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Fatigue Analysis of 6300 Liters Pressure Vessel By Using Cyclic Service

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Abstract — High pressure is developed in pressure vessel so pressure vessel has to withstand several forces developed due to internal pressure so selection of pressure vessel is most critical for safety purpose. The 6300 Liters pressure vessel has to be designed according to ASME standards. In this paper fatigue analysis including stress analysis. This study performing fatigue analysis during arbitrary transients and a result are applied to a finite element model along with the pressure to calculate the stress and deformation values also estimate number of cycles as per ASME standard. Fatigue analysis was done to calculate allowable useful life cycle of the vessel against the designed cycle.

Keywords-component; Pressure Vessel, Fatigue Damage Factor, Stress Analysis, Fatigue Penalty Factor, Fatigue Strength Reduction Factor.

I. INTRODUCTION

A pressure vessel is defined as a container with pressure differential between inside and outside, except for some isolated situations. High pressure is developed in pressure vessel so pressure vessel has to withstand several forces developed due to internal pressure so selection of pressure vessel is most critical. The fluid inside pressure vessel may undergo state of change like in case of boilers. Pressure vessel has combination of high pressure together with high temperature and may with flammable radioactive material because of hazards it is important to design pressure vessel such that no leakage can take place as well as pressure vessel is to be designed carefully to cope with high pressure and temperature plant safety and integrity are of fundamental concern in pressure vessel design and these depends on adequacy of design codes. For safety purpose pressure vessel has to be designed according to ASME standards. The life of vessel under cyclic service is related to the intensity of the number of cycles it is exposed to. The fatigue life curves used under ASME VIII division-2 to calculate the permitted cycle life of a vessel are based on a large factor of safety compared with actual cycle life curves. We are using VIII division-2 fatigue method to calculate an allowable number of operating cycles with a factor of safety, not to predict the cycle life of the vessel which normally will be larger. The determination of the need for a fatigue evaluation is in itself a complex job best left to those experienced in this type of analysis.

II. DESIGN OF PRESSURE VESSEL AS PER ASME CODES

DESIGN DATA			
Design Drawing			
Design Code	ASME Section VIII, Division 1 & 2 Ed.2010		
1. Fluid in Service	AIR	6. Inside Diameter	1368 mm
2. Internal Design Pressure (P)	2.750MPa	7. Tan Length of Vessel	3879 mm

3. Internal Operating Pressure	2.50MPa	8. Overall Length of Vessel	4500 mm
4. Internal Design Temperature	75^{0}C	9. Corrosion Allowance	1mm (Internal)
5. Internal Operating Temp.	65^{0}C	10. 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	
Operating Frequency for Vertical Acceleration		0.33 Hz	

Materials of Construction & Properties for Analysis

Components	Material Grade	Design Temp.	Elastic Modulus
Shell, Dish end, Lifting Lug, Lifting Lug Pad & Wear Plate for Saddle	SA 516 Gr. 70	75^{0} C	199.33 x10^{3}
Base Plate, Rib Plate, Web Plate, Washer, Bottom Plate for Saddle,	SA 36	75^{0} C	199.33 x10^{3}

Poisson's Ratio for above materials is 0.3

III. FEA OF 6300 LITERS PRESSURE VESSELS

The objective of analysis was to check fatigue life of 6300Ltr Air Receiver for cyclic pressure service and impact loading service in accordance with ASME Section VIII, Div-2 Part 5 Ed 2010.To study the stress levels, finite element based stress analysis is carried out.3D model is created including geometric details for corroded condition as per drawing.

i. Model is analyzed with given data cases.
   1. Data case 1 – Cyclic Pressure Service of 2.1MPa to 2.5MPa.
   2. Data case 2 - Cyclic Pressure Service of 0 to 2.5MPa.

ii. Fatigue calculations performed based on the results of FEA analysis as per ASME Section VIII, Division 2, Part 5.

3.1. Finite Element Model

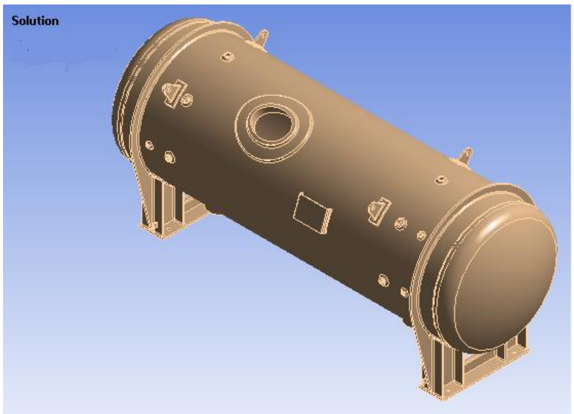

Figure 3.1 3D CAD Model

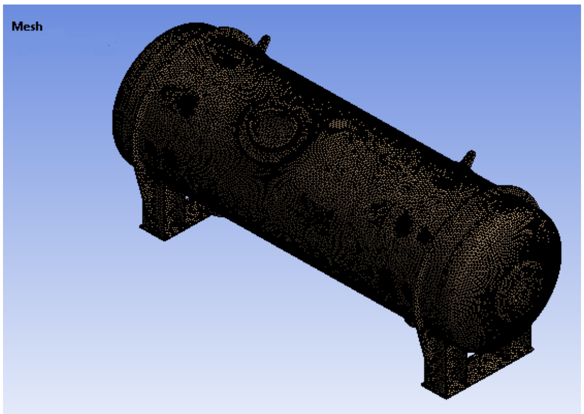

Figure 3.2 Meshed FEA Model

Finite Element Model is based on 3D CAD model of 6300L air receiver, consisting of Shell, Dish ends, Nozzles, Lugs & saddle supports. The 3D geometry is meshed using Solid 187 having element size of 30mm.Total numbers of elements are – 469825

3.1.1. Loading for data case 1.

Internal Pressure of 2.1 & 2.5MPa are applied as a surface load on all internal pressure retaining surfaces of the air receiver.

3.1.2. Result for data case 1

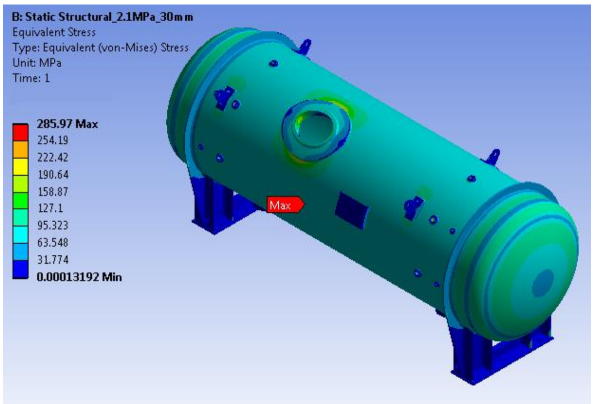

Fig 3.3: Von Misses plot for 2.1MPa

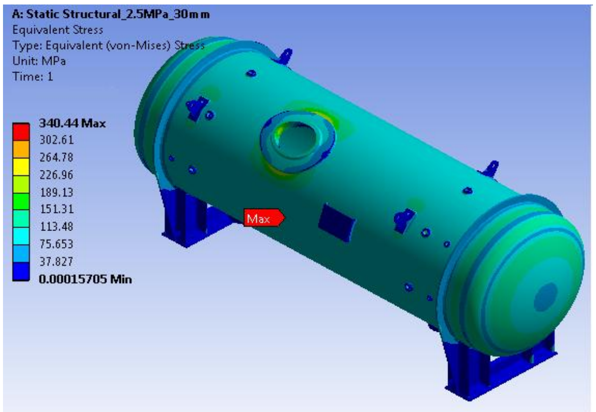

Fig 3.4: Von Misses plot for 2.5MPa

Maximum stress of magnitude 285.97MPa & 340.44MPa occurs at the fillet of nozzle.

3.1.3. Deformation plot

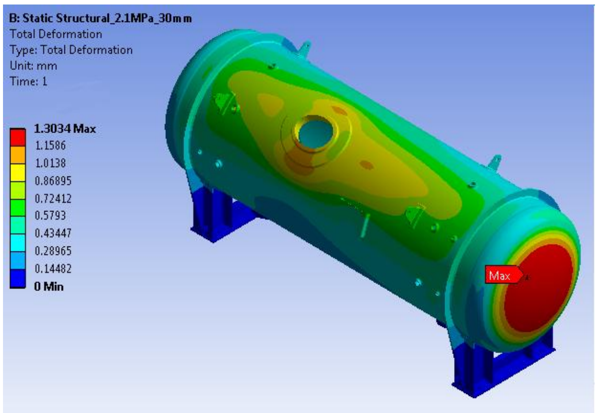

Fig 3.5: Deformation plot for 2.1MPa

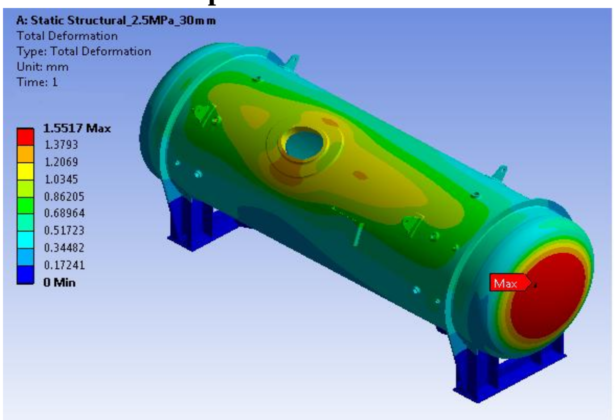

Fig 3.6: Deformation plot for 2.5MPa

Maximum deformation of 1.3034 & 1.5517mm occurs near dish end.

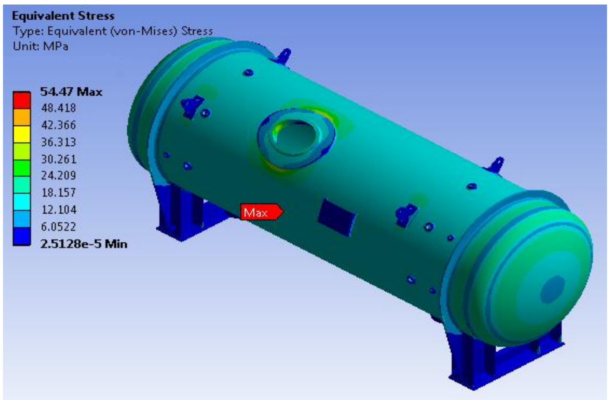

Fig 3.7: Stress range plot for data case 1

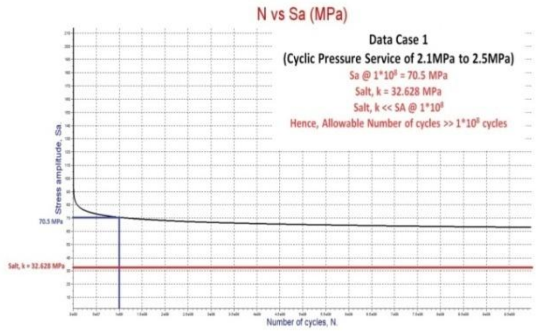

Fig 3.8: Graph (N vs. S_a (MPa))

Stress range plot for Data Case 1 with magnitude 54.47MPa.

3.2 Boundary Condition

Entire Model: The entire air receiver will be mounted through fixed saddle supports.

IV. FATIGUE LIFE CALCULATION ACCORDING TO ASME SECTION VIII, DIVISION 2, PART 5, 2010.

4.1 Data case 1(Cyclic Pressure Service of 2.1MPa to 2.5MPa)

According to design data, for data case 1 (Cyclic Pressure Service of 2.1MPa to 2.5MPa) design number of cycles are 1×10^8.According to point number 5.5.3.2 (ASME Section VIII, Division 2, Part 5, Point 5.5.3.2) the effective alternating stress amplitude for the k^{th} cycle.

$$S_{alt,\,k} = \frac{K_f . K_{e,k} . \Delta S_{p\,,k}}{2}$$

Where,

$\Delta S_{P,\,k} = 54.47\ N/mm^2$

K_f = Fatigue Strength reduction factor = 1.2 (ASME Section VIII, Div 2, Table 5.11)

$K_{e,k}$ = Fatigue Penalty Factor = 1 (since $\Delta S_{n,k} < \Delta S_{PS}$; i.e. 54.47 < 500MPa (max [3S, 2Sy]))

After solving this we get,

$S_{alt,\,k} = 32.682\ N/mm^2$

As seen in graph figure 2.9 value of $S_{alt,k}$ is much less than value of S_a at 1×10^8 cycles. Hence allowable numbers of cycles for present data case for entire model are more than 1×10^8 cycles.

4.2 Data case 2(Cyclic Pressure Service of 2.1MPa to 2.5MPa)

$$S_{alt,\,k} = \frac{K_f . K_{e,k} . \Delta S_{p\,,k}}{2}$$

Where,

$\Delta S_{P,\,k} = 340.44\ N/mm^2$

K_f = Fatigue Strength reduction factor = 1.2(ASME Section VIII, Div 2, Table 5.11)

$K_{e,\,k}$ = Fatigue Penalty Factor = 1 (since $\Delta S_{n,k} < \Delta S_{PS}$; i.e. 340.44 < 500MPa (max[3S, 2Sy]))

After solving this we get,

$S_{alt,\,k} = 204.264\ N/mm^2$

To calculate the design number of cycles following formula is used

The design number of design cycles, N, can be computed from Equation (3.F.1) or Table 3.F.10 based on the stress amplitude S_a which is determined in accordance with Part 5 of this Division

$$N = 10^X$$

Where,

$$X = \frac{C_1 + C_3 Y^1 + C_5 Y^2 + C_7 Y^3 + C_9 Y^4 + C_{11} Y^5}{1 + C_2 + C_4 Y^2 + C_6 Y^3 + C_8 Y^4 + C_{10} Y^5}$$

$$Y = \left\{\frac{S_a}{C_{us}}\right\} \times \left\{\frac{E_{FC}}{E_T}\right\}$$

Where E_T = Young's modulus for material.

The coefficients C1, C2... are calculated from table 4.1 – Coefficient of fatigue curves.

Carbon, Low Alloy, Series 4XX, High Alloy Steels, and High Tensile Strength Steels for Temperature not exceeding 371^0C $\sigma_{uts} \leq 552$ MPa (80 ksi)

Table 4.1 – **Coefficient of fatigue curves 110.1 (ASME Section VIII, Division 2, Part 3)**

Coefficients C_i	**$48 \le S_a < 214$ (MPa)** **$7 \le S_a < 31$ (ksi)**	**$214 \le S_a \le 3999$ (MPa)** **$31 \le S_a \le 580$ (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
9	7.137717E-04	0.0
Note : E_{FC} = 195E3MPa (28.3E3ksi)		

After solving the above equations we get,
N = 25577
Fatigue damage factor = 1000 / 25577 = 0.03909
This factor is much less than unity.
Design is safe.

V. ASME SECTION VIII, DIVISION 2 2010, FATIGUE ANALYSIS

Case	Pressure 1	Pressure 2	Range	Number of Cycles
1	2.100	2.500	0.400	100000000
2	0	2.500	2.500	1000

Calculation for Pressure Range:
Compute Primary Membrane Stress [S]:
= P / (E x ln((2 x t + D) /(D)))
= 0.400/ (1.00 x ln ((2x15.000+1370.000)/(1370.000)))
= 18.4660MPa
Sample calculation for the Intensified Stress Amplitude [S_a]:
= S x 3.3 / 2
= 18.466 x 3.3/2
= 30.4688MPa
Stress Factor used to compute X [Y]:
= $(S_a/C_{us}) \times (E_{FC}/E_t)$ Imperial Units 3.F.3
= (4.4/1) x (28300000/28952368)
= 4.3194ksi
Compute from Equation 3.F.2 [X]:
= $(C_1+C_3 x Y+C_5 x Y^2+C_7 x Y^3+C_9 x Y^4+C_{11} x Y^5)/$
$(1+C_2 \text{ x } Y+C_4 x Y^2+C_6 x Y^3+C_6 x Y^4+C_{10} x Y^5)$
= 11.1453
From the table, E_{FC} = 195128MPa

Compute the Number of Cycles from Equation 3.F.1 [N]:
$= 10^{X}$
$= 10^{11.145}$
$= 1.396368 \times 10^{11}$ Cycles

CONCLUSION

Fatigue analysis is carried out for entire equipment for specified regeneration cycles and found fatigue life more than required cycles. Accordingly we conclude that all evaluation points for fatigue are within allowable limits specified by code. The maximum fatigue damage fraction observed which less than unity as required by code. It will determine the safety of the vessel design prior to manufacturing; thereby reducing the manufacturing time significantly, prevented the probability of fatal accidents through fatigue analysis. Changes in the design for strength improvement made easier by providing information on vessel regions requiring geometrical changes.

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