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# Design & Finite Element Analysis of Pressure Vessel

Kunal Khadke[1], Dr.Dinesh Chawde[2]

*[1]2nd Year M.Tech, CAD/CAM & Robotics, [2]Professor, Department of Mechanical Engineering, KJ Somaiya College of Engineering, Mumbai, India*

*Abstract: This document - Pressure vessels are of immense importance in most of the industries today & are drastically used in many fields such as chemical, petroleum, military industries as well as in nuclear power plants. Catastrophic accidents can occur due to rupture of pressure & as a result they should be designed & analysed with immense care & precision. The exact estimation of stresses due to the applied mechanical & thermal loads are the common problems faced by any engineer while designing the vessel. This paper aims to design of pressure vessel using ASME Code Book, accordingly model the vessel in Solidworks & carrying out the finite element analysis for the Pressure vessel using ANSYS. The critical parameters of pressure vessel that are taken into account includes internal pressure, seismic, wind loads & operational loadings. The paper focuses upon the design of thin walled pressure vessel particularly a tall vessel where the vessel is designed using manual calculations & performed FEA to determine the stress in the vessel due to multiple loadings. It consists of design of thin walled pressure vessel made up of homogeneous material where the stresses are plotted along with the contour plots & have verified the FEA results with the analytical solution.*

*Keywords:  pressure vessel, design, ASME, finite element analysis, ANSYS*

## I.    INTRODUCTION

The pressure vessel is a vessel designed to contain a gas or liquid at a pressure significantly different from its ambient pressure. Ultra high pressure units are expected to be widely applied in nuclear industry, power generation sector, offshore facilities & chemical industries. Stress analysis & evaluation of stress in pressure vessels is important for solving basic design problems. Each pressure vessel must operate at the design temperature and pressure, which are the safety limits of the pressure vessel. This is because accidental leakage or leakage of contents may pose a hazard to the environment. Some of the well-known standards are ASME BPVC (American Society of Mechanical Engineers Boiler and Pressure Vessel Code), Title VIII, and API (American Petroleum Industry) Pressure Vessel Inspection Code 510. The paper focuses upon the design of thin walled pressure vessel particularly a tall vessel where the vessel is designed using manual calculations & performed FEA to determine the stress in the vessel due to multiple loadings. The main aim of this report is to investigate and determine the stress analysis through the wall of cylindrical pressure vessels – thin wall pressure vessels subjected to different loads

## II.    PROBLEM DEFINITION

Improper design and construction, and irregular testing and inspection endanger the safety of pressure vessels. When fluids are stored under pressure, it is more likely to rupture and leak. Defects such as corrosion, loss of thickness, mechanical and metallurgical defects, cracks, mechanical deformation etc. will adversely affect the operation of this device. Pressure vessels that do not comply with standard codes can be very dangerous. In fact, there have been many fatal accidents in the entire history of their operation and development. As a result, the goal of the paper is to analyse the stress concentration generated in the pressure vessel and predict the maximum load that the vessel can take considering all of the design parameters & environmental factors thereby avoid catastrophic failures in the future

## III.    OBJECTIVES

The objectives of the proposed model are summarized below are to perform Stress analysis of thin wall pressure vessel using ANSYS which will include modelling & FEA analysis the pressure vessel along with the appropriate supports in Solidworks & ANSYS respectively using design calculation for a particular design pressure Secondly it also aims to analyse the contour plot the Maximum Principal Stress, maximum tensile stress & Von Mises stress in the pressure vessel upon loading.

It also gives an idea to understand the critical areas of failure & accordingly provide reinforcement to the areas. When a pressure vessel has complex geometry or loading conditions such that traditional methods (i.e. hand calculations) are inadequate in accurately evaluating the vessel, finite element analysis can be used to ensure the design is acceptable.

## IV. LITERATURE REVIEW

Senthil Anbazhagan in this paper shows an analytical approach based wind and seismic design recommendations for vertical tall process column & focuses on how the wind and seismic analysis is a fundamental requirement for equipment design in the oil and gas industry. In this article, column heights are designed to withstand external seismic and wind action and some important design steps to consider to avoid column breakage. M. A. Khattak depicts in his paper the common root causes of pressure vessel failures & conclude a brief review of various pressure vessel failure mechanisms is presented, taking into account the toughness of heat-aged samples, the effects of heat aging on the adsorbed hydrogen and plastic regions of the crack tip of welded steel, and the fatigue cracking mechanisms of welds. Effect of fatigue crack growth in austenitic steel and thermally aged samples in welded steel area also observed. G.S. Vivek paved the way for design and analysis of vertical pressure vessel where a case study was conducted by performing a linear static analysis on the vertical pressure vessel for stress analysis performed in accordance with the ASME code, and through this analysis, the FEA analysis result showed that the equivalent stress concentration was under various pressure conditions under the maximum allowable stress SA 516 Gr 70 in pressure. Apsara C. Gedam depicts stress analysis of pressure vessel with different end connections on the journal published by her where the goal of paper is to compare stress distribution in the pressure vessel for different end connections viz. hemispherical, flat circular, standard ellipsoidal and dished shaped pressure vessel heads. Norliza Rahman enlightens an interactive short cut method for pressure vessel design based on ASME where he proposed interactive decompression vessel design method based on ASME standards. In this article, it has proposed an interactive, abbreviated method for designing pressure vessels to determine the best and most reliable application of the system. The system has been proven to pass testing and validation using two different approaches. Testing and learning from last year's project shows a comparison between manual ASME code calculations and iPVD calculations

## V. METHODOLOGY

When a pressure vessel has a complex shape or loading conditions that make it unsuitable for accurate evaluation of the vessel by conventional methods & manual calculation, finite element analysis can be used to determine if the design is acceptable. There are several types of load conditions that can be analysed using FEA. These include internal pressure, external pressure, self-weight, thermal loads, cyclic loads, shocks and shocks, seismic loads, wind loads, vibration loads, and loads from external nozzles. This report enlightens about how boundary conditions can be implemented for particularly a tall vessel which is subjected to structural loadings & how the stress is generated in different parts of the vessel

## VI. PROPOSED WORK

A tall vessel has been designed using Solid works where the tower is divided into six sections. An ellipsoidal head at the top followed by cylindrical shells & flanges mounted on each shells. At the lower section there is a conical shell & lastly there is one more ellipsoidal head to bottom of the vessel & all of these shells are supported by skirt support.
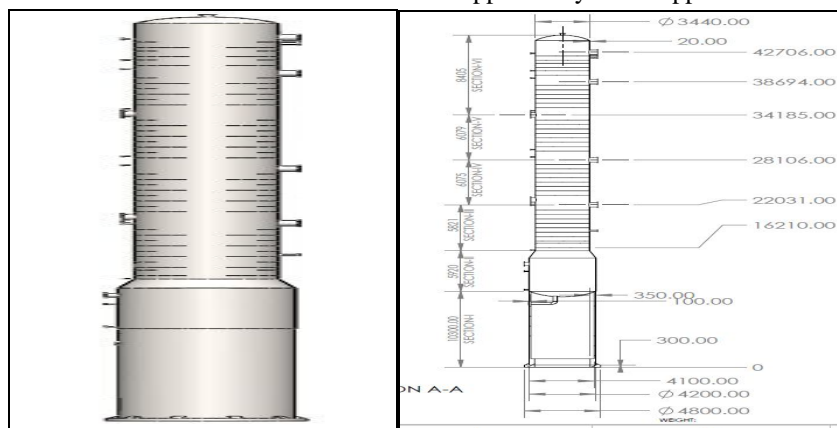


Figure 1. Geometry of tall vessel

Table I. Parameters of the design

| ASME Sec. Viii, Div. 1 Latest | | |
|---|---|---|
| P | Design Pressure | 1.5 MPa |
| T | Design Temperature | $250^0$ C |
| E | Weld Joint Efficiency | 0.85 |
| Ca | Corrosion Allowance | 1.5 mm |
| Material Of Construction | | |
| Shell | Sa 515 Gr. 70 (Tensile strength) = 400MPa | |
| Skirt | Sa 515 Gr. 70 (Tensile strength) = 400MPa | |
| Construction Details | | |
| D | Shell Id | 3.4 m |
| H | Overall Column Height | 42.6 m |
| H | Skirt Height | 10.2 m |
| Type of dished end : 2 : 1, Ellipsoidal<br>Insulation: Yes (HOT) 100 mm.<br>Internals :40 Trays (Valve Type) | | |

*A. Elliptical Head Calculation*

Internal Pressure

Required Thickness due to Internal Pressure [tr]:

= (P*Do*Kcor)/(2*S*E+2*P*(Kcor-0.1)) per ASME Code Section VIII Division 1

= (1.5*4.0*0.998)/(2*137.9*1.0+2*1.5*(0.998-0.1))

= 21.4985 + 3.1750 = 24.6735 mm

External Pressure

Calculation for Maximum Allowable External Pressure (MAEP):

| Tca | OD | D/t | Factor A | B |
|---|---|---|---|---|
| 21.825 | 4.00 | 183.28 | 0.0007578 | 75.76 |

EMAP = B/(K0*D/t) = 75.764/( 0.9 *183.2765 ) = 0.4593 N./sq.mm.

P = 1.67 * External Design pressure for this head.

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

= ((2*S*E*t)/(Kcor*D+0.2*t))/1.67 per Appendix 1-4 (c)

= ((2*137.9*1.0*21.8249)/(0.998*3.9563+0.2*21.8249))/1.67

= 0.912 N./sq.mm.

Maximum Allowable External Pressure [MAEP]

= 0.912 N./sq.mm

*B. Cylindrical Shell*

Required Thickness due to Internal Pressure [tr]:

= (P*Do*Kcor)/(2*S*E+2*P*(Kcor-0.1)) per

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= (1.5*3.4*0.997)/(2*137.9*1.0+2*1.5*(0.997-0.1))

 = 18.2669 + 3.1750 = 21.4419 mm.

Results for Maximum Allowable External Pressure (MAEP):

| Tca | OD | SLEN | D/t | L/D | Factor A | B |
|-----|-----|------|--------|--------|----------|-------|
| 21.825 | 4.00 | 4.73 | 183.28 | 1.1823 | 0.00045 | 45.72 |

EMAP = (4*B)/(3*(D/t)) = (4*45.718)/(3*183.2765) = 0.3326 N./sq.mm

Maximum Allowable External Pressure [MAEP] = 0.3326 N. /sq.mm

*C. Wind Load Analysis*

| | |
|---|---|
| Factor of Importance | 1.000 |
| Static & Dynamic Load factor (Gh, Gbar) | 1.218 |
| Shape factor for the vessel | 0.632 |
| Average speed of wind | 31.3 m/sec |
| Category of exposure | C |
| Alpha value from table | 7.0000 |
| Zg value from table | 900.0000 |
| Do value from table | 0.0050 |


*D. Calculation for the First Element*

Factor of roughness = 1.0

Values [cf1] and [cf2] as H / D < 25.0

CF = CF1 + (CF2-CF1) * (H/D - 7.0)/(25.0 - 7.0 ) = 0.6 + (0.7- 0.6) * (12.828 - 7.0)/( 25.0 - 7.0 ) = 0.632

Value of Alpha, Zg is taken from reference tale [Alpha, Zg]

For category of exposure C:

Alpha = 7.0, Zg = 274.32 m.

Height of considered section [z]

= Centroid Height + Base Height = = 4.75 + 0.0 = 4.75

z = Max (4.572, 4.75) = 4.75 m.

Coefficient of pressure velocity [kZ]:

= 2.58(z/zg )^(2/Alpha) : z is Elevation of First Element = 2.58(4.75/900) ^ (2/7.0)

= 0.81

Ratio of height to diameter:

= Maximum Height (length) ^2 / Sum of Area of the Elements = 149.285(^2)/1737.224 = 12.828

Frequency of Vibration = 1.196 Hz

Outer Diameter of the vessel =3 m.

Damping factor of Vibration (Operating) Beta = 0.01

For category of terrace C

S = 1.0, Gamma = 0.23, Drag Coeff. = 0.005, Alpha = 7.0

Compute [fbar]

= 10.5 * Frequency (Hz) * Vessel Height (ft) / (S * Vr(mph)) = 10.5 * 1.196(Hz) * 149.285(ft)/S * 1.0(mph)

= 26.777

Calculation for wind pressure [qz]

Factor of importance: I = 1.0

Wind Speed = 31.292 m/sec Converts to 70.0 mph

qz = 0.00256 * kZ * (I * Vr)² = 0.00256 * 0.81*(1.0 * 70.0)² = 10.157 psf

Converts to: 486.292 N./m²

Force on the First Element [Fz]

= qz * Gh * CF * Wind Area  = 486.292 * 1.218 * 0.632 * 45.589   = 17085.02 N.

Using the above calculations for any element, load for wind loads at various heights are calculated & stated in the figure 2 below

| Wind Height m. | Wind Diameter m. | Wind Area m² | Wind Pressure N./m² | Element Wind Load N. |
|---|---|---|---|---|
| 4.75 | 4.8 | 45.5887 | 486.292 | 17085 |
| 9.7 | 4.8 | 1.91952 | 596.335 | 882.156 |
| 11.9 | 4.8 | 19.1952 | 632.201 | 9352.12 |
| 14.8337 | 4.43924 | 8.52123 | 673.284 | 4421.43 |
| 17.82 | 4.08 | 16.316 | 709.509 | 8921.38 |
| 20.7305 | 4.08 | 7.42784 | 740.85 | 4240.86 |
| 23.141 | 4.08 | 12.237 | 764.503 | 7209.66 |
| 26.141 | 4.08 | 12.237 | 791.599 | 7465.18 |
| 29.141 | 4.08 | 12.237 | 816.555 | 7700.54 |
| 32.141 | 4.08 | 12.237 | 839.739 | 7919.17 |
| 35.141 | 4.08 | 12.237 | 861.424 | 8123.67 |
| 38.141 | 4.08 | 12.237 | 881.824 | 8316.06 |
| 41.641 | 4.08 | 16.316 | 904.224 | 11369.7 |
| 44.4901 | 4.08 | 6.83732 | 921.485 | 4855.52 |

Figure 2. Wind loads at different sections of tower

*E.  Earthquake Analysis Results*

The UBC Zone Factor for the Vessel is  0.3000

The Importance Factor as Specified by the User is  1.000

The UBC Frequency and Soil Factor (C) is  1.408

The UBC Force Factor as Specified  3.000

The UBC Total Weight (W) for the Vessel is  1029194.5 N

The UBC Total Shear (V) for the Vessel is  144936.9 N.

The UBC Top Shear (Ft) for the Vessel is  8484.4 N

| Distance to Support m. | Cumulative Wind Shear N. | Earthquake Shear N. | Wind Bending N.mm. | Earthquake Bending N.mm. |
|---|---|---|---|---|
| 4.75 | 107862 | 144937 | 2.648E+09 | 4.546E+09 |
| 9.7 | 90777.5 | 142099 | 1703791616 | 3.182E+09 |
| 11.9 | 89895.3 | 139710 | 1667642496 | 3.126E+09 |
| 14.86 | 80543.2 | 133587 | 1326627328 | 2.579E+09 |
| 17.82 | 76121.8 | 130016 | 1176168064 | 2.326E+09 |
| 20.7305 | 67200.4 | 120380 | 889407744 | 1824758272 |
| 23.141 | 62959.5 | 115277 | 770849088 | 1610105856 |
| 26.141 | 55749.9 | 105891 | 592712704 | 1278219776 |
| 29.141 | 48284.7 | 95289 | 436597632 | 976327168 |
| 32.141 | 40584.1 | 83470.2 | 303240320 | 708079552 |
| 35.141 | 32665 | 70434.6 | 193322096 | 477128640 |
| 38.141 | 24541.3 | 56182.4 | 107477864 | 287126016 |
| 41.641 | 16225.2 | 40713.4 | 46303224 | 141723408 |
| 44.141 | 4855.52 | 18195.3 | 4124633 | 23858292 |

Figure 3. Bending moments due to seismic & wind loads

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*F.  FEA: Boundary conditions*

The boundary conditions are applied in case of the maximum load conditions where the vessel is subjected to high seismic & wind load. Along with these external factors, the vessel is also subjected to load due to its own self weight & the hydrostatic pressure of the liquid present inside the vessel situated at height from section VI to section II. Internal pressure of 1.5MPa is applied on to the inner walls of the vessel. Accordingly, the bending moments due to wind & seismic loads are applied at the particular heights as shown in table
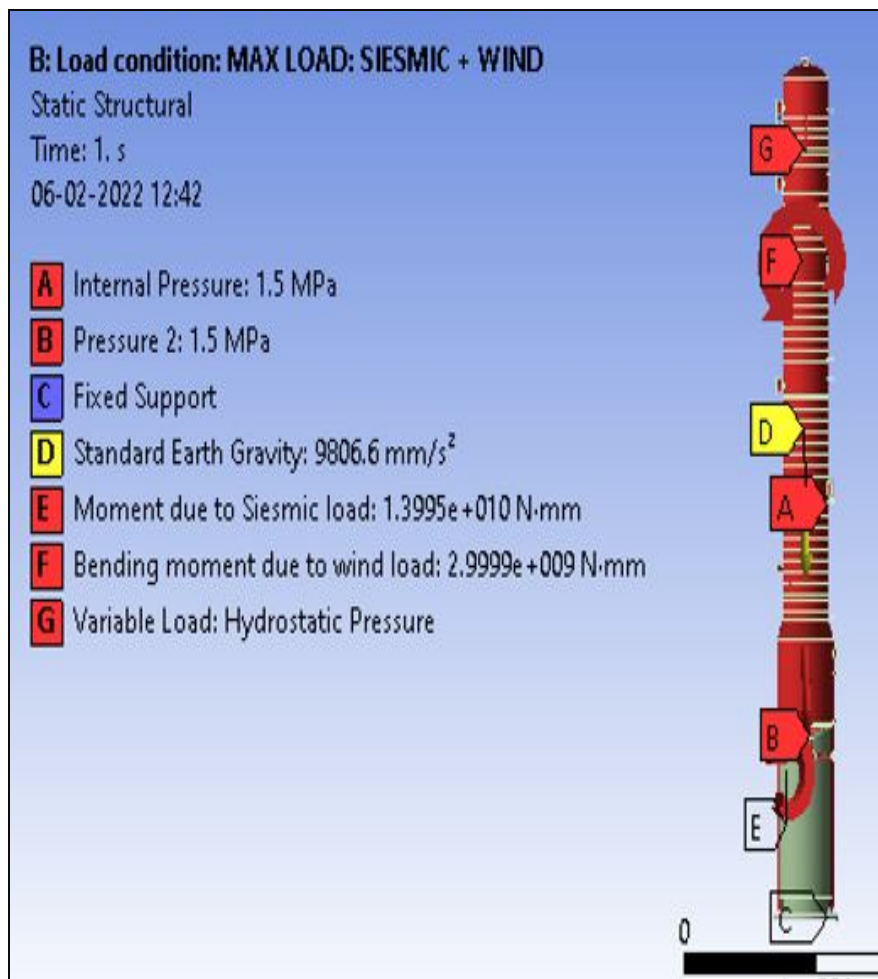


Figure 4. Defined boundary conditions

| Parameter | Magnitude | Location |
|---|---|---|
| Internal Pressure ($P_i$) | 1.5 MPa | Inner walls of the cylinder Normal direction to the surface |
| Fixed support | - | Base Plate |
| Standard Earth Gravity – Dead Weight of the vessel | 309390 kg | At the centre of gravity |
| Variable Load: Hydrostatic Pressure | -Filled with water | Fluid is filled from section VI to section II |
| Bending Moment due to wind Load | 30,00,000 N-mm | At the Centre of gravity of the vessel |
| Bending Moment due to Seismic Load | 1,34,00,000 N-mm | At $\frac{2*H}{3}$ |

Table II. Location & Magnitude of boundary conditions

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### G. FEA Results

After pre-processing, meshing & setup of the model in ANSYS, the setup is solved w=using ANSYS Structural Workbench solver. After observing the contour plot for the factor of safety the minimum factor of safety (0.9) is observed at the junction of conical shell & cylindrical shell due to bending loads because of the seismic & wind loads. Fig. 5 clearly shows the factor of safety at the nozzle openings where there is removal of material in the shells for welding the nozzle. A significant factor of safety (more than 1.5) is observed at each of the nozzle openings. So it can be concluded that the thickness of nozzle at the nozzle opening is adequate enough to sustain the loads ate maximum loading condition
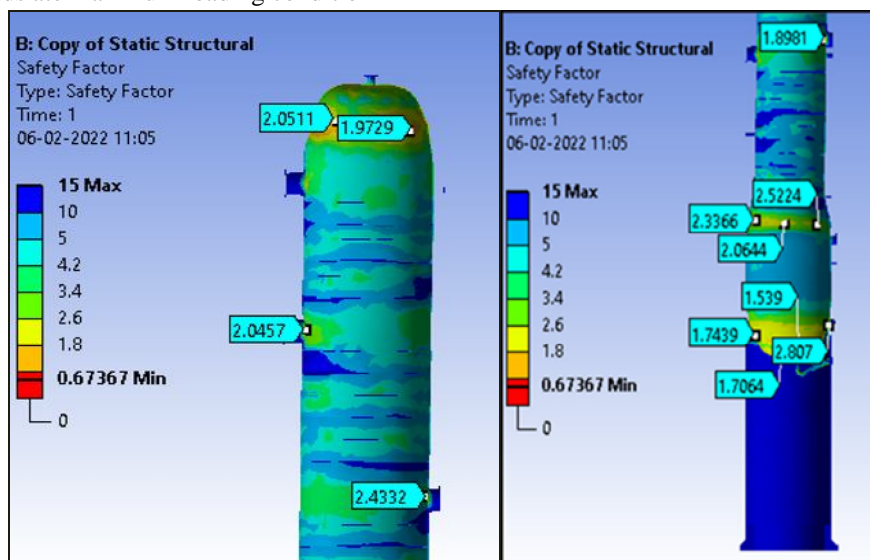


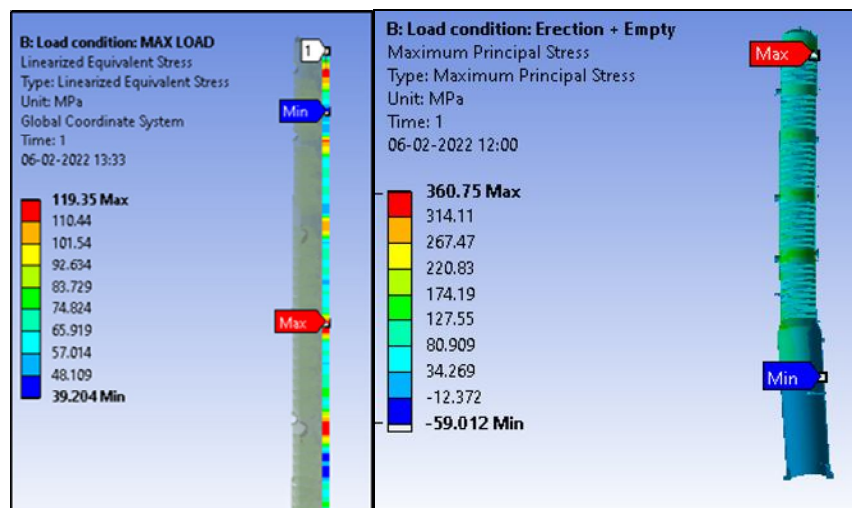Figure 5. Contour plots for factor of safety



Figure 6. Contour plots foe equivalent stress & maximum principal stress for the tower

In the above case of boundary conditions, the maximum stress will be induced in the third case. So the results in case of the third case are plotted in the figure 6 mentioned below. The equivalent stress & maximum principal stress contours are shown in figure. It is clearly evident that the equivalent stress (342.85 MPa) & the maximum principal stress (360.75 MPa) are well below the maximum tensile strength of the material. So it can be concluded that the design is safe. The total deformation in the vessel is observed at the top of the vessel (82 mm) which is majorly due to the wind loads acting on the vessel. In case of strong winds, the vessel will experience movements at the top of the vessel

FEA results along the stress concentration lines

SCL plot 1: Stress concentration line along the walls of the cylindrical shell at maximum loading condition is shown in figure 6
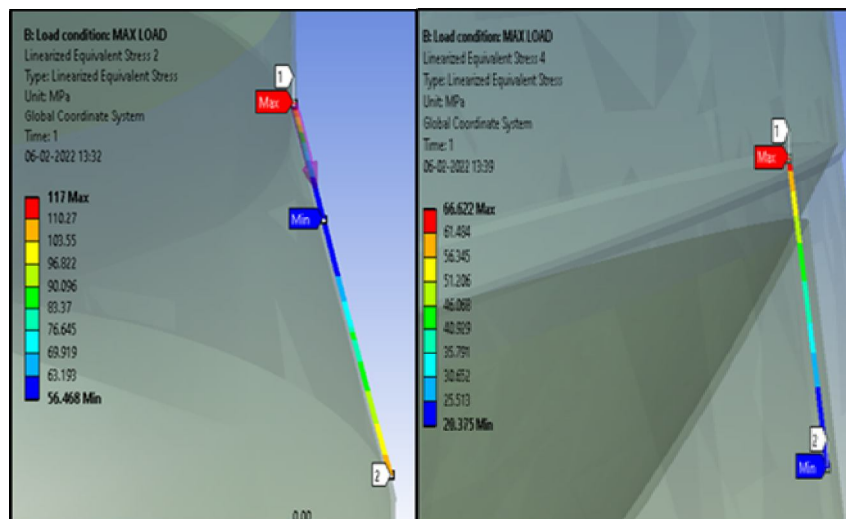
Figure 7. Linearized stresses along the SCL - stress concentration lines

SCL Plot 2: Stress concentration line along the junction between the nozzle & the shell at maximum loading condition is shown in figure 7

SCL Plot 3: SCL at junction of pressure vessel shell & skirt support at maximum load condition is shown in figure 7

## VII.    CONCLUSION

FEA Analysis has been carried out on the CAD models – Vertical pressure vessel & tall vessel under operating, erection and maximum loading condition & it is clearly visible from the stress plots & contours that the maximum stress induced in the pressure vessel walls in these conditions are well below the permissible allowable stress in the material chosen for manufacturing of the vessel. A tall pressure vessel is designed using ASME Code Section VIII Division – 1 & accordingly FEA has been performed on it using ANSYS to validate the design. The designed pressure vessel is safe for all of the mechanical loads in operating, lifting, erection & operating conditions. The FEA results obtained in the thick cylinder problem matches with the analytical solution & as a result we can conclude that the FEA is successfully executed. Stresses at stress concentration lines are also plotted in the report & areas of high stress concentration are clearly seen in the contours mentioned on the report

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