DESIGN AND ANALYSIS OF COMPOSITE PRESSURE VESSEL AS PER ASME SECTION X

Mr. Raj C Patel[1], Mr. Hitesh R.Raiyani [2]
[1]*Department of Mechanical Engineering (CAD/CAM), LJIET, Ahmedabad,*
[2]*Department of Mechanical Engineering (CAD/CAM), LJIET, Ahmedabad.*

Abstract — *A pressure vessel is designed for performing hot water heater. This chamber has to deal with the temperature of up to 85˚C and pressure up to 145 psi. The pressure inside makes the pressure vessel an internal pressure vessel. The reduction of weight of the pressure vessel without significant of any properties of pressure vessel also the safety of these vessel is prime concern as their failure loads to huge loss of properties. Thus, the ASME codes section X are widely used to design these pressure vessels that has in build factor of safety in the design procedure. Here, the whole pressure vessel is designed and optimization including shell, nozzle, end closures, using ASME Code section X. The optimum end closure design is selected by comparing them for cost and deflection. The designs are analyzed using ANSYS Multiphysics (Workbench 15.0) and validated by using Theory of Plates..*

Keywords – *internal pressure vessel, ASME code, cylindrical pressure vessel, end closure*

I INTRODUCTION

Pressure vessel can be simply defined as “a container with a pressure differential between inside and outside.” According to ASME (Section X): “Pressure Vessels are containers for the containment of pressure either external or internal.” The pressure can be applied from an External Source, or by the application of heat from a Direct or Indirect Source, or any combination thereof.

Figure-1. Cylindrical Pressure Vessel [7]

The term "vertical turret lathe" is applied to machines wherein the same essential design of the horizontal version is upended, which allows the headstock to sit on the floor and the faceplate to become a horizontal rotating table.

II LITERATURE REVIEW

[1] N.Karthik : Authors concluded that design of pressure vessel using ASME is found out to be thicker and heavier than the design using finite element method.

[2] J.H. Wang : Investigated the buckling behavior of cylinder shell with different types of joint as rigid, semi rigid and flexible. Authors found out that the more joint is flexible, the more readily buckle occurs.

[3] Yongmei Zhu : Authors focuses on spherical shells with openings and reinforcement under uniform external pressure to minimize the buckling instability. The critical load of a spherical shell with an opening was approximately 4.4%–8% lower than that of a complete spherical shell.

[4] Khaled M. Shokry : The mechanical properties such as tensile strength, compressive strength and hardness at different temperatures are studied. Author concluded that, no change in dimensions stability or loss of weight for GRP at all temperature and period of exposure. And the ultimate tensile strength, compressive strength and hardness of all tested GRP composite decrease over a temperature of 100°C

[5] M. S. EL-Wazerya : In this research work, an E-glass fiber with random oriented reinforced polymer composite was developed by hand lay-up technique with varying fiber percentages (15%, 30%, 45%, and 60% by weight percentage). 60% fiber by weight percentage gives the max tensile strength & bending strength.

III DESIGN OF PRESSURE VESSEL AS PER ASME SECTION X

Material Selection

Laminate materials

Fiber-reinforced plastic materials shall hereinafter be designated as laminates.
(a) Laminates, as herein considered, are composite structures consisting of one or more of the following reinforcements embedded in a resin matrix:

(1) glass
(2) carbon or graphite
(3) aramid

(b) The Fabricator shall keep on file the published specifications for all laminate materials used in each vessel fabrication, the material manufacturer's recommendations for storage conditions and shelf life for all laminate materials, and the material manufacturer's certification that each shipment conforms to said specification requirements. This certification shall be part of the Procedure Specification.
Fiber system
Glass fibers
The glass fibers used in any of the fabrication processes permitted by this Section shall be one or more of the following glass compositions:
(a) Type A
(b) Type E.

Table 1 Properties of E- glass fiber

Poisson Ratio	Elastic Limit	Tensile Strength	Density
0.23	2875 Mpa	105.83 Mpa	2.6 mg/m^3

1) Cylinder Thickness (t) Calculation

a) Cylindrical Shells Under Uniform Internal Pressure.

The minimum thickness of cylindrical shells under internal pressure shall be the greater of (a) or (b) below, but not less than 1/4 in. (6 mm).

a) Longitudinal Stress

t1=PR/(2(o.oo1E1))

b) Circumferential Stress

t2=PR/(o.oo1E2)

where
E1 = tensile modulus in longitudinal direction
E2 = tensile modulus in circumferential direction
P = internal pressure
R = inside radius
t1 = structural wall thickness for longitudinal stress
t2 = structural wall thickness for circumferential stress

Specification
Capacity : 40 gallon or 0.151 m3
Pressure : 145 psi
Diameter : 610 mm
Circumferential stress :

t1=PR/(o.oo1E1)

t1=(999.74× 〖10〗^3× 0.610)/(0.001 × 85 × 〖10〗^9)

t1=7.74 mm ≅8 mm

Now find length

$0.151=\pi/4\ d^{2}\times l$

$l=(4\times 0.151)/(\pi\times (0.610)^{2})$

$l=516\cong 520$ mm

2) Head thickness calculation

Thickness of Heads Under Internal Pressure. The required thickness of vessel heads under internal pressure shall be computed by the appropriate equation [(a) or (b) below].

a) Ellipsoidal Head

$t=PD/(2(o.oo1E_{hd}))$

b) Hemispherical Head

$t=(PR_s)/(2(o.oo1E_{hd}))$

where

D = inside diameter

Ehd = design modulus for the head

P = internal pressure

Rs = inside spherical radius

t = head wall thickness

Ellipsoidal Head

$t=PD/(2(o.oo1E_{hd}))$

$t=(999.74\times 10^{3}\times 0.610)/(2(0.001\times 85\times 10^{9}))$

$t_1=3.74$ mm $\cong 4$ mm

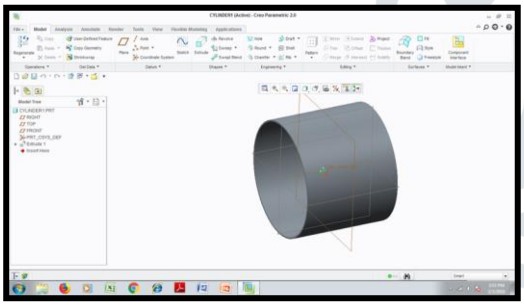

Figure-2 Model for cylinder shell

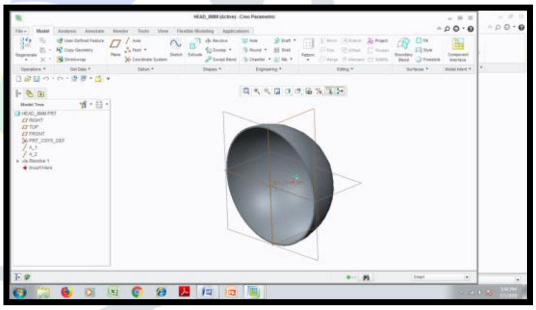

Figure-3 Model for Head

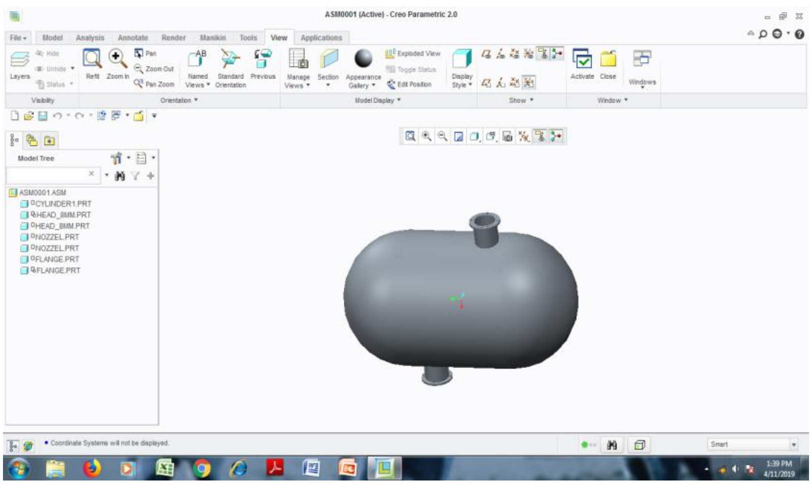

Figure-4 model

IV ANALIYSIS OF PRESSURE VESSEL AS PER ASME SECTION X

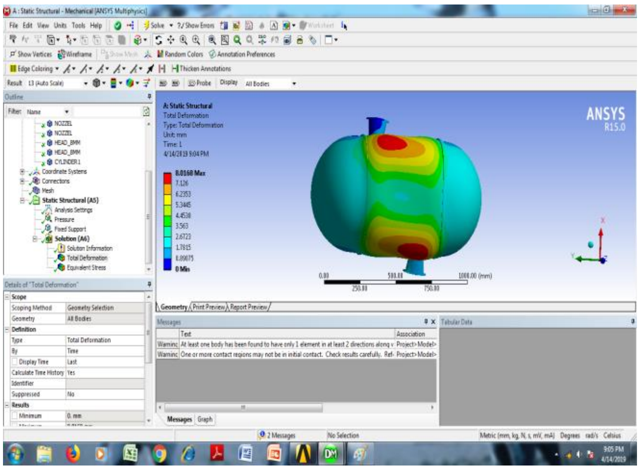

Figure-5 Total deformation

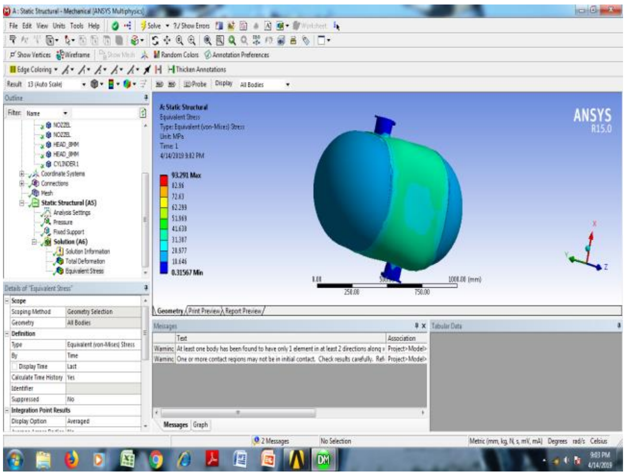

Figure-6 total equivalent stress

Here total deformation 8.0168 mm max and total equivalent stress 93.291 Mpa max.
So our design is safe as per ASME section X.

V CONCLUSION

It is concluded that the design of pressure vessel as per ASME section X a compact design, light weight and gives more strength as compare to ASME section VIII. Pressure vessel design as per ASME section X gives high strength and rigidity also its satisfied the todays demand of space requirement and weight optimization.

REFERENCES

[1] N.Karthik, Design Optimization and Buckling Analysis of Pressure Vessel, International Journal of Computer Engineering In Research Trends, Volume 3, Issue 5, ISSN (O): 2349-7084, May-2016, pp 246-250

[2] J.H. Wang, A. Koizumi, “Buckling of cylindrical shells with longitudinal joints under external pressure”, Thin Walled Structures, 48, pp 897-904, 12 June 2010

[3] Yongmei Zhu, “Opening reinforcement design and buckling of spherical shell subjected to external pressure” 25 October 2017

[4] Khaled M. Shokry, “Effect of Temperature on Mechanical Properties of Glass Reinforced Plastic GRP” World Applied Sciences Journal 31 (7), ISSN 1818-4952, july 2014, pp 1341-1344.

[5] M. S. EL-Wazerya, Mechanical Properties of Glass Fiber Reinforced Polyester Composites , International Journal of Applied Science and Engineering , march 2017, pp 121-131

[6] ASME Boiler and Pressure Vessel Code, Section-X, Rules for Construction of Pressure Vessels, ASME, New York (2015)

[7] Pressure Vessel Head design, accessed on 12th September, 2018 http://www.boardmaninc.com/news.html/2016/08/15/volume-vi.iii-asme-pressure-vessels/