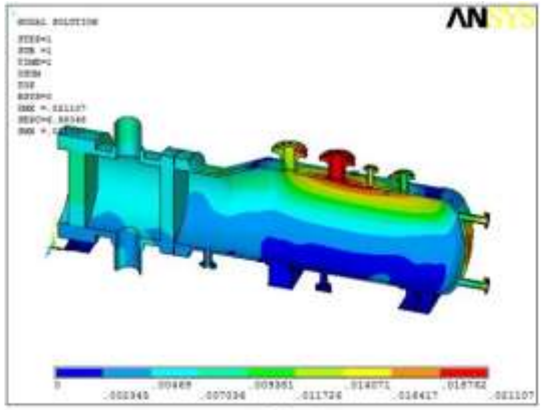
Load Case I – Vacuum Load Case

Atmospheric Pressure - 0.010545 Kg/mm^2 is applied at all External surfaces of Heat Exchanger to evaluate Stresses & Deformation in Heat Exchanger. No Internal pressure was applied for this load case..

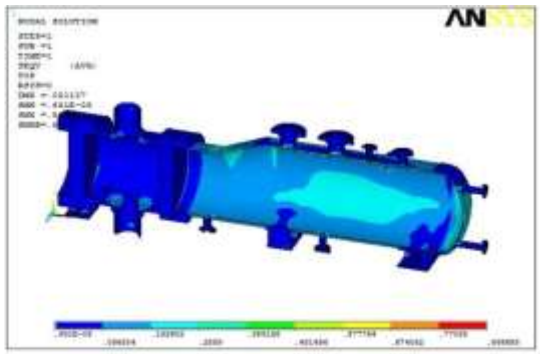


Applied atmospheric External Pressure.

RESULTS –



Deformation

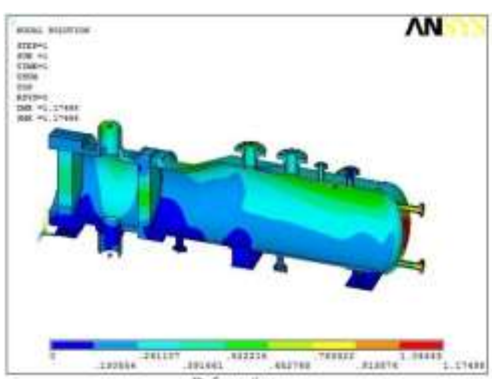


Von mises stress distribution

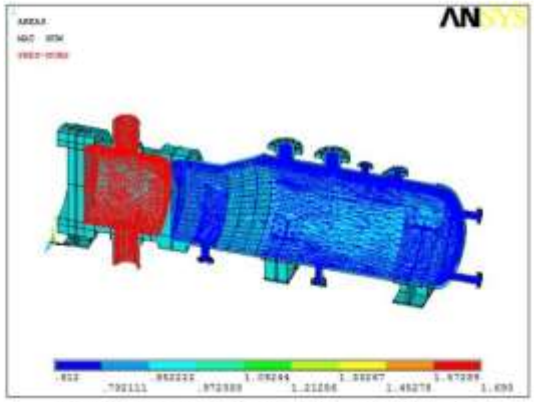
Load Case II – Mechanical Load Case

Internal design load is applied at Shell & Tube side region & Structural Analysis was carried to evaluate Stresses & Deformation in Heat Exchanger.

Internal Design Pressure, Pi (Shell Side)	0.612 Kg/mm^2
Internal Design Pressure, Pi (Channel Side)	1.693 Kg/mm^2

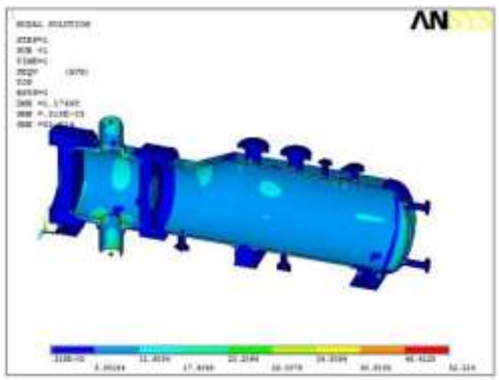


Deformation



Mechanical load - Design Pressure applied at Shell & Channel Side

RESULTS -



Von Mises Stress distribution

8. CONCLUSION:

On the basis of above study it is clear that the given design of shell and tube heat exchanger is capable of resisting all the static forces acting on it during various operations as all the values of stresses and deformation are within the limit. The results are satisfactory although we think that the thermal analysis of the same model must be done which will be the subject of a future research.

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