# Design of E-Glass Fibre Rain forced Plastic Pressure Vessel Design as per ASME Sec X

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## Abstract

One Glass Fiber Reinforced Plastic pressure vessel, subjected to internal design pressure of 90 psi was designed in accordance with the procedures set out in ASME Section X, mandatory design rules for class II vessels with Method A Design Rules. Destructive testing was done to find out the modulus of elasticity and flexural modulus as per ASTM D 3039. All the design factors were considered and designed the safe pressure vessel.

**Keywords: Design of Composite Pressure Vessel, ASME Sec X, Material Testing, Reinforcement Pad Dimensions**

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## I. INTRODUCTION

Glass fibers are primarily composed of silicon dioxide with some modifying agents [2]. E-glass (electrical glass) accounts for the largest production of glass fibers in industry due to its low cost despite its mechanical properties that are lower than other grades of glass fibers. It is specially used in making chemical processing tanks and pressure vessels. These tanks and vessels are usually large in size. Pipe branch connections, manholes etc. in the vessels are very common and are potential weak points. These opening contain discontinuities in the structure. The large size combining with the discontinuities limit the operating pressure of these vessels. Each tank and vessel is custom made and they are different. Economic consideration does not allow manufacturers to destructively test one vessel to prove that another identical one meets the design requirements. These vessels therefore fall under the standard code for design and fabrication of FRP pressure vessels, ASME has published Boiler and Pressure Vessel Code Section X: Fiber-Reinforced Plastic pressure vessels Work has been reported for investigation of design of FRP vessels based on philosophy of ASME Boiler and Pressure Vessel Code. Hisao Fukunaga et al. [3] discussed the optimum design of helically wound composite pressure vessels. J. Blachut [4] analysed filament wound torispheres under external pressure. Paul et al. [5] discussed damage criterion approach to design fiber reinforced vessel. Baoping Cai et al. [6] presented reliability-based load and resistance factor design of composite pressure vessel under external hydrostatic pressure.

Work related to investigation of design of FRP pressure vessels based on philosophy of ASME Boiler and Pressure Vessel Code has been reported but very little work has been reported about the step by step procedure for the design of various components of FRP pressure vessel and selection of dimensions of certain components as per the standards given in ASME section X. The present work deals with the design of small FRP pressure vessel subjected to internal design pressure of 90 psig with methodology given in ASME Section X including material testing. After design, this vessel will be fabricated and burst test will be carried out by applying hydrostatic pressure for research work.

## II. PROBLEM DEFINITION

The vessel is designed with the methodology of ASME Section X, Class II vessel. In this approach also Method A was used to design the vessel. It will be constructed of mat laminate with contact molding method. Section X requires the use of lamina properties coupled with lamination analysis to determine the laminate properties for use in method A design. These properties were used together with the lamination theory equations in Section X. Fig.1 shows the overall dimensions of shell and nozzle with nozzle location. The specification of nozzle has been given in Table I.

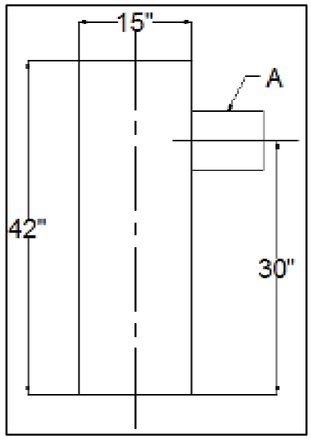

Fig. 1: Overall dimensions of shell and nozzle with nozzle location.

Table - 1
Specification of Nozzle

| *Mark* | *Nominal Size (in)* | *Elevation (in)* | *Orientation (Deg)* | *Location* |
|---|---|---|---|---|
| *A* | *3* | *30* | *90* | *Shell* |

## III. Material testing

As per the methodology of ASME, section X, to design FRP pressure vessel, mechanical properties of FRP material like Modulus of Elasticity, Flexural Modulus and Poisson‘s Ratio are required. To find out these properties Article RD-12 is used for mat laminate The thickness was taken as per ASTM D 3039: Standard Test Method for Tensile properties of Polymer Matrix Composites Materials. And these obtained values by Rangoli FRP, Vapi are used by us. The selection of resin and fiber for this panel was done as per ASME Section X. It was manufactured from Isophthalic Polyester resin and E‘glass fiber, Chopped Strand Mat (CSM) by a commercial fabricator using a hand lay-up technique.

Table – 2
Mechanical Properties of Test Specimen (Lamina Properties)

| | |
|---|---|
| *Mean value of Modulus of Elasticity for specimens.* | *$6.5*10^6$ psi* |
| *Mean Value of Flexural Modulus* | *$3.2*10^6$ psi* |
| *Poisson‘s Ratio* | *0.28* |

Table – 3
Material Properties

| | |
|---|---|
| *Tensile modulus in axial direction(E1)* | *$6.5*10^6$ psi* |
| *Tensile modulus in hoop direction(E2)* | *$1.8*10^6$ psi* |
| *Flexural modulus in axial direction(E1f)* | *$3.2*10^6$ psi* |
| *Flexural Direction in hoop direction(E2f)* | *$3.2*10^6$ psi* |
| *Poisson‘s ratio for stress in axial direction and contraction in hoop direction(v1)* | *0.28* |
| *Poisson‘s ratio for stress in hoop direction and contraction in axial direction(v2)* | *0.28* |

## IV. Design of FRP pressure vessel

ASME Section X has two classes of vessels: I and II, both of which differ in scope. The classes are distinguished as follows:

- Class I vessel design is qualified through possibly destructive fatigue and pressure testing of a prototype.
- Class II vessel design is qualified through mandatory design rules and non-destructive acceptance testing, which includes an Acoustic Emission examination

In present work, class II vessel design approach is used. The pressure scope for Class II vessels is more complicated, depending on the size of the vessel. There are two methods for design calculations of Class II Vessels: Method A that uses design rules, and Method B that provides for design by stress analysis. Irrespective to class or methods, Section X vessels must be between 6 in. and 192 in. in internal diameter. Vessels designed by Method A are limited to 100 psi internal design pressure and 144 in. internal diameter. Vessels designed by Method B rules shall have pressure and diameter restrictions as follows:

Vessels may be designed using a combination of Methods A and B. For such vessels the maximum design pressure is limited to 100 psig with a maximum inside diameter of 144 in. Vessels designed by either Methods A or B are limited to a maximum external pressure of 15 psig.

In present work, Method A was used and the internal diameter of vessel was considered as 15 in. (381 mm) and internal design pressure of vessel was considered as 90 psi.

## V. DESIGN CALCULATION

$D_o$= outside diameter
$E_1$ = 6.5*$10^6$ psi
= tensile modulus in the longitudinal (axial) direction, psi
$E_2$ = 1.8*$10^6$ psi
= tensile modulus in the circumferential (hoop) direction, psi
F = design factor: 5 for external pressure on cylinders,10 for external pressure on spheres and heads, and internal pressure on reinforcements
L = 42 in.
= length of cylinder
P = 90 psi
= internal pressure
$P_a$ = allowable external pressure
R = 7.5 in.
= inside radius, shell or head
r = 1.5 in.
= inside radius, nozzle (in.)
$S_S$ = 1,000 psi
= secondary bond shear strength
t = thickness, in.
$v_1$ = 0.28
= Poisson's ratio, longitudinal (axial) direction
$v_2$ = 0.28
= Poisson's ratio, circumferential (hoop) direction

### A. Cylindrical Shells under Uniform Internal Pressure

Design of Shell: Article RD-1171.1 of ASME section X gives the following rule for the minimum thickness of a cylindrical shell subjected to internal pressure. The minimum thickness of cylindrical shells under internal pressure shall be greater of (a) and (b) below, but not less than ¼ in. (6 mm).

Table – 4
Nozzle and Nozzle Flange Dimensions [1]

| *Mark* | *Nominal Size(in.)* | *Flange OD(in.)* | *Bolt Circle (in.)* | *Bolt Hole Dia. (in.)* | *Bolt Size(in.) and Number* | *Flange Thickness (in.)* | *Nozzle Thickness (in.)* |
|---|---|---|---|---|---|---|---|
| *A* | *3* | *7 1/2* | *6* | *3/4* | *5/8 and 4* | *1/2* | *1/4* |

Table – 5
Shell Flange Dimensions [1]

| *Nominal Size(in.)* | *Flange OD(in.)* | *Bolt Circle (in.)* | *Bolt Hole Dia. (in.)* | *Bolt Size(in.) and Number* | *Flange Thickness(in.)* | *Shell Thickness (in.)* |
|---|---|---|---|---|---|---|
| *15* | $23\frac{1}{2}$ | $21\frac{1}{4}$ | $1\frac{1}{8}$ | *1 and 16* | *1* | *0.38 (As per design calculations)* |

#### 1) Longitudinal Stress

$$t_1 = \frac{PR}{2(0.001E_1 - 0.6P)} \quad (1)$$

$$= \frac{90 * 7.5}{2(6500 - 0.6 * 90)}$$

= 0.052 inch

#### 2) Circumferential Stress

$$t2 = \frac{PR}{(0.001E_2 - 0.6P)} \quad (2)$$

$$= \frac{90*7.5}{(1800-0.6*90)}$$

= 0.38 inch

Where $t_1$= structural wall thickness for longitudinal stress, $t_2$= structural wall thickness for circumferential stress, P = internal pressure = 90 psi, R= inside radius of shell = 7.5 in. Therefore, let $t$ = $t_2$ =0.38 in.

### B. Spherical Shells under Internal Pressure

Assume hand lay-up sphere with $E1 = E2 = 1.8*10^6$ psi:

$$t = \frac{PR}{2(0.001E - 0.6P)} \quad (3)$$

$$= \frac{90*7.5}{2(1800 - 0.6*90)}$$

= 0.18 in.

### C. Cylindrical Shells under External Pressure

Critical Length: Assume $t$ = 0.38in. Therefore $Do$ =2($R$ + $t$) = 15.76 in.

Do = 2(R + t) = 2 (7.5 + 0.38) = 15.76 in.

$$L_C = 1.14\sqrt[4]{(1-v_1v_2)}.D_o.\sqrt{\frac{D_o}{t}} \quad (4)$$

$$L_C = 1.14\sqrt[4]{1-(0.28*0.28)} * 15.76\sqrt{\frac{15.76}{0.38}}$$

= 143.19 in. > L

$E_r = \sqrt{E_1E_2}$

$$= \sqrt{1.8*6.5} * 10^6$$

= 3.42*106 psi

$$K = 3.6 - \frac{2E_r}{E_1+E_2}$$

= 2.77

For external pressure, let $F$ = 5:

$$P_A = \frac{K\frac{E_r}{F}\frac{D_o}{L}(\frac{t}{D_o})^{5/2}}{1-0.45\sqrt{\frac{t}{D_o}}}$$

$$P_A = \frac{2.77*\frac{3.42*10^6}{5}*\frac{15.76}{42}(\frac{0.38}{15.76})^{5/2}}{1-0.45\sqrt{\frac{0.38}{15.76}}}$$

= 69.015 psi

### D. Spherical Shells under Uniform External Pressure

Assume hand lay-up sphere with $E1$= $E2$ = 1.8*$10^6$ psi. Therefore, let $E$ = 1.8*$10^6$ psi and $F$ = 10. Also,
Assume: $t$ = 0.38 in. Therefore, $Ro$ = $R$ + $t$ = 7.88 in.
Allowable pressure:

$$P_A = \frac{0.41t^2\times\frac{E}{F}}{R_o{}^2\times\sqrt{3(1-v_1v_2)}} \quad (5)$$

$$= \frac{0.41*0.38^2*\frac{1.8*10^6}{10}}{7.88^2*\sqrt{3(1-0.28*0.28)}}$$

= 62.073 psi

### E. Thickness of Heads under Internal Pressure

Assume hand lay-up head with $E1$ = $E2$ = 1.8*$10^6$ psi. Therefore, let $E$ = 1.8 * $10^6$ and let
$D$ = 2$R$ = inside diameter = 15 in.

#### 1) Ellipsoidal Head

$$t = \frac{PD}{2(0.001E-0.6P)} \quad (6)$$

$$= \frac{15*90}{2((0001*1.8*10^6)-(0.6*90))}$$

= 0.38 in.

#### 2) Hemispherical Head

$$t = \frac{PR}{2(0.001E-0.6P)} \quad (7)$$

$$= \frac{15*90}{2((0.001*1.8*10^6)-(0.6*90))}$$

= 0.19 in.

### F. Thickness of Heads under External Pressure

Assume hand lay-up head with E1 = E2 = 1.8 * $10^6$psi. Therefore, let E = 1.8 * $10^6$psi, and
F = 10. Also,
Assume: t = 0.38 in. so: Do = 2R + t = 15.76 in. and
Ko = 0.9 for 2:1 ellipsoidal heads

$$P_A = \frac{0.41t^2 \times \frac{E}{F}}{(K_o D_o)^2 \times \sqrt{3(1-v_1 v_2)}} \quad (8)$$

$$= \frac{0.41 * 0.38^2 \times \frac{1.8*10^6}{10}}{(0.9 * 15.76)^2 \times \sqrt{3(1-0.28*0.28)}} = 31.85 \text{ psi}$$

### G. *Reinforcement of Openings and Nozzle Attachments*

Nozzle attachments and reinforcing pad design: Table RD-620.1 of ASME Section X gives the dimensions of nozzles and their flanges constructed of hand layup and pressure-moulded FRP. Nozzles and flanges of these dimensions satisfy the design requirements of Section X. Flanges or nozzle designs not listed in this table can be designed by using Article RD-1176. Table V lists the nozzle and nozzle flange dimensions and Table VI lists the shell flange dimensions for the test vessel, which were taken from Table RD-620.1 of ASME section X and are for 90 psig internal pressure. Article RD-1174 of ASME section X gives the methodology to design the reinforcement for openings in shell and head. Typical dimensions of reinforcement pad and nozzle overlay are given in Fig. 4. For the reinforcing pad and the nozzle overlay to be fully defined, criteria (I) through (IV) below must be met.

#### 1) *Length of Secondary Overlay on Nozzle*

The secondary bond length, $L_b$ on the nozzle shall be sufficient to withstand the internal pressure acting against the cross-sectional area of the nozzle.

For internal pressure, let *F* = 10; also, let *r* = 3 in. = inside radius of nozzle.

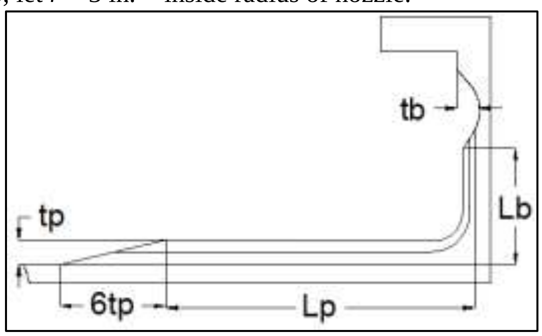


Fig. 2: Dimensions of reinforcing pad and nozzle overlay [1]

$$L_b = \frac{Pr}{2\frac{S_s}{F}} \quad (9)$$

$$= \frac{90*1.5}{2\frac{1000}{10}}$$

$$= 0.675 \text{ in.} < 3 \text{ in.}$$

$$= 3 \text{ in.}$$

Where P = Internal pressure which is 90 psi (Table x), r = Internal radius of nozzle which is 3 in, $S_s$ = secondary bond strength in shear which is 1000 psi maximum as per ASME section X and F= Design Factor which is 5 for external pressure on cylinders and 10 for external pressure on spheres and heads and internal pressure on reinforcements as per ASME section X. So for test vessel F= 10.

By putting above values one gets $L_b$ = 0.675 in. which is less than 3 in. So $L_b$ = 3 in.(76.2 mm).
or 3 in, whichever is greater.
So $L_b$ = 3 in.

Table – 6
Component wise pressure

| *Component* | *H (in.)* | *P (psig)* |
|---|---|---|
| *Shell* | *42* | *90* |
| *Nozzle* | *30* | *89* |

#### 2) *Thickness of Secondary Overlay on Nozzle*

The secondary overlay thickness $t_b$ on the nozzle shall be sufficient to withstand the nozzle internal pressure.

$$t_b = \frac{Pr}{S_{ut} - 0.6P}; \text{ Sut} = 0.001*E = 1.8 * 103 \quad (10)$$

$$= \frac{90 * 1.5}{(1.8 * 10^3) - (0.6 * 90)}$$

$$= 0.077 \text{ in.}$$

Or 0.25 in., whichever is greater
So $t_b$ = 0.25 in.

#### 3) *Thickness of Reinforcement Pad on Shell or Head*

The thickness tp of the reinforcing pad shall be the greater of (1) and (2) below:

 

- A thickness of secondary overlay with strength equivalent to the tensile strength in the circumferential direction of the shell thickness removed:

$$t_{p1} = \frac{PR}{0.001E_2} \tag{11}$$

$$= \frac{90*7.5}{0.001*1.8*10^6}$$

$$= 0.375 \text{ in.}$$

- A thickness of secondary overlay, which when added to the shell thickness, will reduce the bending stress at the opening to an allowable level. The allowable bending stress shall be defined as 0.1% of the flexural modulus of the reinforcing laminate in its circumferential direction. It will be calculated as follows.

a) Step 1: Compute Beta factor

$$\beta = \frac{\sqrt[4]{3(1-v_1v_2)}}{2}\frac{r}{\sqrt{Rt}} \tag{12}$$

$$= \frac{\sqrt[4]{3(1-0.28*0.28)}}{2} * \frac{1.5}{\sqrt{7.5*0.38}}$$

$$= 0.57$$

Putting R = 6 in., r = 1.5 in., t = thickness of shell =0.5614 in and ν1, ν2 equal to 0.32 and 0.32 respectively one gets β= 0.5234

b) Step 2

Figure RD-1174.3 of ASME section X, gives the value of stress concentration factors (Kt) for a circular hole in a pressurized cylindrical shell. For test vessel the value of Kt for membrane plus bending from this graph is $K_t = 5.2$

$$S_{max} = S_2 * K_t = 1.8 * 10^3 * 5.2 = 9360 \text{ psi}$$

Where $S_2$ = allowable stress for the shell laminate in the circumferential direction, defined as 0.001 $E_2$ = 0.001 x $1.8*10^6$ psi = $1.8*10^3$ psi

So $S_{max}$ = 9360 psi

c) Step 4

Determine the moment M associated with the stress $S_{max}$ being applied at the edge of the opening. Assume a moment beam 1 in. wide and having a thickness equal to the shell thickness, with the stress decreasing away from the edge of the opening.

$$M = \frac{S_{max}t^2}{6} \tag{13}$$

$$= \frac{9360 * 0.38^2}{6}$$

$$= 592.8 \text{ in-lb}$$

Where t = vessel shell thickness = 0.38 in

So M = 592.8 in. lb

- Step 5: Determine the thickness of reinforcement that will reduce the stress imposed by the moment M to the allowable Sf, defined as 0.001Ef, Assume an effective equivalent moment to be M/2.

$$t_{p2} = \sqrt{\frac{6\frac{M}{2}}{S_f}} - t \tag{14}$$

$$= \sqrt{\frac{6 * \frac{592.8}{2}}{20 * 10^3}} - 0.38$$

$$= -0.081 \text{ in.}$$

Where $S_f$ = allowable stress, defined as $0.001E_f = 20 * 10^3$. Here $E_f$ is nothing but the flexural modulus of reinforcing laminate in the circumferential direction which is equal to $20 * 10^6$ psi (Table III). t = thickness of vessel shell = 0.38 in.

By putting above values, one can find out thickness of reinforcing pad ($t_p$) as $t_p$= - 0.081 in.

The thickness of the reinforcing pad shall be the greater of the thickness computed for (1) and (2) above. So $t_p$= 0.375 in.(9.5 mm)

Length of Reinforcing Pad on shell ($L_p$): The reinforcing pad shall project a distance $L_p$ from the edge of the opening. The distance $L_p$ shall be at least as great as the greater of (1) and (2) below. The pad shall terminate in a taper over an additional distance six times the thickness of the pad.

- A secondary bond area on the shell shall provide sufficient shear area to resist the internal pressure force on the nozzle. By convention this area will be expressed in terms of a distance $L_p$ that the reinforcing pad will extend out from the nozzle and will be computed as follows:

$$L_p = \frac{\pi L_c P}{\frac{4S_s}{F}} \tag{15}$$

$$= \frac{\pi*3*90}{\frac{4*1000}{10}}$$

$$= 2.12 \text{ in.} < 3 \text{ in}$$

$$L_p = 3 \text{ in.}$$

$$L_c = 2r = 2*1.5$$

 

= 3in.

Where F= design factor = 10, $L_c$= Longest chord length of opening = 3 in., P = internal pressure = 90 psi, $S_s$= Secondary bond shear strength= 1000 psi

– Minimum Reinforcing Pad Requirements

1) For nozzles 6 in. diameter and less, $L_p$ shall be at least as great as $L_c$.
2) For nozzles greater than 6 in. (152 mm) diameter, but less than or equal to 12 in. (305 mm) diameter, $L_p$= 6 in. or ½ $L_p$, whichever is greater.
3) For nozzles greater than 12 in. diameter, $L_p$ shall be at least as great as ½ $L_c$. See Fig. 4

So for test vessel, the diameter of nozzle is 3 in. which is less than 6 in. so $L_p$ shall be at least as great as $L_c$. that is 3 in.

So finally $L_p$ = 3 in.(76.2mm)

$$t_o = \frac{P(R+t)}{0.001E_2} \qquad (16)$$

= 0.394 in.

$$L_o = \frac{PR}{(\frac{2S_S}{F})}$$

## VI. Conclusions

In the present work, an attempt has been made to design glass-fiber reinforced plastic pressure vessel subjected to internal pressure of 90 psig. Design is based on general accordance with the procedures set out in ASME Section X, mandatory design rules for class II vessels with Method A - Design Rules. All the design factors were considered and designed the safe pressure vessel. Following dimensions of FRP pressure vessel and reinforcement pad are drawn from present design

1) Shell Thickness (t) = 0.38 in.
2) Length of secondary overlay on nozzle ($L_b$) = 3 in.
3) Thickness of secondary overlay on nozzle ($t_b$) = 0.25 in.
4) Thickness of reinforcement pad on shell ($t_p$) = .375 in.
5) Length of Reinforcing Pad on shell ($L_p$) = 3 in.

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