Design and Fatigue Calculations of Pressure Vessel as Per ASME

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Abstract— Pressure Vessel design using ASME BPVC Section VIII, the Division 1 includes an alternative set of rules for material, design, fabrication, inspection and testing of pressure vessels having internal pressure. The Division 2 also has provisions to utilize finite element analysis, to determine stresses in the equipment and traditional theoretical calculation approach to estimate fatigue life of the vessel.

Key words: Pressure Vessel, ASME

I. INTRODUCTION

A. Pressure Vessel

A pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. The pressure differential is dangerous and many fatal accidents have occurred in the history of their development and operation. Consequently, their design, manufacture, and operation are regulated by engineering authorities backed by legislation. The commonly used shapes of pressure vessel are spherical, cylindrical, cylindrical shells with hemispherical ends. Pressure vessels shapes made of sections of spheres, cylinders, and cones are usually employed.

Cylindrical Pressure Vessel is widely used in the industries. A common design is a cylinder with end caps called heads. Head shapes are frequently either hemispherical or dished (Torispherical). More complicated shapes have historically been much harder to analyze for safe operation and are usually far more difficult to construct. The wide application of this vessel has made the studies and engineering design to be more important than before. The best design need to be obtained in order to ensure the safety, performance and reliability of the vessel. The large pressure difference necessitates careful designing of the vessels to avoid fatal accidents, which is why their design, manufacturing and operation are often regulated by engineering authorities. The ASME Boiler and Pressure Vessel Code (BPVC) is one such set of mandatory guidelines to design pressure vessels in accordance with the standards, to ensure prolonged useful life as well as safety. Within the ASME BPVC Section VIII, the Division 2 includes an alternative set of rules for material, design, fabrication, inspection and testing of pressure vessels having internal pressure. Exceeding 15psi. The Division 2 also has provisions to utilize finite element analysis, to determine stresses in the equipment and traditional theoretical calculation approach to estimate fatigue life of the vessel (mentioned in Part 5 “Design by Analysis Requirements”).

B. Objective

1) Protection against Plastic Collapse.
2) Protection against Local Failure.
3) Protection against Collapse from Buckling
4) Protection against Failure from Cyclic Loading

C. Scope of Fatigue Consideration in Design

Fatigue is a phenomenon associated with variable loading or more precisely to cyclic stressing or straining of a material. Just as we human beings get fatigue when a specific task is repeatedly performed, in a similar manner metallic components subjected to variable loading get fatigue, which leads to their premature failure under specific condition. Fatigue failure results mainly due to variable loading or more prissily due to cyclic variations in the applied loading or induced stress so starting from the basic concepts of (variable) non static loading, we will be discussing in detail how it leads to fatigue failure in components, what factor influence them, how to account them & finally how to design parts components to resist failure by fatigue.

II. DESIGN DATA

Design Code	ASME Sec. VIII, Division 1 & 2 Edition 2010.
Fluid in Service	Air
Internal Design Pressure (P)	2.750MPa
Internal Operating Pressure	2.50MPa
Internal Design Temperature	75^0C
Internal Operating Temperature	65^0C
External Design Pressure	NA
Inside Diameter	1368 mm
Tan Length of Vessel	3879 mm
Overall Length of Vessel	4500 mm
Corrosion Allowance	1mm (Internal)
Position For Hydro test	Horizontal
Design No. of Cycles for Data Case 1 (2.1 – 2.5MPa)	@ 1 x 10^8 Cycles
Design No. of Cycles for Data Case 2 (0 - 2.5MPa)	@ 1000 Cycles
Design No. of Cycles for Vertical Acceleration (+/-5g)	For N > 1 x 10^8 Cycles
Operating Frequency for Vertical Acceleration	0.33 Hz

Table 1: Design data for pressure vessel

A. Material for Construction:-

Components	Material Grade
Shell, Dish end, Lifting Lug, Lifting Lug Pad & Wear Plate for Saddle	SA 516 Gr. 70
Base Plate, Rib Plate, Web Plate, Washer, Bottom Plate for Saddle,	SA 36

Table 2: Materials for pressure vessel

Poisson’s Ratio for above materials is 0.3

B. Material Properties:

Material	Design Temperature	Elastic Modulus (MPa)
SA 516 Gr. 70	75^0C	199.33 x10^3
SA 36	75^0C	199.33 x10^3

Table 3: Material properties for pressure vessel

C. Chemical Composition of Material:

SA 516 Grades 70. (Carbon Steel Plate)			
Composition	%	Composition	%
C	0.10/0.20	P	0.03
Si	0.60	S	0.03
Mn	0.03	Al	0.02
Mo	0.08	Nb	0.01
Cr	0.03	Ti	0.03
Cu	0.30	V	0.02
Ni	0.30		

Table 4: Chemical Composition of SA 516 Grades 70.

<table>
<tr><th colspan="6">SA 36 (Carbon Steel Plate)</th></tr>
<tr><th>Thickness</th><th>C</th><th>Si</th><th>Mn</th><th>P</th><th>S</th></tr>
<tr><td>T ≤ 20</td><td rowspan="2">0.25</td><td rowspan="2">0.40</td><td>-</td><td rowspan="5">0.4</td><td rowspan="5">0.05</td></tr>
<tr><td>20 < T ≤ 40</td><td rowspan="2">0.8 -1.2</td></tr>
<tr><td>40 < T ≤ 65</td><td>0.26</td><td rowspan="3">0.15/0.40</td></tr>
<tr><td>65 < T ≤100</td><td>0.27</td><td rowspan="2">.85 -.12</td></tr>
<tr><td>T > 100</td><td>0.29</td></tr>
</table>

Table 5: Chemical Composition of SA 36

III. CALCULATIONS OF PRESSURE VESSEL AS PER ASME CODE, SECTION VIII, AND DIVISION 1, 2010

Pressure Vessel Dish End [LHS & RHS]

Material UNS Number: K02700

A. Inside Corroded Head Depth (h) :-

$$h = L - \sqrt{\left(L - \frac{D_i}{2}\right) \times \left(L + \frac{D_i}{2} - 2 \times r\right)}$$

$$h = 1321 - \sqrt{(546) \times (1635)}$$

$$h = 286.456 \text{ mm}$$

B. M factor for Torispherical Heads (Corroded):-

$$M = (3 + \sqrt{\frac{(L+C)/(r+C)}{4}})$$

$$M = (3 + \sqrt{\frac{(1230+1.00)/(140.00+1.00}{4}}$$

$$M = 1.4887$$

C. Required thickness due to internal pressure [t_r]:-

$$t_r = \frac{(P \times L \times M)}{(2 \times S \times E - 0.2 \times P} + C$$

$$t_r = \frac{(2.750 \times 1231 \times 1.4887)}{(2 \times 137.90 \times 1.00 - 0.2 \times 2.750)} + 1.0$$

$$t_r = 18.3114 + 1.00$$

$$t_r = 19.3114 \text{ mm}$$

D. Maximum Allowable Working Pressure at given thickness, corroded [MAWP]:-

$$MAWP = \frac{(2 \times S \times E \times t)}{(M \times L = 0.2 \times t)}$$

$$MAWP = \frac{(2 \times 137.90 \times 1.00 \times 19.00)}{(1.4887 \times 1231.00 + 0.2 \times 19.00)}$$

$$MAWP = 2.853 \text{MPa}$$

E. M factor for Torispherical Heads(New):-

$$M_{new} = (3 + \sqrt{\frac{L/r}{4}})$$

$$M_{new} = (3 + \sqrt{\frac{1230/140.00)}{4}})$$

$$M_{new} = 1.4910$$

F. Maximum Allowable Pressure [New]:-

$$P_{new} = \frac{(2 \times S \times E \times t)}{(M \times L + 0.2 \times t}$$

$$P_{new} = \frac{(2 \times 137.90 \times 1 \times 20)}{(1.4910 \times 1230 + 0.2 \times 20)}$$

$$P_{new} = 3.001 \text{MPa}$$

1) Actual Thickness at given pressure and thickness, corroded[S_{act}]:-

$$S_{act} = \frac{(P \times (M \times L + 0.2 \times t))}{(2 \times E \times t)}$$

$$S_{act} = \frac{(2.75 \times (1.4887 \times 1231 + 3.8))}{(2 \times 1.00 \times 19.00)}$$

$$S_{act} = 132.912 \text{MPa}$$

2) Straight Flange Required Thickness:-

$$t_{sf} = \frac{(P \times R)}{(S \times E - 0.6 \times P)} + c$$

$$t_{sf} = \frac{(2.750 \times 685)}{(137.90 \times 1 - 0.6 \times 2.75)} + 1.00$$

$$t_{sf} = 14.827 \text{ mm}$$

3) Straight Flange Maximum Allowable Working Pressure:-

$$MAWP_{SF} = \frac{(S \times E \times t)}{(R + 0.6 \times t)}$$

$$MAWP_{SF} = \frac{(137.90 \times 1.00 \times 24.00)}{(685.00 + 0.6 \times 24.00)}$$

$$MAWP_{SF} = 4.732 \text{Mpa}$$

a) Pressure Vessel [SHELL]:-

1) Required thickness due to internal pressure [t_r]:-

$$t_r = \frac{(P \times R)}{(S \times E - 0.6 \times P)}$$

$$t_r = \frac{(2.750 \times 685.00)}{(137.90 \times 1 - 0.6 \times 2.750)} + C$$

$$t_r = 13.8275 + 1.00$$

$$t_r = 14.8275 \text{ mm}$$

2) Maximum Allowable Working Pressure at given thickness, corroded[MAWP]:-

$$P_{MAX} = \frac{(S \times E \times t)}{(R + 0.6 \times t)}$$

$$P_{MAX} = \frac{(137.90 \times 1.00 \times 16.00)}{(684.00 + 0.6 \times 15.00)}$$

$$P_{MAX} = 2.981 \text{MPa}$$

3) Maximum Allowable Pressure,(New):-

$$P_{new} = \frac{(S \times E \times t)}{(R + 0.6 \times t)}$$

$$P_{new} = \frac{(137.90 \times 1.00 \times 16.00)}{(684.00 + 0.6 \times 16.00)}$$

P_{new}=3.181MPa

4) Actual stress at given pressure and thickness, corroded[S_{act}]:-

$$S_{act} = \frac{(P\times(R+0.6\times t))}{(E\times t)}$$

$$S_{act} = \frac{(2.750\times(685.00+0.6\times15.00))}{(1.00\times15.00)}$$

$$S_{act} = 127.249\text{MPa}$$

IV. CALCULATIONS OF PRESSURE VESSEL FOR FATIGUE ANALYSIS USING ASME CODE, SECTION VIII, AND DIVISION 2, 2010:-

Table 3.F.1 – Coefficients for Fatigue Curve 110.1 – Carbon, Low Alloy, Series 4XX , High Alloy Steels, And High Tensile Strength Steels For Temperature not Exceeding 371^{0}C (700^{0}F)

$\sigma_{uts} \leq 552$ MPa (80 ksi)

C_i	48 $\leq S_a <$ 214 (MPa) 7 $\leq S_a <$ 31 (ksi)	214 $\leq S_a \leq$ 3999 (MPa) 31 $\leq S_a \leq$ 580 (ksi)
1	2.254510E+00	7.999502E+00
2	-4.642236E-01	5.832491E-02
3	-8.312745E-01	1.500851E-01
4	8.634660E-02	1.273659E-04
5	2.020834E-1	-5.263661E-05
6	-6.940535E-03	0.0
7	-2.079726E-02	0.0
8	2.010235E-04	0.0
Note : E_{FC} = 195E3 MPa (28.3E3 ksi)		

Table 6: Coefficient of fatigue curves 110.1 ASME Section VIII, Div. 2, Part 3

A. Design Case:-

Case	Pressure1	Pressure 2	Range	No. of Cycles
1	2.10	2.500	0.400	1x10^8
2	0	2.500	2.500	1000

Table 7: Design case for pressure vessel

Stress	Longitudinal Plane		Transverse Plane	
	Inside Corner	Outside Corner	Inside Corner	Outside Corner
σ_n	3.100	1.200	1.000	2.100
σ_t	-0.200	1.000	-0.200	2.600
σ_r	$-\frac{t}{R}$	0.000	$-\frac{t}{R}$	0.000
σ	3.300	1.200	1.200	2.600

Table 8: Internal Pressure Loading

Stress	Longitudinal Plane		Transverse Plane	
	Inside Corner	Outside Corner	Inside Corner	Outside Corner
σ_n	3.100	1.200	1.000	2.100
σ_t	-0.200	1.000	-0.200	2.600
σ_r	-0.0219	0.000	-0.0219	0.000
σ	3.300	1.200	1.200	2.600

Table 9: Tabulated Internal Pressure Loading

B. Calculation for Peak Stress:

1) Compute Primary Membrane Stress [S]:

$$S = \frac{P}{\left(E \times \ln\left(\frac{2\times t+D}{D}\right)\right)}$$

$$S = \frac{0.400}{\left(1.00\times\ln\left(\frac{2\times15.00+1370.00}{1370.00}\right)\right)}$$

$$S = 18.4660\text{MPa}$$

2) Sample Calculation for the Intensified Stress Amplitude [S_a]

$$S_a = S\times\frac{3.3}{2}$$

$$S_a = 18.466\times\frac{3.3}{2}$$

$$S_a = 30.4688\text{MPa}$$

3) Stress Factor used to compute X [Y]:-

$$Y= \frac{S_a}{C_{us}}\times\frac{E_{FC}}{E_t}$$

$$Y= \frac{4.4}{1.00}\times\frac{28300000}{28952368}$$

$$Y= 4.319\text{ksi}$$

Compute from equation(X):-

$$X = \frac{C_1+C_3\times Y+C_5\times Y^2+C_7\times Y^3+C_9\times Y^4}{1+C_2\times Y+C_4\times Y^2+C_6\times Y^3+C_8\times Y^4}$$

$$X = 11.1453$$

C Factors used in the above equation:

C1 = 2.25451 C2 = -.464224 C3= -.831275
C4 = 0.863466E-01 C5 = 0.202083
C6 = -.694053E-02 C7 = -.207973E-01
C8 = 0.201024E-03 C9 = 0.713772E-03
C10 = 0.00000 C11 = 0.00000

From the table, E_{FC} = 195128MPa

Compute the Number of Cycles [N]:-

$$N = 10^X$$

$$N = 10^{11.1453}$$

$$N = 1.394 \times 10^{11}$$

Case 1 Peak Stress: Adjusted below per above Pressure Index table:-

Stress	Longitudinal Plane		Transverse Plane	
	Inside Corner	Outside Corner	Inside Corner	Outside Corner
σ_n= 9.233	28.622	11.080	9.233	19.389
σ_t = 9.233	-1.847	9.233	-1.847	24.006
σ_r = 9.233	-0.202	0.000	-0.202	0.000
σ= 9.233	30.469	11.080	11.080	24.006

Table 10: Peak stress for data case 1

Stress	Longitudinal Plane		Transverse Plane	
	Inside Corner	Outside Corner	Inside Corner	Outside Corner
σ_n= 57.706	178.88	69.247	57.706	121.183
σ_t = 57.706	-11.541	57.7.6	-11.541	150.036
σ_r = 57.706	-1.264	0.0	-1.264	0.000
σ= 57.706	190.43	69.247	69.247	150.036

Table 11: Peak stress for data case 2.

Case	Stress Intense	N cycles	N_{Max} cycles	Damage Factor
1	30.469	1x10^8	0.1390x10^{12}	0.001
2	190.430	1000	0.3080x10^5	0.032
Total Damage Factor:**0.033**				

Table 12: Fatigue damage factor for data case 1 & 2.

V. CONCLUSION

From design data we find out allowable cycles for data case 1 & 2 comparing with design cycle, allowable cycles for data case 1& 2 are more as compare to the design cycles and finding total fatigue factor for data case 1 & 2 are 0.033 which is less than 1.Hence we say that our design is safe because fatigue damage factor is less than unity.

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