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Design and Analysis of Vapour Absorbing Machine

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ABSTRACT: *A vapor absorption machine (VAM) is a machine that produces chilled water using a waste heat source rather than electrical input as in the more familiar vapor compression cycle. Vapor absorption machine works on the principle of absorption refrigeration cycle. Vapor absorption machine consists of two rectangular pressure vessel shells. Both shells operate under vacuum. As these pressure vessels are subjected to vacuum, there is a chance of external pressure failure. Therefore, design of components of vapor absorption machine is carried out using ASME and TEMA code. Design pressure is 1 Bar and design temperature is 150^{o}C. Modeling of vapor absorbing machine is carried out in PRO-E software. Finite element analysis of vapor absorption machine is done on ANSYS Workbench 11.0.*

Keywords: *Pressure vessel, ASME, TEMA, PRO-E, ANSYS vacuum, external pressure*

I. INTRODUCTION

Most of industrial process uses a lot of thermal energy by burning fossil fuel to produce steam or heat for the purpose. After the processes, heat is rejected to the surrounding as waste. This waste heat can be converted to useful refrigeration by using a heat operated refrigeration system, such as an absorption refrigeration cycle. Both vapor compression and absorption refrigeration cycles accomplish the removal of heat through the evaporation of a refrigerant at a low pressure and the rejection of heat through the condensation of the refrigerant at a higher pressure. The basic difference is that an electric chiller employs a mechanical compressor to create the pressure differences necessary to circulate the refrigerant whereas the absorption chillers use waste heat source and do not use a mechanical compressor.

Main components of vapour absorbing machine are evaporator, absorber, generator and condenser. VAM consists of two rectangular pressure vessel shells. Evaporator and absorber are contained inside lower shell and generator and condenser are contained inside higher shell. Pressure in the lower shell is of the order of 6 mm of Hg(abs). While, pressure in the higher shell is of the order of 80 mm of Hg(abs). Because of inside vacuum, there is a chance of failure of pressure vessel due to external atmospheric pressure of 1 Bar. Hence, pressure vessel is designed for external pressure of 1 Bar. Design temperature for vessel is 150^{o}C. Design of pressure vessel is carried out using ASME code. Procedure for deign of external pressure is given in UG-28 of ASME Sec viii Div 1.

Construction of rectangular pressure vessel of VAM consists of two L shape plates which are welded to form rectangular shell. This shell is closed on both sides by tube sheets. Tubes are inserted in these holes. In evaporator tubes carry water to be cooled. And in absorber tubes carry water required for absorber. Stay rods are used inside vessel to maintain shape of shell against external pressure. Stiffeners are welded externally to shell plate. Cross section of stiffener is ‘C’ shape. Water boxes are used for storing water flowing through evaporator and absorber. Water boxes are connected to tube sheet with the help of bolting.

ASME Sec viii Div 1 is used for the design of shell plate, tube sheet, stiffener and bolt required for connecting water box to tube sheet. Section 5 (Mechanical standards TEMA class RCB Heat exchangers) of TEMA code is used for the design of tube sheet. Model of VAM pressure vessel is created in PRO-E. It is followed by finite element analysis in ANSYS workbench 11.0. Equivalent stress and total deformation are determined in analysis.

II. EXTERNAL PRESSURE FAILURE

The mechanism of external pressure failure is different from internal pressure failure. Internal pressure failure can be understood as a vessel failing after stresses in part or a large portion exceeds the materials strength. In contrast, during external pressure failure the vessel can no longer support its shape and suddenly, irreversibly takes on a new lower volume shape. It loses its stability. The first picture shows an internal pressure failure. The second picture shows vessels of reduced volume after external pressure failure.

Stability: A stable system is one that is stronger than required. When the vessel is pushed on, it pushes back and returns to its original shape. As external pressure is added to the system, the vessel has less reserve strength left to push back. Eventually the vessel reaches a point where it has very little reserve strength. When push on the wall of the vessel is applied, it cannot push back. At this point the vessel can change shape to a smaller volume configuration.

Figure: 1) Failure due to internal pressure

Figure: 2) Failure due to external pressure

The factors which determine the ability of a pressure vessel to withstand external pressure are
1) Thickness (t)
2) Unsupported length of vessel (L)
3) Larger dimension of rectangular cross section of vessel shell (D)
4) Material of construction

Maximum allowable pressure for vessel can be increased by
1. Increasing thickness (t)
2. Decreasing unsupported length of vessel (L)
3. Decreasing larger dimension (D)

But D and L cannot be changed because vessel size is designed for required volume for reactions taking place inside. Hence only thickness can be changed.

III. DESIGN OF PRESSURE VESSEL

3.1. Shell plate design

Design procedure for calculating thickness of shell plate suitable to withstand external pressure is given in UG-28 of ASME Sec viii Div 1 code. First assume initial thickness and calculate maximum allowable external pressure (Pa). This calculated allowable pressure must be greater than external design pressure (P). If it is lower than external design pressure then thickness of vessel should be increased. Given input is D = 1734 mm, L = 3960 mm, t = 5mm, P = 1 Bar = 0.01033 Kg/mm2. Rectangular shell with above dimensions is shown in figure 3.

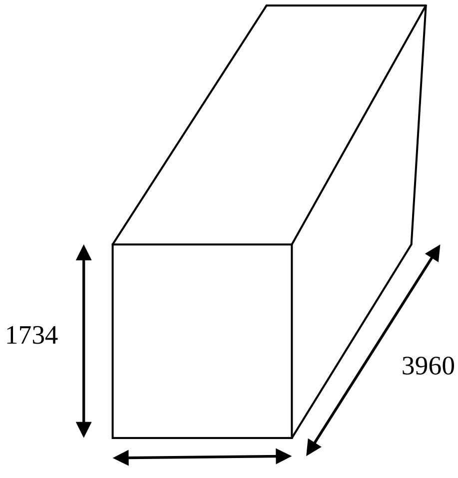

Figure 3: Rectangular pressure vessel

L/D_o=3960/1734= 2.283, D/t= 134/5 = 346.8.
For this ratios from figure G value of factor A = 0.00009.

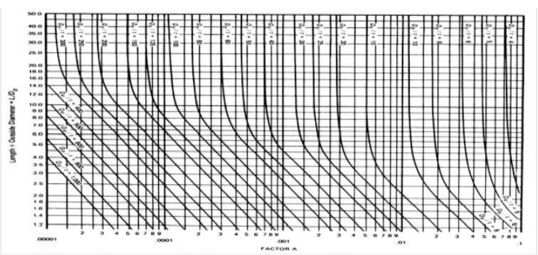

Figure 4: Graph G for determining factor A

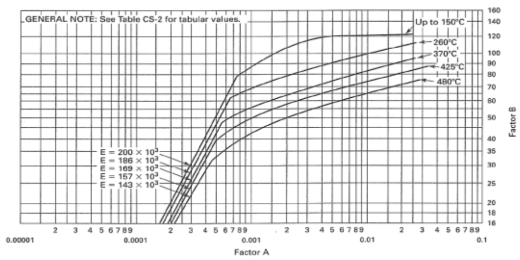

Figure 5: CS-2 chart for determining factor B

As A falls to left of curve use formula

$$P_a = \frac{2AE}{3(D_o/t)}$$

After substituting values we get max allowable pressure (pa) = 0.0037 Kg/mm2. It is lower than design pressure of 0.01033 Kg/mm2. Therefore 5 mm thickness is not safe. Hence, thickness is increased to 6mm. For 6 mm thickness again same calculations as above are performed. After calculation max allowable pressure is 0.00691 Kg/mm2. It is lower than design pressure. Hence, again thickness should be increased. But, continuous increase in thickness is not an economical option. Therefore to increase maximum allowable pressure unsupported length of vessel should be decreased. For this purpose stiffeners are used. Stiffener reduces the unsupported length of vessel. Four stiffeners are used at a distance of 792.4

mm thereby reducing unsupported length from 3960 mm to 792.4 mm. Calculations are performed with t = 6 mm and L = 792.4 mm.
L/D=792.4/1734 = 0.4569
D/t= 1734/6 = 289
D/t = 289 & L/D= 0.4569
From fig. G, Factor, A= 0.0073
From Chart CS-2, Factor, B = 6.2
Using formula

$$P_a = \frac{4B}{3(D_o/t)}$$

For this combination max allowable pressure was found to be 0.028 Kg/mm2. Now in this case it is greater than external design pressure (0.01033 Kg/mm2). Therefore design is safe.

3.2. Design of stiffener

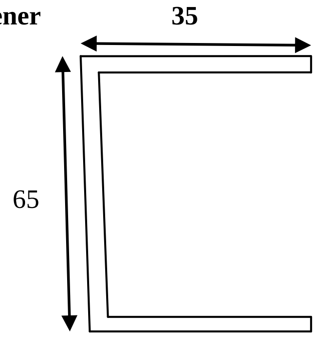

Figure 6: ‘C’ shape cross section of stiffener

Stiffeners have to satisfy certain requirements of moment of moment of inertia. Condition is that available moment Of inertia with neutral axis parallel to axis of shell should be greater than required moment of inertia. If above condition is not satisfied then stiffener with new cross section is taken. Required moment of inertia is calculated using procedure given in UG-29 of ASME Sec viii Div 1.

$$B = {}^{3}/_{4}\left(\frac{P * D}{t + {}^{A_s}/_{L}}\right)$$

From the formulaValue of factor B is 18.34
From value of B and from CS-2 chart value of factor A = 0.00017

Required moment of inertia is given by

$$I_s = \frac{\left[D^2 * L * \left(t + {}^{A_s}/_{L}\right) * A\right]}{14}$$

After substituting values we get Is = 199479.4 mm4.

Cross section of stiffener is ‘C’ shape. Available moment of inertia for ‘C’ shape cross section is 441203.5 mm4. Therefore, Available moment of inertia of stiffening ring is greater than required moment of inertia.

3.2.1 Strength of attachment weld

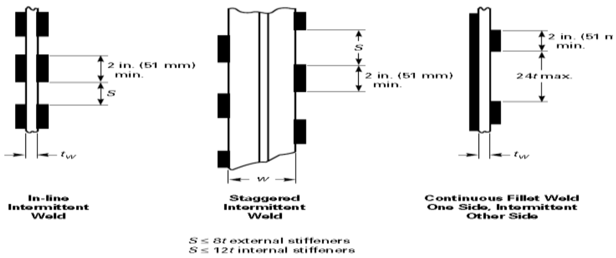

Figure 7: Arrangement of attachment weld

Stiffeners are welded to the shell with leg size of 6 mm. Procedure for checking of strength of attachment weld is given in UG-30 of ASME Sec viii Div 1. Criterion for checking weld strength is that actual load acting on the weld must be less than allowable weld for the load. The actual load on the weld is a combination of the radial pressure load between the stiffeners, the weld shear flow due to the radial load through the stiffener, and the external design load carried by the stiffener. After calculations of strength of attachment weld, it is found that actual load acting on weld is 108.04 N/mm. And allowable load for weld is 455.4 N/mm. Thus, allowable load for weld is greater than actual load on the weld. Therefore, fillet weld leg size of 6 mm is strong enough to withstand external pressure load.

3.3. Design of tube sheet

Tube sheet is a plate which is drilled with pattern of holes. Tubes are inserted in these holes. Rectangular shell is closed on both sides by tube sheets. Tube sheet is designed using both ASME and TEMA code.

a) Thickness of tube sheet after ASME calculations:

Design of non circular head with bolts is given in UG-34 of ASME sec viii Div 1. After calculations, thickness of tube sheet is 19 mm.

b) Thickness of tube sheet after TEMA calculations:

Section 5 of TEMA gives design of tube sheet. In TEMA tube sheet is designed for two conditions

1) For bending
   Tube sheet thickness for bending is 22 mm
2) For shear
   Tube sheet thickness for shear is 0.42 mm.
   Therefore according to TEMA, thickness of tube sheet is 22 mm

3.4. Design of bolts required to connect water box to tube sheet

Two water boxes are bolted to tube sheet. The bolts should be designed to contain the pressure and for the preload required to prevent leakage through the gasket. For sustaining such loads, provided area of bolts must be greater than required bolt area. Appendix 2 of ASME Sec viii Div 1 gives design procedure for bolts. After following the procedure we come to know that for evaporator side water box to maintain leak proof joint 36 bolts of M12 size are sufficient to withstand pressure of 1 Bar. And for absorber side water box 42 bolts of M12 size are sufficient to maintain leak proof joint.

IV. FINITE ELEMENT ANALYSIS OF VAPOUR ABSORBING MACHINE

Geometry of VAM is prepared in PRO-E. Then it is saved in IGES file format and imported to ANSYS workbench. Material used for vapor absorbing machine is low carbons steel SA 516 Gr 70. It is a linearly isotropic material.

Properties of material are given below:

Youngue's modulus	2.e+005 Mpa
Poisson's ratio	0.3
Density	7.856.e-006 kg/mm3
Tensile yield strength	250 Mpa
Tensile ultimate strength	460 Mpa

Element type used is Solid 186. It is a higher order 3-D 20-node solid element that exhibits quadratic displacement behavior. Meshing of geometry is carried out using hex dominant method. Meshing of geometry is shown below

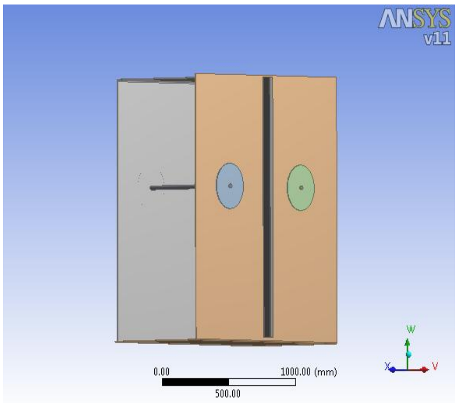

Figure 8: Geometry of VAM

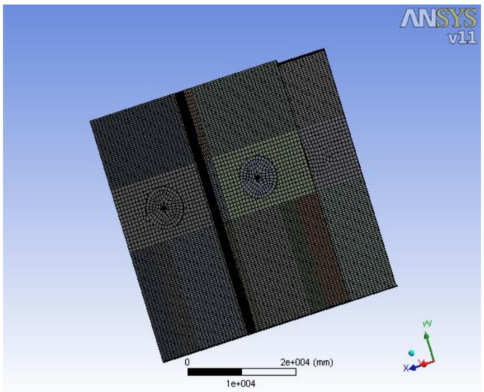

Figure 9: Meshing of VAM model

External pressure of 1 Bar is applied on geometry. Following boundary conditions are applied: Two fixed supports of 130 mm width at a distance of 150 mm from ends are given to the bottom plate of shell. Shell surface attached with tube sheet is fixed. Stay rods and stiffener are fixed to shell. After applying boundary conditions, model is solved for von-mises (equivalent) stress and total deformation occurring in the model. Results are shown below

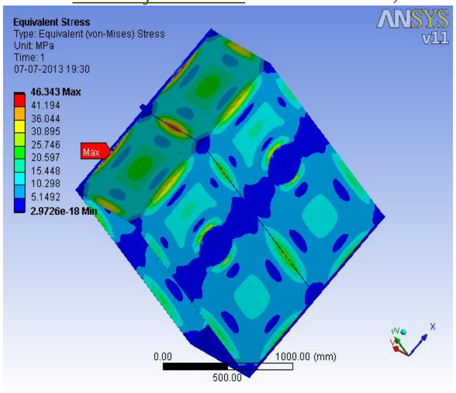

Figure 10: Equivalent stress in model

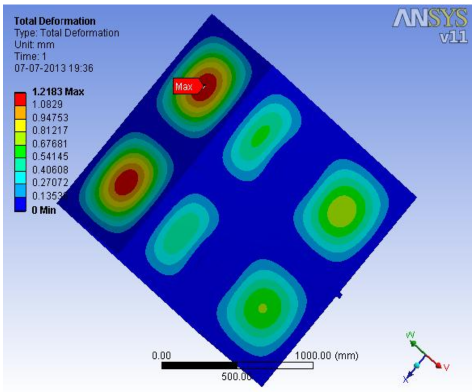

Figure 11: Total deformation

Maximum stress in the model is 46.34 Mpa and its location is at the connection of two L shape plate. And total deformation is 1.2183 mm. Max stress (46.34 Mpa) is lower than allowable stress (138 Mpa). Therefore there is no chance of failure due to external pressure.

V. CONCLUSION

Shell thickness of 6 mm with four stiffeners is safe for external design pressure of 1 Bar. Tube sheet thickness according to ASME code is 19 mm. And according to TEMA it is 22 mm. For external loading of 0.1 Mpa, 'C' shape cross section of stiffener with given dimension is suitable. Stiffener is welded to shell with one side continuous weld and other side intermittent weld. Weld leg size of 6 mm is enough to carry radial pressure load between stiffeners and weld shear flow doe to radial load between stiffeners. Forty two bolts with size of M 12 are sufficient to hold water box and absorber side tube sheet assembly for design pressure of 1 Bar. Thirty six bolts with size of M 12 are sufficient to hold water box and evaporator side tube sheet assembly for design pressure of 1 Bar. After finite element analysis on ANSYS we come to know that, maximum stress (46.34 Mpa) generating in the vessel is lower than allowable stress (138 Mpa) for vessel material.

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