DESIGN OF PRESSURE VESSEL SADDLE AND ZICK ANALYSIS

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Abstract - *Supports are the important part of a pressure Vessel to holding for different purposes during manufacturing and in process plant. Different supports are required to hold pressure vessel like skirt support, lug support, saddle support. For horizontal pressure vessel saddle supports are permanently welded with the Vessel. In saddle there should be a proper thickness of base plate, web plate thickness, rib plate thickness and proper number of rob plate should be used to design efficient saddle and it may fail due to own of pressure vessel. It will also decrease the cost of saddle. By adding saddle pressure vessel will generate stresses at different parts of the pressure vessel. These stresses must be considered during designing of saddle. Otherwise, the pressure vessel which was designed by proper ASME codes, it may be failed due to stresses generated at pressure vessel due to saddle. This paper contains proper methodology to design the saddle and also consider the generated stresses. It also contains analysis for stresses given by Zick scientist.*

***Key Words*: Pressure vessel, Saddle, Stresses, Zick Analysis, Costing**

1.INTRODUCTION

The horizontal pressure vessel is required to be supported otherwise the vessel may be damaged and Vertical pressure vessels are required to be supported at saddle for post weld heat treatment, during the transportation of pressure vessel, during the processing in the plant and also during hydrotest. The pressure vessels, in horizontal condition, are usually supported at the vertical cradles. These cradles are called saddle. These saddles are used for the transportation of the pressure vessels are called shipping saddles. The main aim of this project is to setup a generalized methodology to design the Saddles and stresses generated due to saddle at the pressure vessel. The project work also carried about Zick Analysis.

2.DESIGN OF PRESSURE VESSEL SADDLE [2]

For designing the saddle support it is required to a calculate the thickness: Top flange, thickness (tf), Base thickness (tw), Stiffener thickness (ts), Web thickness (tb)

Table -1: Sample Table format

Vessel Type	Cylindrical
Vessel Position	Horizontal
Design Pressure Required (P)	257.9 psi
Design Temperature	149 F
Radiography	0.85
Vessel Inside Diameter	61.2598 Inch
Vessel Outside Diameter	62.9921 Inch
Vessel Wall Thickness (t)	0.86614 Inch
Corrosion Allowance (C)	0.23622 Inch
Vessel weight (Empty)	16337.3 lb
Vessel Weight (Liquid)	228383 lb
Saddle to saddle distance (A)	22.0472 Inch
Vessel Head thickness (th)	0.7874 Inch

Table -2 : Material Data

Material Shell & Heads Saddle	ASME SA516 Grade 7 ASME SA283 Grade C
Minimum Tensile Strength Shell & Heads Saddle	 6000 psig 55000 psig
Minimum Yield Strength Shell & Heads Saddle	 32000 psig 30000 psig
Allowable Tensile Strength Shell & Heads Saddle	 20000 psig 15700 psig

Vessel Layout

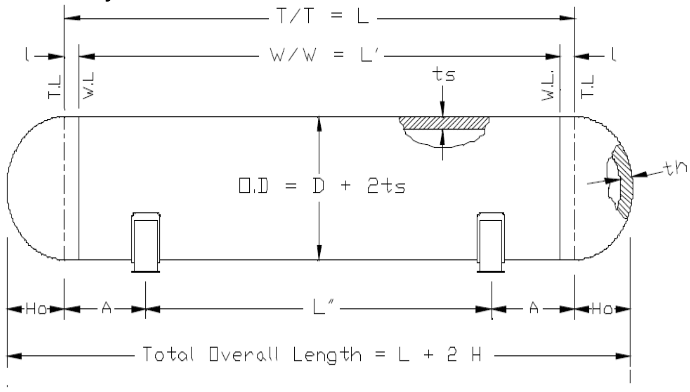

Fig -1: Pressure Vessel Layout

Distance from tangent line to Saddle support can be found from the trial and error method.

Head wall thickness(min.) (t_h) = 0.7874 Inch

Saddle support to tangent line distance (A) = 22.0472 Inch

Tangent line to head depth distance,

$H_o = \frac{D}{4} + t_h = \frac{61.2598}{4} + 0.7874 = 16.102$ Inch

Shell length (Tangent to tangent) (L) = 188.964 Inch

Shell length (Welding to welding), (L′) = 186.614 Inch

Distance between saddle to saddle

$L" = L - 2A = 189.764 - (2 * 22.0472)$

$= 145.669$ Inch

External depth of Head,

$H = H_o\ l = 16.1024 + 1.5748 = 17.6772$ Inch

Total vessel overall length,

$L + 2H_o = 189.764 + (2 * 16.1024) = 221.969$ Inch

Vessel weight (empty) (W) = 16377.3 lb

liquid weight (water) (W_c) = 22383.1 lb

Vessel total weight (W_t) = 38760.4 lb

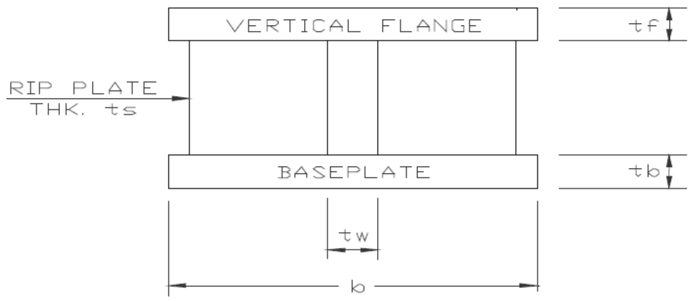

Fig -2: I Section view of Saddle

2.1 Determination of top flange thickness (t_f)

$$\text{Top Flange thickness } (t_f) = \sqrt{\left(6 * \frac{M_b}{S_b}\right)}$$

S_b = Allowable design bending stress = 0.66 = 19800 psig

$$M_b = \frac{P_{ll}}{b} * \frac{b}{2} * \frac{b}{4} = \frac{P_{ll} * b}{8}$$

$$M_b = \frac{P_{ll} * b}{8} = \frac{500.925 * 11.811}{8} = 739.555 \frac{lb}{inch}$$

$$P_{ll} = \text{Linear Load per unit length } \frac{lb}{inch}$$

$$P_{ll} = \frac{Q}{R_o}\left[\frac{1 + \cos B}{L - B + \cos B * \sin B}\right]$$

$$P_{ll} = \frac{19380.2}{31.4961} * \left[\frac{1 + \cos 120}{186.146 - 120 + \cos 120 * \sin 120}\right]$$

$$P_{ll} = 500.925 \frac{lb}{inch}$$

$$\text{Flange thickness } (t_f) = \sqrt{6 * \frac{M_b}{S_b}} = \sqrt{6 * \frac{739.555}{19800}}$$

$$= 0.4734 \text{ Inch}$$

2.2 Determination of web plate thickness (t_w)

Assumed minimum thickness of plate 0.5 Inch

Area (a) = $1 * 0.5$ $Inch^2$

Minimum radius of gyration(k) = $0.289 * t_w$

$$\frac{P}{a} = \frac{18000}{1 + \left[\frac{1}{18000} * \left(\frac{h}{k}\right)^2\right]}$$

$$h = t_w * \sqrt{\left(\frac{1500}{P_{ll}}\right)\left(18000 t_w - P_{ll}\right)}$$

$$h = 0.5 * \sqrt{\frac{1500}{500.925} * \{(18000 * 0.5) - 500.925\}}$$

$$= 2127 \text{ mm} > \text{Distance (C)} = 961 \text{ mm}$$

So we can use web thickness $t_w = 0.5518$ Inch

Rib plate width,

$$R_w = \frac{1}{2}[b - (2\,cl + t_w)]$$

$$= \frac{1}{2}[11.811 - (2 * 0.59055 + 0.5518)]$$

$$R_w = 5.03937 \text{ Inch}$$

Width of Top Flange (B) =

$$(2 * \text{ Width of Stiffeners}) + (2 * 55) + t_w = 14.9606 \text{ Inch}$$

2.3 Calculation of base plate thickness (tb)

$$T_b = \sqrt{\left[\frac{(Q * b)}{26400} * m\right]}$$

$$T_b = \sqrt{\left[\frac{(19380.2 * 11.811)}{26400} * 62.9921\right]}$$

$$T_b = 0.98425 \text{ Inch}$$

2.4 Determination thickness of stiffeners (t_s)

An approximate number for stiffeners is:

$$N = \frac{m_m}{24} + 1$$

where m_m = Base plate length in inch

$$m_m = 0.8 * \text{vessel O.D.} = 0.88 * 62.9921$$

$$m_m = 50.3937 \text{ Inch}$$

From Equation (4.4),

$$N = \frac{m_m}{24} + 1 = \frac{50.3937}{24} + 1 = 3.09974$$

Therefore we can use 4 number of stiffeners.

The stiffener thickness (t_s) is 0.375 Inch minimum for pressure ves

We can use 4 stiffeners with thickness $t_s = 0.551$ Inch

The distance between two stiffeners is,

$$S_s = \frac{[m - 2(0.5t_s + cl)]}{(n - 1)}$$

$$= \frac{[62.9921 - 2(0.50.86614 + 0.59055)]}{(4 - 1)} = 20.4199 \text{ Inch}$$

Width of outside stiffeners $(S_w) = 2\,R_w + t_w$

$$= (2 * 5.03937) + 0.55118 = 10.6299 \text{ Inch}$$

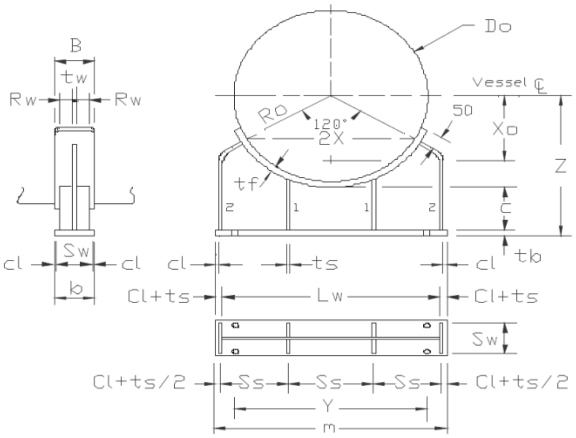

Fig -3: Designed saddle 2D Drawing

3 ANALYSIS OF PRESSURE VESSEL SADDLE [1]

Stresses at Pressure Vessel by attaching two number of saddles,

3.1 Longitudinal Stresses [3]

Longitudinal force = $5.32 * 10^3$ lb

Maximum moment $= Fl * Z = 3.77 * 10^5$ lb. inch

$$\text{Bending moment} = fo * Q\,(Z - xo)$$
$$= 86860.31 \text{ lb . inch}$$

Bending moment accured by the weight of the vessel is

$$\left(Q * \frac{b}{2}\right) fo * Q\,(Z - xo) = Q * \frac{b}{2}$$

$$\text{Minimum value for } b = 2 * \frac{BM}{Q} = 8.96382 \text{ Inch}$$
$$= 227.6811 \text{ mm} < 300 \text{ mm}$$

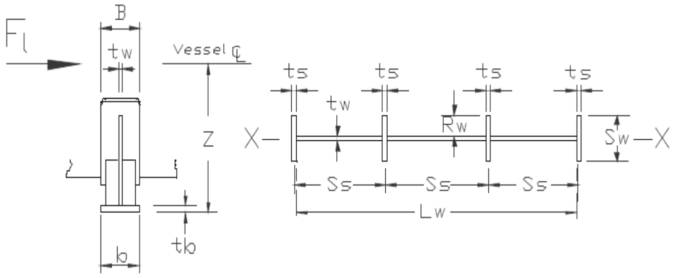

Fig -4: FBD for stresses in longitudinal direction

Cross − sectional Area, A

$$= (L_w * t_w) + (2\,S_w * t_s) + [2(n-2)R_w * t_s]$$

$$A = (37.04186 * 0.55118) + (2 * 469.147 * 0.86614) + [2(4-2)5.0393 * 0.86614]$$

$A = 56.2899 \text{ Inch}^2$

Moment of inertia, I_{xx}

$$= L_w * \frac{{t_w}^3}{12} + (n-2)t_s + \frac{2t_s({S_w}^3 - {t_w}^3)}{12}$$

$$I_{xx} = 37.04186 * \frac{0.5511^3}{12} + (4-2)0.866\frac{469.147^3}{12} + \frac{2 * 0.866(469.147^3 - 0.5511^3)}{12}$$

$I_{xx} = 221.512 \text{ Inch}^4$

$$\text{Section modulus,} = L_w * \frac{{t_w}^2}{6} + 2t_s * \frac{{S_w}^2}{6} + \frac{2t_s\,({S_w}^3 - {t_w}^3)}{6\,S_w}$$

$$W_{xx} = 37.042 * \frac{0.55^3}{6} + \left(2 * 0.866 * \frac{469.147^3}{6}\right) + \frac{2 * 0.866\,(469.147^3 - 0.5511^3)}{6 * 469.147}$$

$W_{xx} = 44.5915 \text{ Inch}^3$

$$\text{Dimension } X_1 = \frac{I_{xx}}{W_{xx}} = 4.96758 \text{ Inch}$$

$$\text{Bending stress, } S_b = \frac{M_{max.}}{W_{xx}} = 8469.26 \text{ psig}$$

$$\text{Bending stress which is allowable } (S') = 1.2 * S$$
$$= 18830 \text{ psig}$$

$$\text{Shear stress } (T) = \frac{Fl}{A} = 94.6738 \text{ psig}$$

Stress which is allowable $(S') = 0.5 * S = 7840$ psig

3.2 Transverse Direction Stresses [3]

Transverse force, $F_t = 2664.59$ lb

Max. moment, $M_{max.} = F_t * Z = 188829$ lb. inch

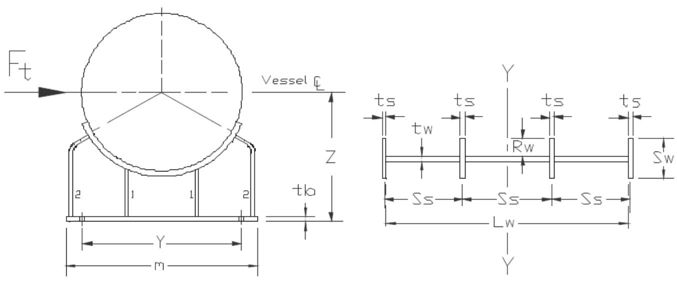

Fig -5: FBD for stresses in transverse direction

Cross − sectional Area, $A = 56.2899 \text{ Inch}^2$

Moment of inertia, I_{yy}

$$= t_w * \frac{{L_w}^3}{12} + [(L_w + 2t_s)^3 - {L_w}^3]\frac{S_w}{12} + \frac{2t_s({S_w}^3 - {t_w}^3)}{12} + 2R_w\frac{(s_s(n-3) + t_s)^3}{12}$$

$$= 0.55\frac{37.04^3}{12} + \frac{[(37.04 + 2 * 0.86)^3 - 469.2^3]469.2}{12} + \frac{2 * 5.04(20.42(4-3) - 0.86)^3)}{12} + 2 * 5.039\frac{(20.42(4-3) + 0.866)^3}{12}$$

$I_{yy} = 22429.5\ Inch^4$

Section modulus, W_{yy}

$$= t_w * \frac{L_w{}^2}{6} + S_w[\frac{(L_w + 2t_s)^3 - L_w{}^3}{6(L_w + 2t_s)} + \frac{2R_w(S_s + t_s)^3}{6}$$

$W_{yy} = 804.78\ Inch^3$

Dimension Y $= \frac{I_{yy}}{W_{yy}} = 23.8703$ Inch

Bending stress, $S_b = \frac{M_{max.}}{W_{yy}} = 234.634$ psig)

Bending stress which is allowable $(S') = 1.2 * S$

$= 18830$ psig

Shear stress (T) $= \frac{Ft}{A} = 47.33$ psig

Stress which is allowable $(S') = 0.5 * S = 7840$ psig

3.3 Calculation of Bending Moment and Bending stress at saddles [1]

Bending moment in saddle (compression and tension) (M_1)

$$M_1 = A * Q * \left(1 - \frac{1 + \frac{R^2 - H^2}{2 * L * A} - \frac{A}{L}}{\frac{4H}{3L} + 1}\right)$$

$$= 19380.2 * 22.047 * \left(1 - \frac{1 - \frac{22.0472}{189.764} + \frac{30.6299^2 - 17.6772^2}{2 * 22.0472 * 189.764}}{1 + \frac{4 * 17.677}{3 * 189.764}}\right)$$

$$M_1 = 823586\ lb.inch$$

Longitudinal bending stress, for tension (S_1)

$$= \frac{M_1}{t_s * R^2 * K_1} = \frac{823586}{0.335 * 15.6299^2 * 0.86614}$$

$$S_1 = 2861.77\ psig$$

Longitudinal bending stress, for compression (S_1)

$$= -\frac{M_1}{K_8 * t_s * R^2} = -\frac{823586}{0.603 * 15.6299^2 * 0.86614}$$

$S_1 = -589.87$ psig

3.4 Bending stress and bending moment at mid-span of Pressure Vessel [1]

Bending moment at midspan (M_1')

$$= Q * L\left(\frac{2\frac{R^2 - H^2}{L^2} + 1}{\frac{4H}{3L} + 1} - \frac{4 * A}{L}\right)$$

$$M_1' = 19380.2 * 189.764\left(\frac{1 + 2\frac{30.6299^2 - 17.6772^2}{189.764^2}}{1 + \frac{4 * 17.6772}{3 * 189.764}} - \frac{4 * 22.0472}{189.764}\right)$$

$M_1{}' = 3333925$ lb. inch

Longitudinal bending stress at midspan, S_1'

$$= -\frac{M_1{}'}{(3.14 * R^2 * t_s)} = -\frac{3333925}{3.14 * 15.6299^2 * 0.866}$$

$S_1{}' = -1235.31$ psig

3.5 Shell Stresses cause because of Internal Pressure [1]

$$S = \frac{P * R}{2 * E_s * T_s} = \frac{255.682 * 30.6299}{2 * 0.85 * 0.866} = 5470.01\ psig$$

3.6 Tangential shear stress [1]

Tangential shear stress at shell (S2)

Where, $A > \frac{R}{2}$ and ring isn't used

$$S_2 = \frac{Q * K_2}{t_s * R}\left(\frac{L - 2A}{\frac{4H}{3} + L}\right)$$

$$= \frac{1.717 * 19380.2}{30.6299 * 0.86614}\left(\frac{189.764 - (2 * 22.0472)}{189.764 + \frac{4 * 17.6772}{3}}\right)$$

$S_2 = 1106.25$ psig

Stress which is allowable in Shell $= 0.8 * S$

$= 16000$ psig

Ratio of the stress acting $= \frac{S_2}{0.8 * S} = 0.06914$

Ratio of the stress acting is lesser than 1. So the Tangential Stress is safe for design.

Tangential shear stress at shell (S2),

Where $A > \frac{R}{2}$ and ring is used

$$S_2 = \frac{K_3 * Q}{R * t_s}\left(\frac{L - 2A}{L + \frac{4H}{3}}\right)$$

$$= \frac{0.319 * 19380.2}{30.6299 * 0.86614}\left(\frac{189.764 - (2 * 22.0472)}{189.764 + \frac{4 * 17.6772}{3}}\right)$$

$S_2 = 301.362$ psig

Shell Stress which is allowable $= 0.8S = 16000$ psig

Ratio of the stress acting $= \frac{S_2}{0.8S} = 0.01884 < 1$

Ratio of the stress acting is lesser than 1. So the Tangential Stress is safe for design.

Tangential shear stress at head (Sh), Where of $A > R/2$ and ring isn't used

$$S_h = \frac{K_2 * Q}{R * t_h}\left(\frac{L - 2A}{L + \frac{4H}{3}}\right)$$

$$S_h = \frac{1.171 * 19380.2}{30.6299 * 0.7874}\left(\frac{189.764 - (2 * 22.0472}{189.764 + \frac{4 * 17.6772}{3}}\right)$$

$S_h = 1216.68$ psig

Stress which is allowable in Head $= 0.8 * = 16000$ psig

Ratio of the stress acting $= \frac{S_2}{0.8S} = 0.07604 < 1$

Ratio of the stress acting is lesser than 1.

So, the Tangential Stress is safe for design.

Tangential shear stress at shell,

Where $A \leq \frac{R}{2}$

$$S_2 = \frac{Q * K_4}{t_s * R} = \frac{0.88 * 19380.2}{30.6299 * 0.86614} = 1205.51 \text{ psig}$$

Allowable tangential stress in Shell $= 0.8 * = 16000$ psig

Ratio of the stress acting $= \frac{S_2}{0.8 * S} = 0.07534 < 1$

Ratio of the stress acting is lesser than 1.

So, the Tangential Stress is safe for design.

Tangential shear stress at head, Where $A \leq \frac{R}{2}$

$$S_2 = \frac{Q * K_4}{R * t_h} = \frac{0.88 * 19380.2}{30.6299 * 0.7874} = 1326.84 \text{ psig}$$

Allowable tangential stress in shell $= 0.8 * = 16000$ psig

Ratio of the stress acting $= \frac{S_2}{0.8 * S} = 0.08287$

Ratio of the stress acting is lesser than 1. So, the Tangential Stress is safe for design.

Additional tangential shear stress at head

In case of $A \leq \frac{R}{2}$

$$S_3 = \frac{Q * K_5}{t_h * R} = \frac{0.401 * 19380.2}{30.6299 * 0.7874} = 604.165 \text{ psig}$$

Stress due to Internal Pressure $= \frac{P * R}{2 * E_s * t_s}$

$$= \frac{257.898 * 30.6299}{2 * 0.85 * 0.86614} = 5470.01 \text{ psig}$$

Allowable tensile stress in Head = 1.25 = 25000 psig

Ratio of the stress acting $= \frac{S_3}{1.25 * S} = 0.2188$

Ratio of the stress acting is lesser than 1. So the Tangential Stress is safe for design.

3.7 Circumferential stress [1]

Circumferential stress at saddle horn

Where $L \geq 8R$ and unstiffened

$$S_4 = -\frac{Q}{4 * t_s * (b + 1.56\, R\, t_s)} - \frac{3 * K_6}{2 * t_s^2}$$

$$= -\frac{19380.2}{4 * 0.86614 * (11.811 + 1.56 * 30.6299 * 0.86614)} - \frac{3 * 0.0238}{2 * 0.86614^2}$$

$S_4 = -1806.46$ psig

Allowable Circumferential stress in Shell,

$= 1.5 * S = 30000$ psig

Ratio of the stress acting $= \frac{S_4}{1.5\, S} = 0.06022$

Ratio of the stress acting is lesser than 1. So the Circumferential Stress is safe for design.

Circumferential stress calculation at horn of saddle,

Where L < 8 R and unstiffened

$$S_5 = \left(-\frac{Q}{4 * t_s * (b + 1.56\, R\, t_s)} - \frac{12 * K_6 * Q * R}{L * t_s^2}\right)$$

$$= \left(-\frac{19380.2}{4 * 0.866(11.81 + (1.56 * 30.629 * 0.866))} - \frac{12 * 0.0238 * 19380.2 * 30.629}{189.764 * 0.866^2}\right)$$

$S_5 = -2828.5$ psig

Allowable circumferential stress in shell $= 1.5 * S$
$= 30000$ psig

Ratio of the stress acting $= \left(\frac{S_5}{1.5\, S}\right) = 0.09428 < 1$

Ratio of the stress acting is lesser than 1. So the Circumferential Stress is safe for design.

Circumferential stress at bottom part of shell,

$$S_6 = \left(-\frac{Q * K_7}{(b + 1.56\, R\, t_s) * t_s}\right)$$

$$= \left(-\frac{19380.2 * 0.76}{0.86614 * (11.811 + (1.56 * 30.6299 * 0.86614))}\right)$$

$S_6 = 1419.03$ psig

Ratio of the stress acting $= \left(\frac{S_5}{0.5\, S}\right) = 0.08869 < 1$

Ratio of the stress acting is lesser than 1. So the Circumferential Stress is safe for design

3.8 Check of tension at Web of the saddle [1]

Horizontal force at web,

$F_w = K_8 * Q = 0.603 * 19380.2 = 22531$ lb

Effective web area,

$A_w = h_s * t_w = 62.992 * 0.551 = 34.7086$ Inch

Stress in web, $S_w = \frac{F_w}{A_w} = \frac{22531}{34.7086} = 649.147$ psig

Stress which is allowable at web, as below

$S_{aw} = 0.6 * Y = 18000$ psig

Ratio $= \frac{S_w}{0.6\, Y} = 0.03606 < 1$

Vertical force at web, $F_v = Q = 37364.8$ lb

Stress in web, $S_v = \frac{F_v}{A_w} = \frac{37364.8}{34.7086} = 1076.53$ lb

Stress which is allowable in web (compression),

$S_{aw} = 0.33 * Y = 9900$ psig

Ratio of the stress acting $= \frac{S_v}{0.33 * Y} = 0.10874 < 1$

3.9 Check the lowest section of Saddle [1]

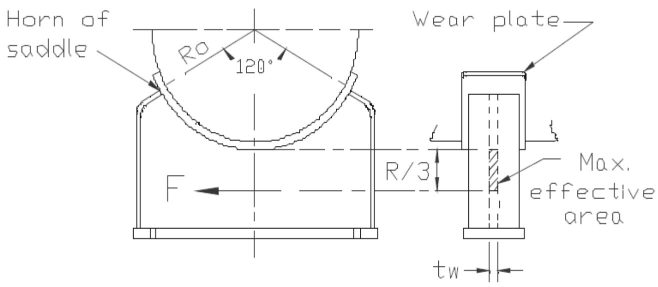

Fig -6: Saddle at lowest section

The lowest section of the saddle is as shown in figure 6

$F = K_{11} * Q = 0.204 * 19380.2 = 7622.42$ lb

For resisting this force, the area of required in web plate required is,

$A = \left(\frac{R}{3}\right) . t_w = \left(\frac{30.6299}{3}\right) . 0.55118 = 5.78668 \text{ inch}^2$

The calculated stress,

$$S_{cal.} = \frac{F}{A} = \frac{7622.42}{5.78668} = 1317.24 \text{ psig}$$

The Stress which is allowable,

$$S_{all.} = \frac{2}{3} . (S_w) = \frac{2}{3} . (649.147) = 10466.7 \text{ psig}$$

Ratio of the stress acting $= \frac{S_{cal.}}{S_{all.}} = 0.12585 < 1$

So, for horizontal force (F) the thickness of the web plate is satisfied.

3.10 Bearing Pressure [1]

$$\text{Bearing Pressure} = \frac{Q}{(b * m)} = \frac{19380.2}{(11.811 * 62.9921)}$$
$$= 26.0486 \text{ psig}$$

Which is lesser than 750 (Allowable)

4. CONCLUSIONS

As per calculation of different for saddle parts and stress calculated from Zick method and the comparison between stresses for different locations are lesser than the Stress which is allowable. So, the design of Pressure Vessel Saddle is safe for manufacturing. Stresses generated at pressure vessel due to attached saddle are lesser than the Stress which is allowable.

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