Life Estimation of Pressure Vessel as per ASME Section VIII Div 2

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Abstract— Pressure vessels are important engineering component in industry. Fatigue failure most common failure in pressure vessels. Aim of the project is to design and estimate fatigue life of pressure vessels under cyclic loading. Design of pressure vessels according to ASME Sec. VIII Div.1 Edition 2017 and design calculation verified with PV Elite software. ASME Sec VIII DIV 1 is used for design calculation and DIV 2 Edition 2017 is used for fatigue life estimation purpose. New methodology is introduced in Edition 2017 by ASME, with new methodology fatigue is estimated . The operating cycles are having pressure range of 0 MPa to 1.5 MPa for the same fatigue life is estimated as per ASME Sec. VIII Div.2.

Keywords—*Pressure vessel, Fatigue life, Finite element analysis, Ansys Workbench,ASME Sec VIII Div 1 &2 .*

I. INTRODUCTION

The pressure vessels are used to store fluids under pressure. Pressure vessels find wide applications in thermal and nuclear power plants, process and chemical industries, in space and ocean depths and in water, steam, gas and in air supply system, in process Industries like chemical and petroleum industries. The term pressure vessel referred to those reservoirs or containers, which are subjected to internal or external pressures. The fluid being stored may undergo a change of state inside the pressure vessels as in case of steam boilers or it may combine with other reagents as in chemical plants. The material of a pressure vessel may be brittle such as cast iron, or ductile such as mild steel.

Construction of Pressure Vessels:

Fig (1): Pressure vessel

The fabrication process has been divided into steps. Before the fabrication process is begun, a few important design steps must be performed. These are given below in brief.

List the Design Criteria: This is where all the technical specifications and requirements are taken from the client. The design criteria will include aspects such as the shape of the vessel, and details such as material of construction, length, diameter, internal pressure & temperature, etc.

Perform Mechanical Strength Calculations: With pressure vessel design software, the mechanical calculations are performed to determine the required material thicknesses and weld sizes.

Generate Fabrication Drawings: Fabrication drawings are produced based on the design criteria and mechanical calculations with software such as AutoCAD. These drawings can be produced in 2D or 3D, but usually 2D will suffice.

Once the design is approved by the client and a third party inspector (A.I.), the fabrication process begins.

Cut The Plate: A large amount of plate is cut into the required width and length for the vessel shells. A specialized cutting torch is used to cut the steel. A variety of torches can be used depending on material and required edge quality. Bevels are usually cut on edges at this time.

Roll the Plate into Cylinders: The plate is rolled into cylinders of required diameter. Usually the plate is rolled cold, but can be done hot in order to use plate rolls of a smaller capacity.

Weld Long Seams of Cylinders: Usually performed with submerged arc welding.

Fit & Weld the Pressure Vessel Cylinders: Usually performed with submerged arc welding. This is only needed if more than one shell is required to meet the required vessel length.

Cut and Form the Pressure Vessel Heads: Various cutting torches can be used to cut the plate for the pressure vessel head. Then the pressure vessel heads are formed using a variety of techniques, usually flanging and spinning or press forming is used.

Fit & Weld Heads to Pressure Vessel Shell: Usually performed with submerged arc welding.

Cut Holes For Nozzles: Holes need to be cut into the steel plate for the required vessel nozzles. This is usually performed with a manual plasma-arc or oxyacetylene cutting torch. Final edge preparations performed by manual grinding.

Install Nozzles: Nozzles and reinforcement pads are welded in usually using a manual wire welding technique.

Install Required Structural Supports and Lifting Devices: These are usually welded on using a manual wire welding technique.

Final Quality Checks: This could include hydro, pneumatic, radiography, ultrasonic, and or magnetic particle testing to ensure integrity of design, material, and welding.

Final Inspections: This usually includes internal, customer, and third party inspections.

II. OBJECTIVE

- To design the pressure vessel using ASME Sec VIII DIV 1 Edition 2017.
- To estimate fatigue life with constant amplitude pressure fluctuation cycles i.e. 0 MPa to 1.5 MPa as per ASME VIII Div 2 Edition 2017.

III. METHODOLOGY

Aim of this project is to estimate the fatigue life of pressure vessel for constant amplitude pressure fluctuation cycles of pressure vessel i.e. 0 MPa to 1.5 MPa. We have calculated all the pressure part thickness for given design parameters and with same dimensional details 3D CAD model is prepared.3D CAD model is imported into ANSYS workbench apply loading and boundary condition to find out Von-mises stresses in vessel. As per part 5 of ASME SEC VIII DIV 2 we have calculated number cycles.

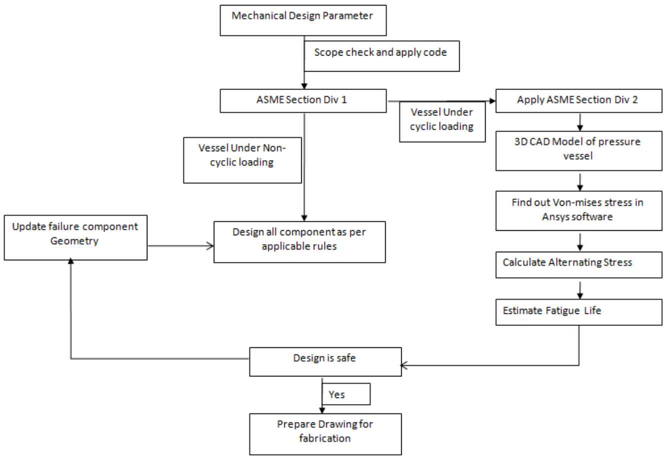


Fig (2): Methodology Flowchart

Following mechanical design parameters are used for research purpose.

Design Code	ASME SEC VIII DIV 1 & 2
Fluid is in used	Water
Internal Design Pressure	1.5 MPa
Internal Design Temperature	77°C
Outside Diameter	1528 mm
Tan Length of vessel	3500 mm
Corrosion Allowance	0 mm

Material	Density (Kg/m3))	Allowable Stress (MPa)	Yield Strength (MPa)	Passion's Ratio	Modulus of Elasticity (MPa)
SA-240 304L (Shell,D/E)	8030	115	150	0.31	1.98E+05
SA-312 TP304 (Pipe)	8030	115	150	0.31	1.98E+05
SA-182 F304 (Flanges)	8030	115	150	0.31	1.98E+05

All mechanical properties are taken at design temprature.

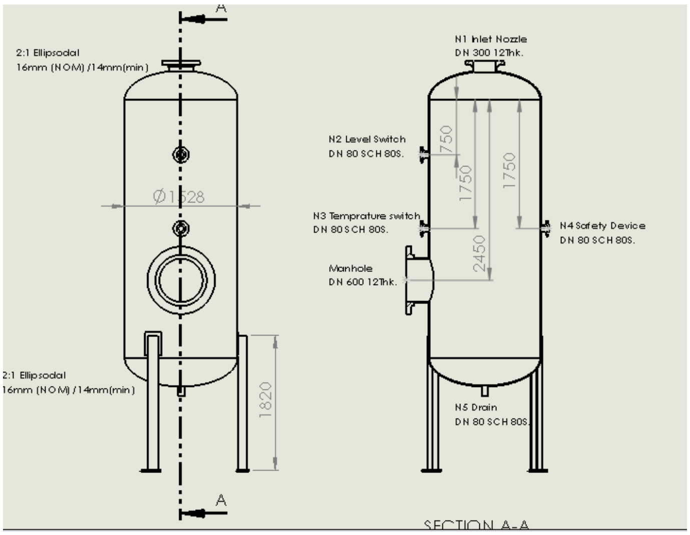


Fig (3): 2D sketch

IV. DESIGN CALCULATION

From ASME Section VIII, div-I, UG27
S=138Mpa
E=1=longitudinal seam efficiency
Pi=1.5Mpa=Internal pressure
R= 750 mm = Internal Diameter

$$t=\frac{PR}{SE-0.6P}$$

t= 9.8511 mm
Provided thickness = 14 mm, Hence design is safe.
Thickness of Ellipsoidal head:

$$t=\frac{PD}{2SE-0.2P}$$

t = 12.94 mm
Provided thickness = 14 mm, Hence design is safe.

V. FINITE ELEMET ANALYSIS

Aim of this project is to estimate the fatigue life of pressure vessel for constant amplitude pressure fluctuation cycles of pressure vessel i.e. 0 MPa to 1.5 MPa. We have calculated all the pressure part thickness for given design parameters and with same dimensional details 3D CAD model is prepared.3D CAD model is imported into ANSYS workbench apply loading and boundary condition to find out Von-mises stresses in vessel. As per Part 5 of ASME SEC VIII DIV 2 Edition 2017 we have calculated number cycles.

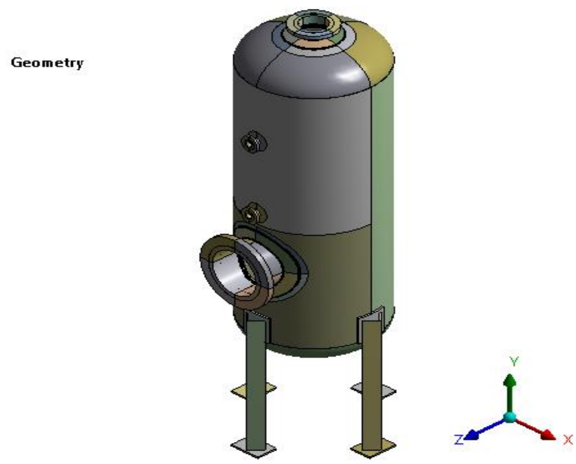


Fig (4): 3D CAD Model

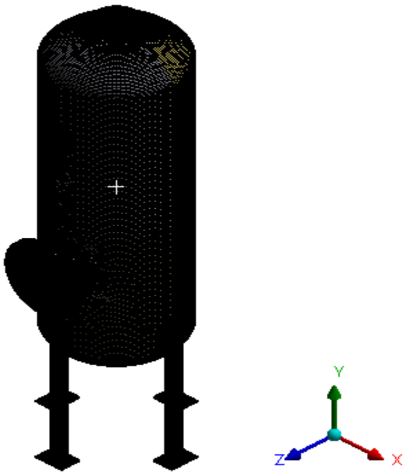


Fig (5): Meshing

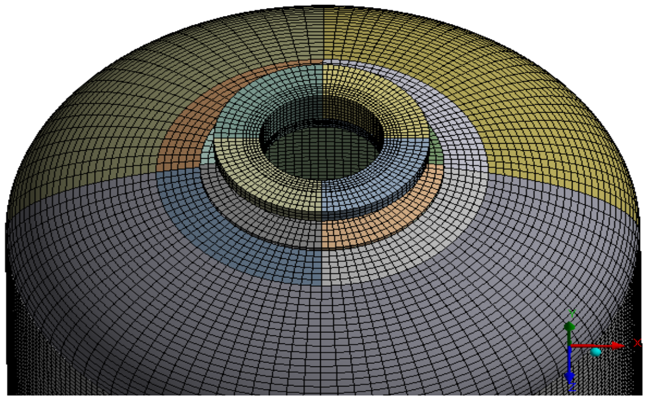


Fig (6): Zoomed Image of meshing

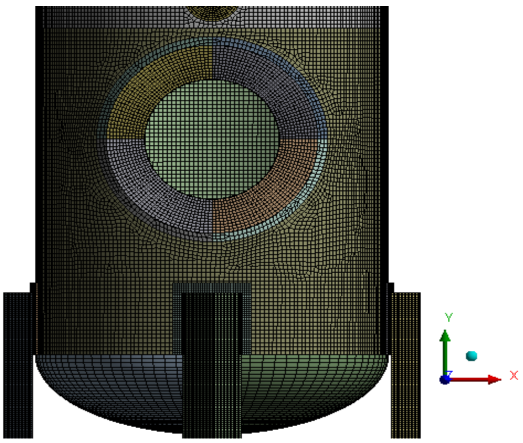


Fig (7): Zoomed Image of meshing

For this analysis Solid 186 element type is considered.

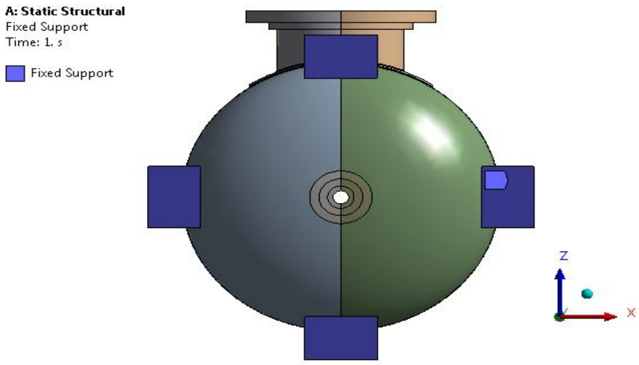


Fig (9): Boundary Condition

Fixed support is applied at bottom base support of legs.

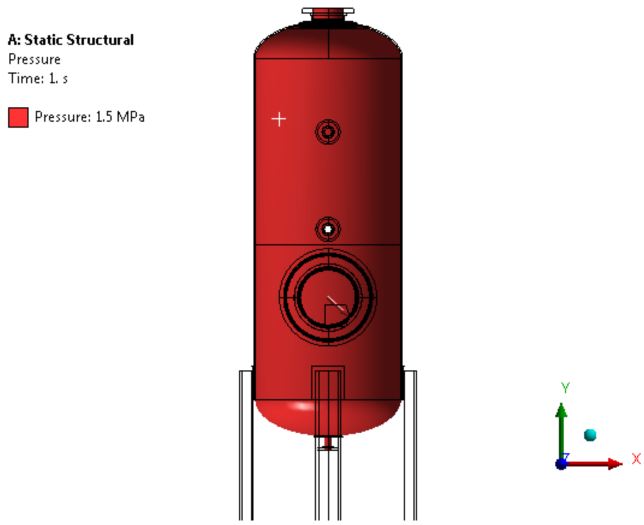


Fig (10): Internal Pressure (1.5 MPa)

1.5 MPa internal pressure is applied to internal faces of pressure vessel.

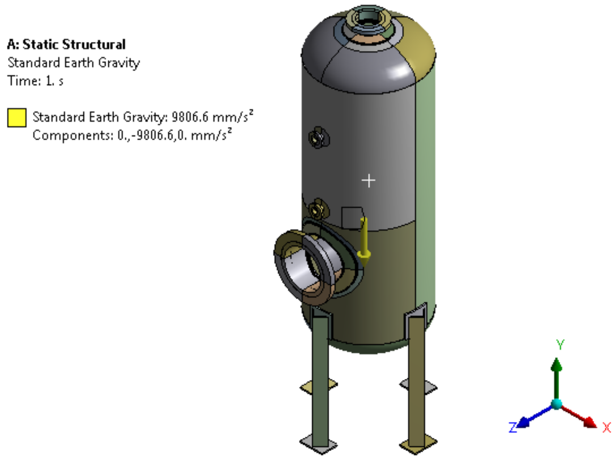


Fig (11): Standard earth gravity

Dead weight of equipment is considered in terms of standard earth gravity.

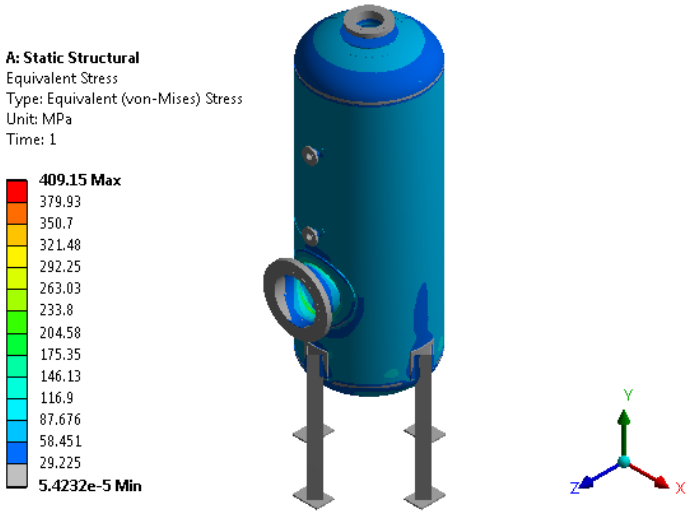


Fig (12): Von-Mises stresses

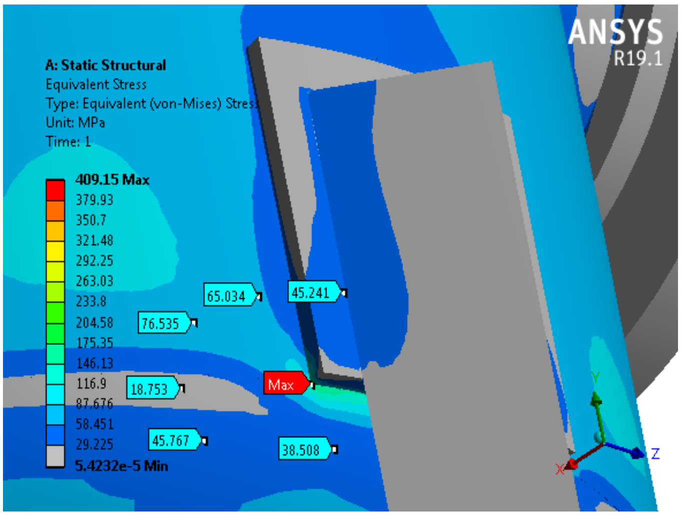


Fig (13): Zoomed view of Max. Von-mises stresses

409.15 MPa Max. Von-mises induced in vessel for given loading and boundary condition.

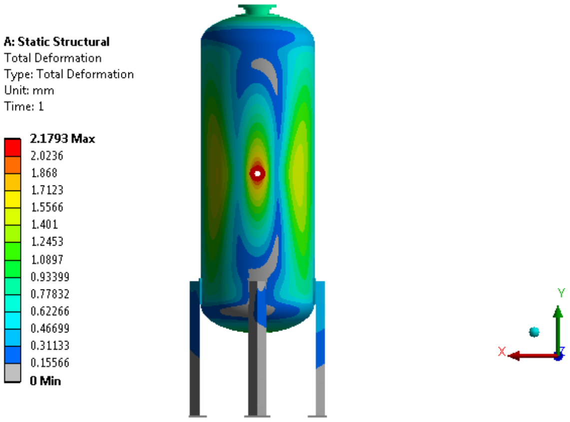


Fig (14): Deformation plot

2.179 mm deformation is observed.

VI. FATIGUE CALCULATION

The Fatigue assessment is carried out by Elastic stress analysis and equivalent stresses – Effective total equivalent stress amplitude is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis. The controlling stress for the fatigue evaluation is the effective total equivalent stress amplitude, defined as one-half of the effective total equivalent stress range calculated for each load case.

The maximum von mises stress for the Case 1 is 0 MPa

The maximum von mises stress for the Case 2 is 1.5 MPa

The operating cycles are having pressure range of 0 MPa to 1.5 MPa so the effective total equivalent stress range

SΔpk = max. Stress at Case 2 – max. Stress at Case 1

= 409 - 0 = 409 MPa

The fatigue strength reduction factor (Kf) for weld is selected of Section VIII Div. 2 Kf = 1

ΔSn,k < ΔSps, the fatigue penalty factor Ke,k = 1.0

Therefore the alternating equivalent stress for the kth Cycle is calculated as follows

$$S_{alt,k} = \frac{K_f \cdot K_{e,k} \cdot \Delta S_{P,k}}{2}$$

$$\text{Salt,k} = \frac{1 \times 1 \times 409}{2}$$

Salt,k = 204.5 MPa

As per Annex 3F ASME SEC VIII DIV 2 Edition 2017

3-F.1.2

Fatigue analysis performed using smooth bar fatigue curve models in equation form is provided below. The fatigue curves and the associated equations for different materials are also shown below.

(a) Carbon, Low Alloy, Series 4XX, High Alloy, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F). The fatigue curve values may be interpolated for intermediate values of the ultimate tensile strength.

$$Y = \log\left[28.3E3\left(\frac{S_a}{E_T}\right)\right] \quad (3\text{-F.1})$$

(1) For $\sigma_{uts} \le 552$ MPa (80 ksi) (see Figures 3-F.1M and 3-F.1)

$$X = -4706.5245 + 1813.6228Y + \frac{6785.5644}{Y} - 368.12404Y^2 - \frac{5133.7345}{Y^2} + 30.708204Y^3 + \frac{1596.1916}{Y^3} \text{ for } 10^Y \ge 20 \quad (3\text{-F.2})$$

$$X = \frac{38.1309 - 60.1705Y^2 + 25.0352Y^4}{1 + 1.80224Y^2 - 4.68904Y^4 + 2.26536Y^6} \text{ for } 10^Y < 20 \quad (3\text{-F.3})$$

$$Y = \log\left(28E3 \times \frac{204.5}{186E3}\right)$$

Y = 1.47

$10^Y > 20$ formula 3-F.2 is applicable.

29.51 > 20 formula 3-F.2 is applicable.

X = 4.35

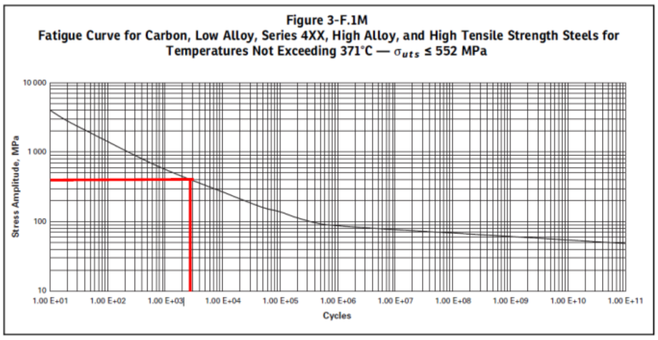
3-F.1.3

The design number of design cycles, N, can be computed from eq. (3-F.21) based on the parameter X calculated for the applicable material.

$$N = 10^X \quad (3\text{-F.21})$$

$N = 10^{4.35}$

N = 22390 cycles



Allowable No. of cycles are also verified by SN curve for respective material. As per SN Curve vessel can withstand 22390 cycles for given boundary condition.

CONCLUSION

Vertical pressure vessel can withstand 22390 pressure fluctuating cycles for given condition in accordance ASME VIII DIV 2 Edition 2017.

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