Design of External Pressure Vessel Having Rectangular Cross Section Using ASME Code

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Abstract- *Pressure vessels form the integral part of industries. They are designed to perform certain applications such as boilers, heat exchangers', external pressure vessels (vacuum conditions). The safety of these vessels is of great concern and therefore need to be designed using some standard code such as ASME (American Standard for Mechanical Engineers). In this project we propose to design an external pressure vessel having rectangular cross-section of 2000 mm x 2000 mm and height 1000 mm the vessels would be designed using design by rule approach of ASME section - VIII division -1 for external pressure vessel. All the manufacturing details will also be discussed based on ASME code.*

Keywords- External Pressure Vessel, Noncircular, Pressure vessel, Rectangular vessel, Stiffeners.

I. INTRODUCTION

A pressure vessel, closed chamber sustaining pressure differences. The pressure, obtained from an external source, or by the application of heat from a direct or indirect source Pressure vessel is use in storage and pressurizations and application of it, in chemical, oil refineries, nuclear reactors, submarines etc. [11]

II. LITERATURE SURVEY

ASME code

American Society of Mechanical Engineers; 1st BPVC issued in 1914. Code Purpose, considering technological advancement to enhance public safety. Code mainly provides, requirements applicable to the Design, fabrication, Inspection, Testing etc.

Rules for construction of pressure vessel is given in the Section VIII Division 1 divided in to three subsections. According to ASME code, we should consider mandatory Appendix-13(ASME section division 1). [8]

Mandatory Appendix 13 for non-circular pressure vessel the rules in Mandatory Appendix 13 cover minimum requirements for the design, fabrication and inspection of single wall vessels having a rectangular or obround cross section. The rule of this appendix applies to the walls and parts of the vessel subject to pressure stresses including stiffening reinforcing member [6]. Especially consideration is given to the calculation of applied and allowable stresses when the structure contains butt welds joint or row of holes at location other than at side plate midlengths [6].

Reinforcing member [4]**.,** It is external strength, attached with the body to establish extra strength resist in external or internal pressure [11]. Reinforcing member shall be placed on the outside of vessels and shall be attached to the plate of the vessels by welding on each side of the reinforcing member [11]

Buckling of stiffened plates and design of stiffeners, this paper presents closed-forms solution of critical instability stresses for simply Supporter rectangular plates with stiffeners in longitudinal or transverse direction, there are only two buckling oddest at are coupled to each other for single direction stiffened plates and four for double direction stiffened plates. [9]

Rectangular Pressure Vessel of finite length [6] This paper investigates the effects of finite length on the design formulae Given by the ASME Code, and a design method based on "large deflection" Theory coefficients for short rectangular pressure vessels. Results based on Analysis are compared with values obtained from finite element computations, and with experimental data from strain gage measurements on a test pressure vessel.

Important Needs to be followed during Pressure Vessel Fabrication and Operation for Avoiding Failures [2]

In this paper, they have discussed some of the code rules to be followed during pressure vessel design, fabrication and operation with respect to international codes and standard recommendations. The violations of codes shall be avoided all the time during design and fabrication to continuously improve the quality of the vessels so that the operation and safety of the vessel would be safe all the time. Based on study, we found that, the above violations were often happening in oil and gas industries, as a result many losses and damages are still happing in these industries. this journal for their vessel design and fabrication would help to prevent failures of vessels in oil and gas industries.

By reviewing paper, it concludes that Design of vessel of non-circular cross section is given and also consists Reinforced vessel of Rectangular cross section. Design of reinforcing member of plate, bar type shape as discussed in literature review Ansys is use in simulation as Cad tool.

III. DESIGN CALCULATION

1. Material selection

Figure 1 Materoal Selection procedure

Austenitic stainless steels have the following features[10]

Easily welded,High ductility,Good corrosion resistance, Suitable for high temperatures to [e.g. 310 to 1100°C],Better resistance and it can be sustain at high temperat.Mean of Candidate material, available in low cost in market as per desire requirement. Because of Cost and need, **SS304** material is selected for pressure vessel.

2.Plate thickness Calculation [6]

All values are taken from ASME SECTION VIII DIVISION I

Nomenclature are given in Appendix

Available Design data are given below- An external pressure vessel having rectangular cross-section of 2000 mm x 2000 mm and height 1000 mm with various ports and reinforcements. Material used= SS304

T= 500 ˚C= 932 ˚F, P= 0.101325 N/mm^2

Plate thickness(mm)	$\frac{2SmA}{ScrA} + \frac{2SmB}{ScrB} \le 1.0$
8	1.25
9	1.19
10	0.2
15	0.15
20	0.087
30	0.04163

Table 1 Plate thickness value as per iteration

In addition to checking each of the four side plates and the two end plates for stability in accordance with given equation

The cross -section shall be checked for column stability by the given equation

Plate thickness(mm)	$\frac{2SmA}{ScrA} + \frac{2SmB}{ScrB} \le 1.0$
10	0.000921
20	0.000902
30	0.000859
40	0.000803
50	0.000743

Table 2 Plate cross section check value

Design is safe at 10 mm thickness of end plate and side plates (there are two end plates and 4 side plates)

2.1 Reinforcement Design-

For this see ASME code non-mandatory appendix 13

$p = \frac{t}{\sqrt{p/SC}}$ Maximum pitch distance 427 mm

Sm= $\frac{Php}{2(A1+pt)}$=0.9789 N/mm2

Sb = $\frac{Ph^2p}{12I_{11}}\frac{(1+\alpha 1^2K)}{1+K}$ = 15.7712 N/mm2

St = Sm +Sb

St =16.7424 N/mm2 < S

So, reinforcement design is safe at dimensions thickness is 10 mm and width is 100 mm

2.2 Different Shape –Non-Circular for Horizontal End Closure –Formed Plain Head Plate Thickness of Plate, t=10 mm

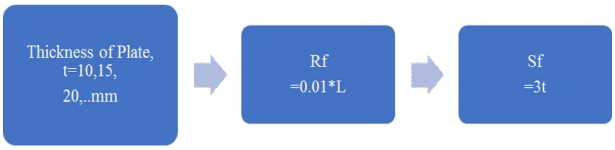


Figure 2 Design Procedure of Head use for vessel

The plain formed heads are used as end closure.
Thickness = t
Length of head plate L =2000mm
t = 10,15,20 mm
Sf =Straight flange length 3*t
Rf -fillet radius = 0.01*L =20mm
h = inside length of vessel
S= Allowable Strength of vessel material
Sf = Allowable Strength determine with design thickness
Put all value with thickness

$$t = 0.4\, h \sqrt{P/Sf}$$

Sf=1.61 *10^{-9} N/mm2
So, Sf<<< S of vessel material

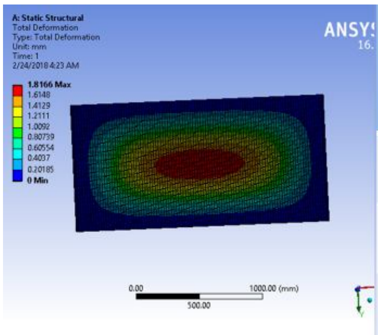
So, it is satisfied to take desire dimension for head

III. ANALYSIS

3.1 Analysis of plate

Using Ansys workbench 16.0 in static structural
Grid independent test for 10 mm plate thickness.

Boundary condition -Uniformly Distributed Load normal to plate, load value is 0.101325 N/mm2Plate is fixed from all sides



No. of div.of element	Deflection value mm	Difference mm
50	1.8092	0.0041
75	1.8133	0.0033
100	1.8166	

Figure 3 Deformation of 10 mm thickness plate

3.2 Analysis of Rectangular pressure vessel with reinforcement

Boundary condition all are same as plate given above
Geometry model of rectangular pressure vessel having reinforcement

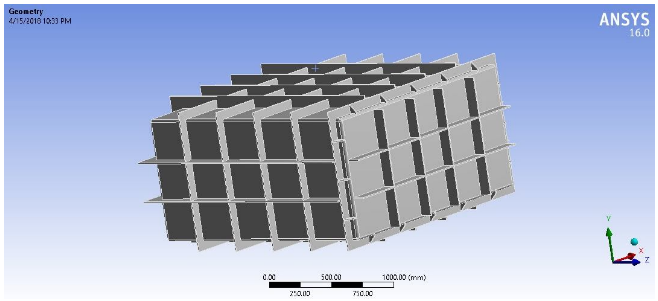


Figure 4 Geometrical model made in Solid works 2018

Deflection Analysis of Rectangular Pressure vessel

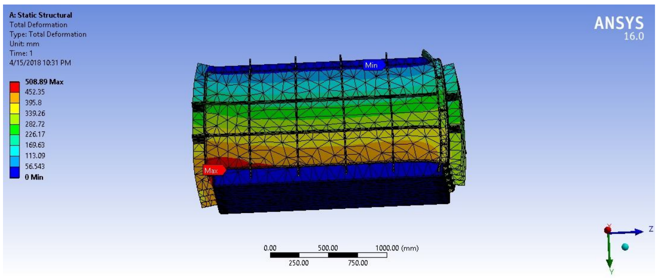


Figure 5 Static structurer analysis using ANSYS-Deformation

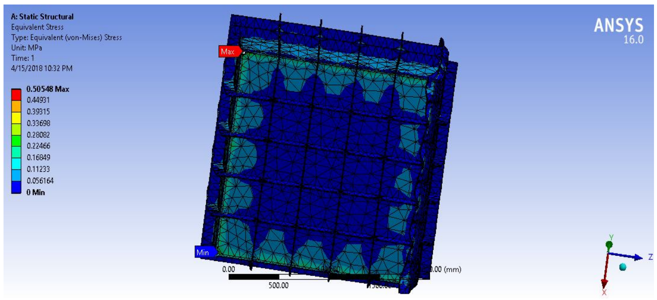


Figure 6 Von messes Stress value of vessel

The models are analyzed for the total deformation and equivalent stress (von-misses stress). The analysis is done for vessel

IV. CONCLUSIONS

An external pressure vessel design, which can survive at the temperature of 500˚C without failure The design will be validated by comparing the values of stresses and Deflection with 3D model analysis Specially non-circular vessel beneficial in the maximum use of area with compare to circular vessel.According to non-mandatory appendix 13 we can easily design noncircular vessel without fail.

APPENDIX

P e = External Pressure N/mm^2,S_{mA}= Compression stress applied to short edge of side panels due to external pressure on the end plates, N/mm^2,S_{mB}= Compression stress applied to Long edge of side panels due to external pressure on the end plates, N/mm^2,S_{crA}= S_{crB}= plate buckling stress when panel is subjected to stresses on two opposite edge in direction indicate by subscripts A and B , N/mm^2,A =Subscripts to identify stress at long side of plate = Subscripts to identify stresses at short side of plate=Poisson ratio of material,E_2 = modulus of elasticity at design temperature, N/mm^2,H=Inside length of short side of vessel, mm ,H= Inside length of long side of vessel, mm ,Ho=Outside length of short side of vessel, mm, ho= Outside length of long side of vessel, mm, T= thickness of plate, mm ,M= bending moment N mm, S =Allowable stress (UG-23) N/mm^2, $K_A =$ Ratio of long to short side length of plate, Plate buckling co efficient ,K_B = Ratio of short to long side length of plate, Plate buckling co efficient ,S_y = Yield Strength of material N/mm^2 ,L_v =Length of vessel, mmS_{crA} = S'_{crA} when $S'_{crA} \leq S_y/2$, $S'_{crA} = \frac{\Omega^2 E_2}{12(1-v^2)}\left(\frac{t}{H}\right)^2 K_A$

= S''_{crA} when $S'_{crA} > S_y/2$, N/mm^2 S''_{crA} = $S_y - \frac{S_y^2}{4S'_{crA}}$

S_{crB} = S'_{crB} when $S'_{crB} \leq S_y/2$, $S'_{crB} = \frac{\Omega^2 E_2}{12(1-v^2)}\left(\frac{t}{h}\right)^2 K_B$

= S''_{crB} when $S'_{crB} > S_y/2$, N/mm^2 , S''_{crB} = $S_y - \frac{S_y^2}{4S'_{crB}}$

R_1=Least radius of gyration of non-circular cross section of vessel, mm

c_1= Distance from neutral axis of cross section of plate, mm,S_b= Bending stress, N/mm^2,I_e= Moment of inertia about axis parallel to long side of vessel and passing through centroid of cross sectional area of rectangular vessel, $\overline{y}$ = Distance from the geometric center of end plate to centroid of cross sectional area of rectangular vessel, mm ,Sa = Allowable strength value N/mm^2 ,Fa = Allowable force value N/mm^2 ,Fe'= Element force value N/mm^2

For Reinforcement, P=External Pressure N/mm^2, p=maximum pitch distance between adjacent reinforcing member mm ,P e=internal pressure N/mm^2, t = thickness of reinforcing member mm ,S=allowable stress at material design temperature N/mm^2 ,C =Plate co efficient (UG-47),J= Stress parameter at joint ,w=width of reinforcing member mm ,h=inside length of long side of vessel mm ,H=in side length of short side of vessel mm, h 1 =Centroidal length of reinforcing member on long side of vessel mm,H1=Centroidal length of reinforcing member of short side of vessel mm,A1=A2=Cross section area of reinforcing member at thickness t mm^2,I11=moment of inertia of combined reinforcing member,I21 = moment of inertia of combined reinforcing member α1=Reinforcing parameter (H/h),K=reinforcing parameter (I21/I11) α 1,E = Co efficient of Joint efficiency ,Sm = Membrane stress N/mm^2,Sb = Bending Stress N/mm^2,St = Total Stress N/mm^2,P=0.101325 N/mm^2 t = 10 mm S=86.10 N/mm^2 C =2.5 (UG-47) J= 2.4 w= 100 mm ,h=1980 mm H= 980 mm h 1 =990 mm H1=420 mm A1=A2=1000 mm^2,I11=833333.33 I21 = 8333.33 ,α 1 =0.54 K=9.99/1000000000 E =0.65

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