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Design, Optimization and Numerical Analysis of Pressure Cylindrical Assembly

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ABSTRACT

Pressure vessels have been widely used for many years in chemical, petroleum, air industries as well as in nuclear power plants. They are usually subjected to high pressures and temperatures which may be constant or cycling. Factors such as vessel material, shape, chemical composition, seismic loads, nozzle loads, vessel mounting, environment of vessels etc. can have different influence on performance of pressure vessels. The fluid being stored may undergo a change of state inside the pressure vessel as in case of steam boilers or it may combine with other reagents as in a chemical plant.

The pressure vessels are designed with great care because rupture of pressure vessels means an explosion which may cause loss of life and property. The problem concerns with the deep investigation of the stress at bottom junction intermediate tori cone joint of the vertical dual shell pressure vessel made up of Inconel 600. The tori cone joint is the component which connects between high and low pressure cylinder in the vertical cylindrical pressure vessel. Intermediate failure is not acceptable in the running condition and thus redesign effort was performed to achieve the positive design margin. This was achieved through shape optimization using FE Software.

Keywords – *Pressure vessel, Seismic Loads, Yield Criteria, Teri cone, Nozzle.*

1. INTRODUCTION

Pressure vessel is equipment which is used to store the liquid or change of state of the fluid at a pressure irrelevant from the atmospheric pressure. The pressure vessel can be either cylindrical, conical shape or spherical in shape which depends on the type of application it is used for. The auxiliaries include nozzles, doubler, pressure pads, flange connections.

Safety is one of the main concerns while designing any pressure vessel, as it contains the fluid at high temperature and pressure. The failure of the vessel will cause loss of life or may damage the property and may cause the health hazard if it is holding the chemical at higher temperature and pressure. It usually stores the fluid at the pressure different from that of the atmospheric pressure.

In industrial plants, fabricating of the dual chambered vessel is done by welding such as submerged arc welding at intermediate tori cone joint. This causes high stresses due to larger welding area at the junction thus weakening the weld region which interns results in failure of the pressure vessel. Hence, by optimizing the weld area we can increase the strength of the pressure vessel.

Fig 1: Pressure Vessel.

Most of the accidents (about 80%) for pressure vessel are resulted from the stress concentration. The associated stress concentration depends on the shape, size and location. The stress concentration effect a life time of pressure vessel crucially. It is very important to minimize the stress-raising effect at the opening or the discontinuous. The optimization has become a significant development area in the pressure vessel design. In general, the purpose of shape optimization is used for finding the best structure with various constraints. In the present research topology optimization is carried out in Ansys workbench 14.5v, has been used in determining the optimum pressure vessel profile which is best suited to sustain or distribute the load in proper manner.

Complicated vessel shape are difficult to manufacture and also unsafe to operate at the given operating conditions. Practically a spherical vessel is complicate from the point of design and manufacture. Thus the most preferred vessel is the cylindrical shape as it is easy and simple in design. It consists of two semielliptical end caps or heads. The ideal economic shape of a cylindrical shaped vessel will be of “1,000 liters (35 cu ft), 250 bars (3,600 psi) and the diameter of may vary from as small as 914.4 millimeters (36 in) and a length of 1,701.8 millimeters (67 in)” including the two end caps or heads. The material selected for the pressure vessel is usually steel whereas in case of spherical vessels, forged parts are welded together.

Vessels which are preferred in smaller applications such as pneumatics consists of pipe arranged with two end caps. It is used in nuclear and thermal power-plants, chemical plants, military, petrochemicals, autoclaves space, air receivers and marine applications, water storage tanks in household applications. In industrial sector, the vessels are used to store the liquid safely at a particular temperature and pressure known as in technical terms known as “design-pressure” and “design- temperature”. Certification and design of “Pressure vessels” is explained by the design codes such as the “ASME Boiler and Pressure Vessel Code in North America”, “The Pressure Equipment Directive of the EU (PED)”, “CSA B51 in Canada, Japanese Industrial Standard (JIS)”, “AS1210 in Australia and other international standards like Lloyd's, Germanischer Lloyd, Stoomwezen, Det Norske Veritas” etc.

In Vertical type Dual Chambered Pressure Vessels, we study the problem of the stress beyond the yield strength of the Inconel 600 material and this study helps to understand the problem in detail with the parameters involved and implementing the necessary solution for the same.

2. GEOMETRY AND FE MODELING

This chapter discusses the geometric creation and computer aided modeling of the intermediate tori cone joint of the pressure vessel. The model consists of the tori cone with top shell and bottom shell, and it is created as per drawing shown in the Fig 2.1 using CATIA V5R20. It consists of the two tori spherical D'ends top and bottom, skirt and tori cone.

2.1 Geometry Details

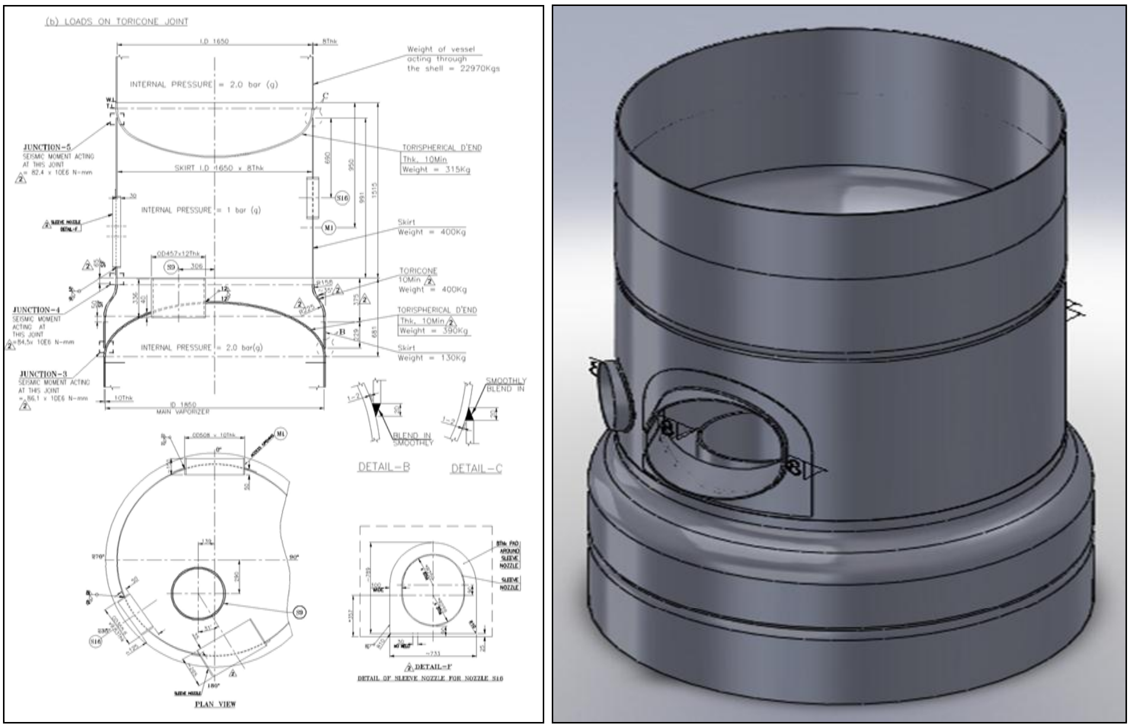


Fig 2.1: CAD drawing and model of the Tori Cone Joint.

The tori cone of the pressure vessel consists of openings which are connected to the various lines and carries air at an operating temperature of 80^0C.

Tori cone joint is, made up of Inconel 600 material and having below geometrical details:

- Internal diameter of top chamber = 1650 mm
- Internal diameter of Skirt = 1650 mm
- Internal diameter of top chamber = 1850 mm
- Thickness = 8 mm
- Height of Skirt = 1515 mm
- Height of bottom chamber = 681 mm

2.2 Material Modeling

Inconel 600 is an alloy of nickel and chromium. It provides good resistance against oxidation and also to corrosion at high temperatures.The chemical combination consists of nickel (Ni) 72%, Chromium (Cr) 14 -16 % and Iron (Fe) 6 -10 %.

Material	Inconel 600
Temperature	80^0C
Young’s modulus, E	$2.9e^7$ Psi
Poisson’s ratio, ν	0.29
Density, ρ	0.304 lb/in^2 = 0.0020 MPa
Coefficient of linear thermal expansion, α	$6e^{-6}$ $mm/^0C/$ mm
strength (σ_y)	35 ksi, 241.31 MPa
Ultimate strength (σ_u)	78 ksi, 537.79 MPa
Endurance strength (σ_e)	26 ksi, 179.26 MPa
Melting point	1413^0C
Coefficient of Expansion	13.3μm/m.^0C
Modulus of Rigidity	75.6 kN/mm^2
Modulus of Elasticity	206 kN/mm^2

Table 2.1: Properties of the Inconel 600.

Properties of Supplied and Heat Treated Materials :

Condition of Supply	**Heat Treatment (after forming)**
Annealed/Spring Temper	Stress relive at 460^0C (860^0F) for 1hr and air cool.

Table 2.2: Properties of Supplied Materials and Heat Treated Materials

Condition	**Approx Tensile Properties**		**Approx Sevice Temperature**	
Annealed	600-750 N/mm^2	87-109 ksi	-200 to $+330^0C$	-330 to $+625^0F$
Spring Temper	900-1250 N/mm^2	131-181 ksi	-200 to $+330^0C$	-330 to $+625^0F$

Table 2.3: Properties of Tensile Properties at Service Temperature.

Note – In both cases slight magnetism may occur below -120^0C (-184^0F).

2.3 FE Modeling

Even though the hex-predominant meshing is difficult we have adopted the hexa meshing to aviod the varatiuon in stress due to FE modelling and to achieve the accurate results. Hex-predominant is a powerful and exceedingly robotized unstructured cross section generator that can deal with lattices of essentially boundless size and unpredictability, comprising basically of hexahedral components yet including triangular or pyramidal cells. It uses propelled lattice calculations to permit the most fitting cell sort to be utilized to create body-fitted lattices for the most broad CAD geometries.

Indeed, the hex-predominant mesher is obliged if your model incorporates CAD formed articles, ellipsoids, circular barrels, or polygonal conduits.

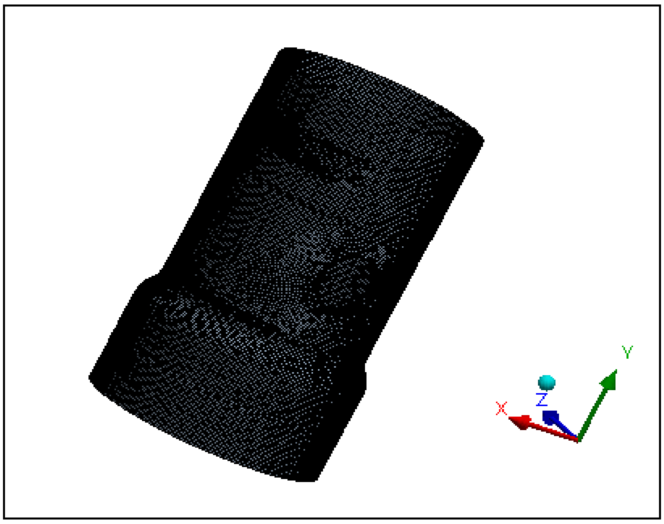


Fig 2.2: FE model of old Geometry.

Tori cone model is built using Hex dominated mesh without compromising the quality of the mesh with 116557 no of elements and 515665 number of nodes.

2.4 Loads and BC's

Fig 2.3, shows the sectional view of the intermediate tori cone and the flow path. This view is helpful for understanding the flow path of the fluid which travels from one end of the pipe to other end through different holes thus varying the fluid pressure. In the intermediate chamber one more opening is provided for releasing the extra line pressure. If the fluid pressure rises beyond the required pressure level then the high pressure fluid is released from the other opening so that the fluid pressure will reduce which interns' helps in smoother operation.

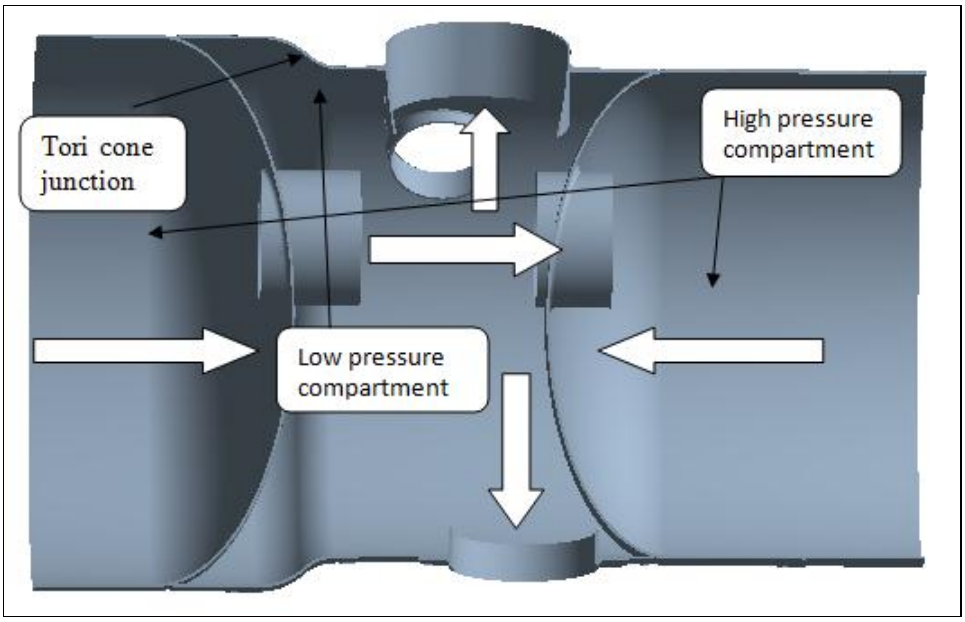
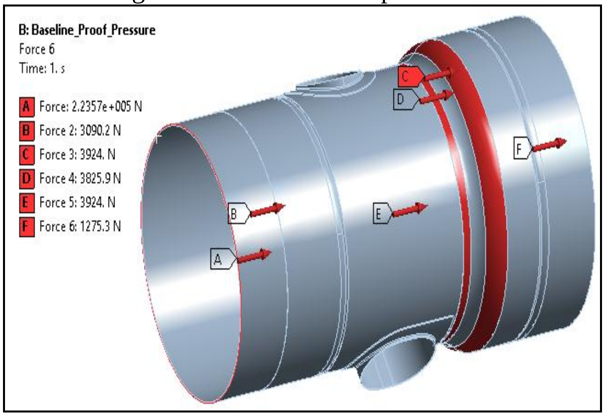


Fig 2.3: Sectional view showing the Flow Path of the Fluid.

Tori cone maintained at uniform pressure of 0.2MPa and intermediate tori cone chamber is maintained at a pressure of 0.1MPa. It is having five openings as per the line application connected, two vertical openings are meant to fluid to enter into this and the two horizontal for the line connected and the last one for releasing the chamber pressure if it exceeds beyond the operating pressure of 0.1MPa. This component is used in the reduction of pressure ranging from 0.2MPa to 0.1MPa.

Forces acting at the Baseline Proof pressure condition is as shown in the fig 2.4.



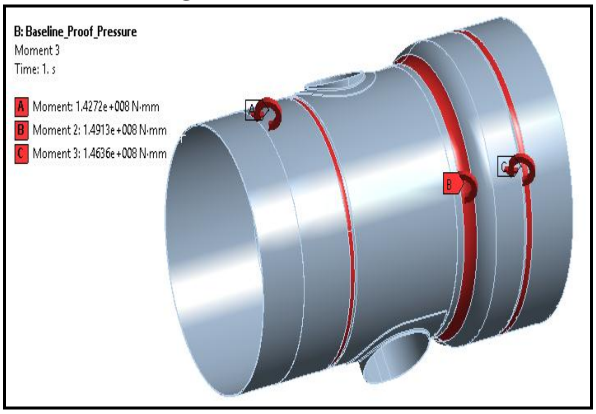


Fig 2.4: Forces and Moment Acting on the Model.

The seismic moments acting at the Baseline Proof pressure condition is as shown in the fig 2.4, three seismic moments will be acting at the three junction joints of the chamber.

The pressure acting on the tori cone is 0.38MPa at the top and bottom chambers is 0.47MPa at the Baseline Proof pressure condition is as shown in the fig 2.5.

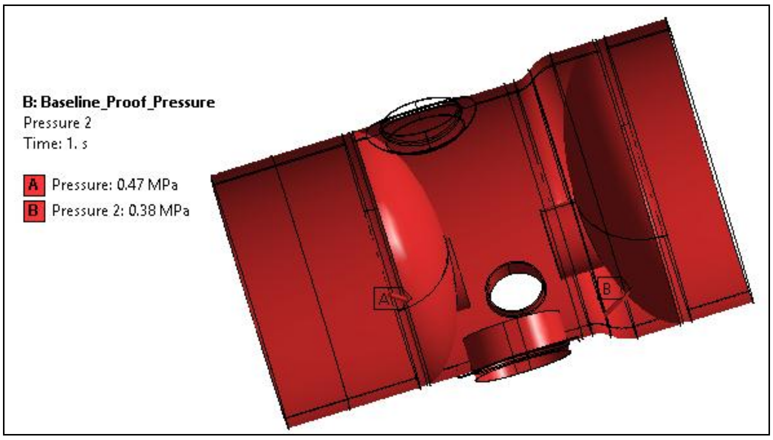


Fig 2.5: Pressure Loads at both the Sections.

3. RESULTS AND DISCUSSION

FE analysis is conducted for initial design for the given loads and then the same procedure is adopted for the new design till the required factor of safety (FOS) is achieved.

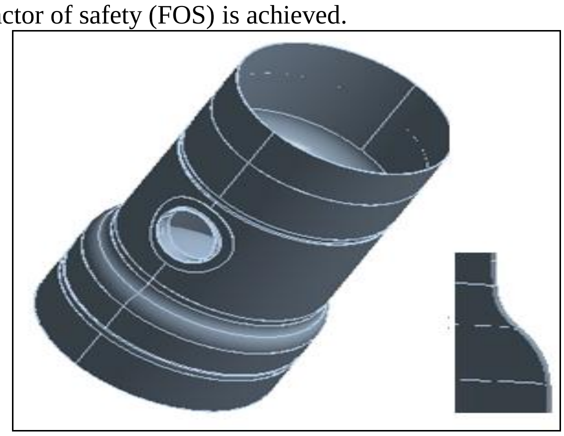


Fig 3.1: Geometry of Old Design- Circular Fillet.

3.1 Structural Analysis (Old Design)

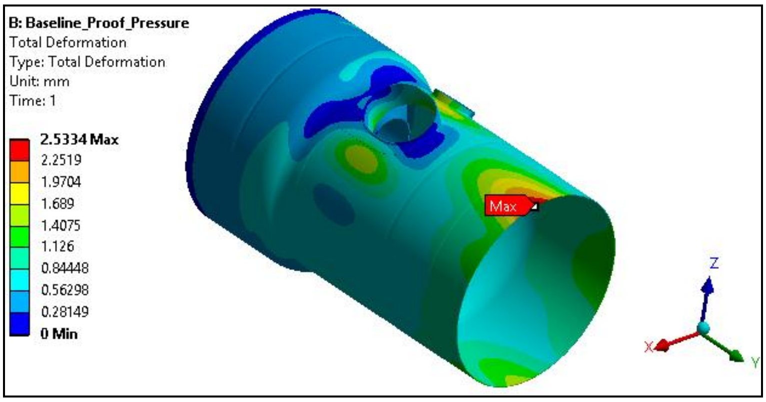


Fig 3.2: Deformation of the model-Old Design.

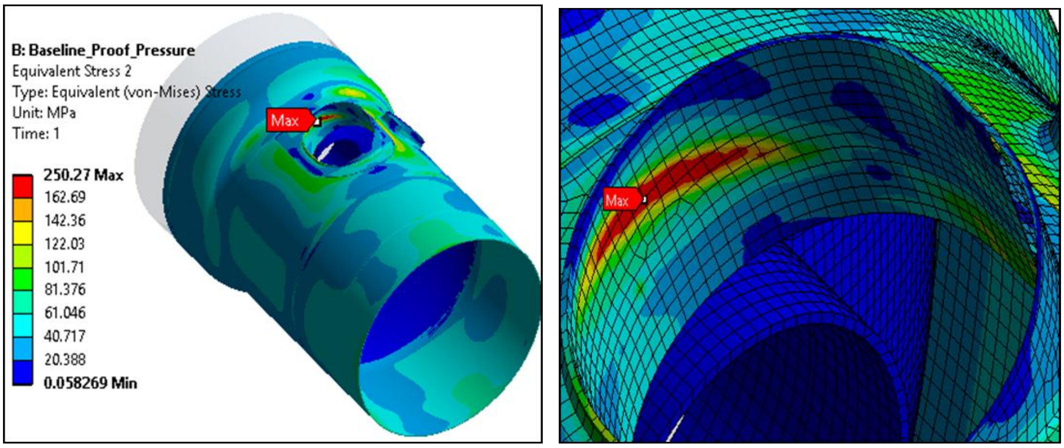


Fig 3.3: Von-Mises Stress Induced in the Model-Old Design.

Max stress observed in the tori cone junction is 250.3MPa and it is more than the material yield strength of 241.3 MPa resulting negative design margin.

Factor of Safety is calculated by,

$$FOS = \frac{\text{Yield strength}}{\text{Stress induced}} \quad \ldots.1$$

$$= \frac{241.3}{250.3} = 0.964043$$

Margin of safety is given by,

$$MOS = FOS - 1 \quad \ldots.2$$

$$= 0.964043 - 1 = -0.03596$$

3.2 Structural Analysis (New Design)

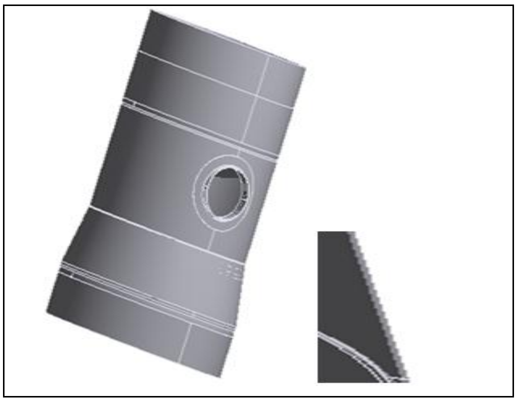


Fig 3.4: Geometry of Old Design- Chamfer.

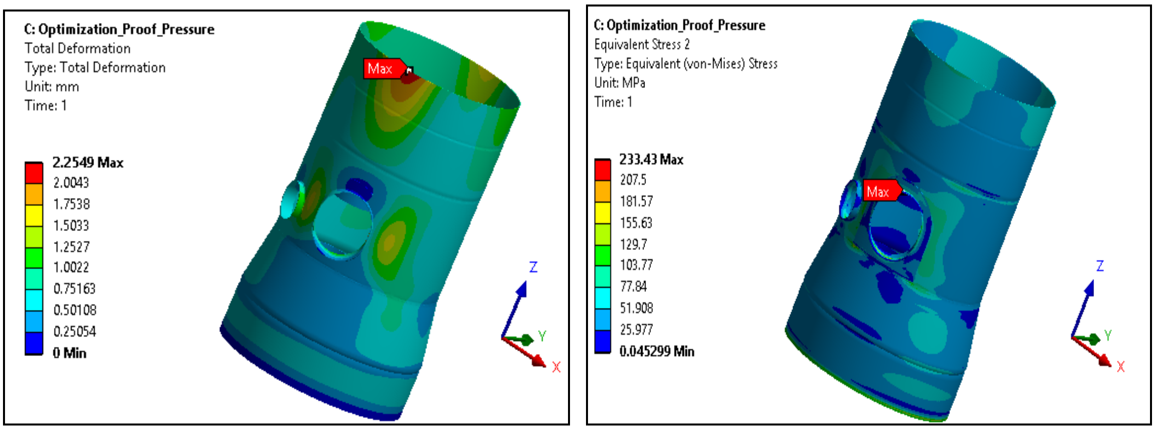


Fig 3.5: Deformation and Von-Mises Stress Induced in the Model-New Design.

3.2 Fatigue Analysis

Fatigue life of the component is checked for minimum and maximum pressure load conditions.

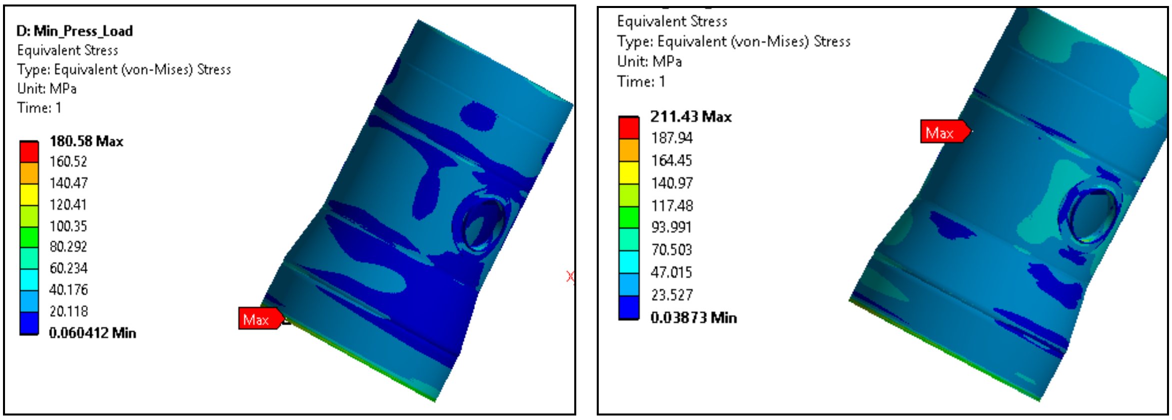


Fig 3.6: von-Mises Stress Plots for Minimum and Maximum Pressure Load Conditions.

From the above figure we can see that the maximum von-Mises stress induced in the minimum and maximum pressure load condition is 180.58 and 211.43 MPa.

Fatigue Calculation Based on SN Curve:

So σ min =180.58 MPa

So σmax =211.43 MPa

We can calculate the alternating and mean stress induced in the component by using the available empirical formula,

Alternating stress σalt is given by;

$$\sigma alt = \frac{(\sigma_{max} - \sigma_{min})}{2} \qquad \ldots.3$$

$$= \frac{(211.43-180.58)}{2} = \frac{30.85}{2}$$

σalt =15.425 MPa

$$\sigma mean = \frac{(\sigma_{max} + \sigma_{min})}{2} \qquad \ldots.4$$

$$= \frac{(211.43+180.58)}{2} = \frac{392.01}{2}$$

= 196.005 MPa

In order to determine the alternating stress at mean stress=1, Goodman relation will be used to calculate the corrected alternating stress:

Corrected Alternating Stress based on Goodman Relation:

$$\sigma corr\ alt = \frac{\sigma_a}{\left(1-\frac{\sigma_{mean}}{\sigma_{ult}}\right)} \quad \text{....5}$$

$$= \frac{15.425}{\left(1-\frac{196.005}{537.79}\right)} = \frac{15.425}{(1-0.3644)}$$

$$= \frac{15.425}{0.6356} = 24.268 \text{ MPa}$$

Margin of Safety (MOS) is given by the fallowing equation,

$$\text{MOS} = \left(\left(\frac{\sigma_e}{\sigma_{corr\ alt}}\right) - 1\right) \quad \text{....6}$$

$$= \left(\left(\frac{179.26}{24.268}\right) - 1\right)$$

$$= 7.3866 - 1 \ = 6.3866$$

Obtained values are plotted on graph which is as shown in the below graph. From the below graph we can notice that, with new design we can achieve infinite life for the given loading conditions.

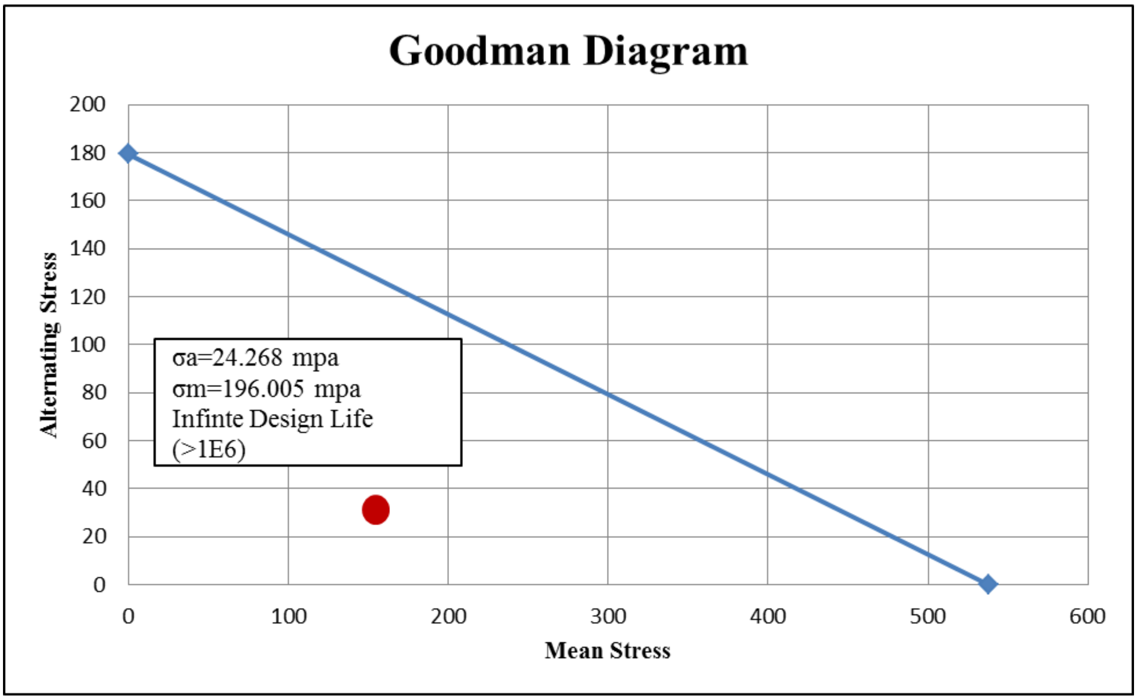


Fig 3.7: Goodman Diagram.

4. Conclusion

Analysis of a tori cone component of pressure vessel made up of Inconel 600 material for air supply application is being carried out at temperature of 80ºC and pressure of 0.1MPa. The torricone is being connected with lines under pressure and other loading conditions.

Because of internal pressure of 0.1Mpa inside the torricone and bottom and top end with 0.2MPa with body weight and the seismic moments, the stress distribution is analyzed using the ANSYS Workbench 14.5 at the junction (250.3MPa) exceeds the yield strength (241.3MPa)of the material. This is very much severe problem as it causes the failure of the component at any moment. So as the material of construction of the component is expensive we cannot take any other chances. The shape optimization of the torricone is carried out, so as to bring the exceeded stress within limit and to provide the safety of the component. After recovering the stress the two cases is being carried out based on the operating conditions given as for minimum and maximum pressure loading conditions and the safety of the component is checked and concluded.

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