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RESEARCH ARTICLE

Theoretical and Finite Element Analysis of Pressure Vessel

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Abstract

Objectives: This study tests the vessel strength and performance of pressure vessel under Internal pressure, Nozzle loads, and Hydro-test using Ansys APDL, validating design alignment with ASME Section VIII following the Design by rule (Analytical) and Design by Analysis (FEA) accurate elastic analysis approach. **Methods:** This study employs ASME methods to validate vessel integrity under various loads. Strength is confirmed through analytical formulas and Finite Element Analysis (FEA) using ANSYS APDL, aligned with widely used ASME BPVC codes in the oil and gas industry. The FE model, utilizing hex elements, ensures result accuracy with a minimum of three elements across thickness. Boundary conditions are validated by comparing hoop stress in FEA with analytically calculated values. ASME's computationally efficient elastic analysis, employing a linear approach, includes stress linearization at discontinuity and non-discontinuity locations, verifying vessel design through analysis. **Findings:** Initial thicknesses for the shell and cone exceeded analytically calculated minimums, affirming vessel structural integrity through ASME's design by rule approach. Finite Element Analysis (FEA) stress analysis at critical points, such as nozzle junctions and other discontinuity areas, validates accuracy through hoop stress checks. Analysis of design and test load cases reveals stress categories well within ASME Sec VIII limits, confirming the vessel's safety and compliance with elastic stress analysis standards. **Novelty:** This method emerges as a reliable tool for vessel design, ensuring safety and ASME compliance, particularly beneficial for industries like oil and gas. It provides precise guidelines utilizing hex mesh, validates boundary conditions through hoop stress comparison, and comprehensively assesses stress in critical and non-critical zones through elastic stress analysis. Addressing common challenges identified in the literature review, this approach enhances the accuracy and reliability of pressure vessel designs in compliance with ASME standards for design and test loadings.

Keywords: Pressure Vessels; Process Industries; Stress; Loads; Pressure; Thermal; Design Validation; ASME; FE analysis

1 Introduction

Design and analysis of pressure vessels by following guidelines from ASME accurately is very important. Already published work follows the ASME guidelines accurately for design by rule but fails to follow ASME guidelines accurately for design by analysis approach. For the Design by analysis approach, proper meshing, sanity check validation, and stress categorization by using stress classification lines are very important steps to ensure the accuracy and reliability of stress assessment. Design by rule is a more conservative method than design by analysis, hence the design by rules creates bulkier pressure vessel design. So, it becomes very crucial to follow the design by analysis guidelines accurately to avoid the bulkier pressure vessel design and ultimately reduce the cost.

1.1 Literature review

Weiya Jin[(1)] has done the pressure vessel analysis as per DBA approach. The obtained results demonstrate the efficacy of the integrated approach in minimizing the material requirements for the pressure vessel. Remarkably, this optimization is achieved while ensuring that the maximum stress, as specified in the ASME BPVC, is maintained within acceptable limits. The findings underscore the potential for leveraging advanced optimization methods to enhance the efficiency and material utilization in pressure vessel design, ultimately contributing to both safety and economic considerations in the engineering domain.

Kristaq[(2)] explores the design and FEA of a vertical pressure vessel following the ASME approach. While the analytical calculations align with the ASME design by rule approach, the FEA model deviates from ASME Design by Rules guidelines. The use of a tetrahedral mesh, identified for its inaccuracies, is a notable limitation. Additionally, the absence of stress linearization as per ASME, coupled with the direct comparison of total stress values with yield strength, results in a bulkier vessel design and increased costs. Crucially, the paper lacks a sanity check to validate the FEA model's boundary conditions.

K M B Karthikeyan[(3)] work involves the design and FE analysis of a vertical pressure vessel, including an axisymmetric analysis at the skirt to dish end. The study compares Design by Analysis (DBA) and Design by Rule (DBR) approaches, noting that DBR is more conservative than DBA. While this observation is significant, the paper falls short by not validating the FEA model's boundary conditions through a sanity check.

Hydrotest is a vital hydrostatic testing method crucial for ensuring pressure vessel safety as per Henry Froats[(4)]. Particularly essential for gas cylinders, boilers, and piping systems handling hazardous materials, hydrotest rigorously assesses structural integrity and leak tightness, preventing catastrophic consequences. Its versatility spans from fire extinguishers to industrial boilers, playing a fundamental role in overall maintenance and safety protocols.

The literature review illuminates prevalent challenges within existing studies, notably deviations from ASME guidelines in Finite Element Analysis (FEA) models, concerns regarding mesh quality, insufficient stress linearization, and the absence of sanity checks for validating FEA model boundary conditions. It is imperative to address these challenges to uphold the precision and dependability of pressure vessel designs in adherence to ASME standards. Furthermore, a critical observation emerges - the often-overlooked significance of hydrotesting in the evaluation process, an essential check that cannot be disregarded. Acknowledging and addressing these aspects are crucial for advancing the accuracy, reliability, and holistic assessment of pressure vessel designs. Subsequent research and enhancements in Finite Element Analysis methodologies are paramount to overcoming these identified limitations, ultimately elevating the safety and efficiency of pressure vessels in diverse industrial applications.

1.2 Scope of Work

In light of industry experiences and available literature, it has been identified that a significant number of pressure vessel failures occur at the junction of the nozzle and shell. These failures result from elevated stresses at specific points of discontinuity, including nozzle-shell junctions, shell end connections, and lug and shell connections. The root cause lies in the simultaneous application of various loads at these discontinuities. While existing literature aligns closely with ASME Section VIII guidelines for the design by rule approach, there is room for improvement in adhering to the exact guidelines for design by analysis. Recognizing that the design by analysis approach is more optimized than the design by rule as observed by K M B Karthikeyan[(3)]. Hydtotest testing is vital for gas cylinders and boilers, where even a minor leak or weakness can lead to catastrophic consequences as per Henry Froats[(4)] and this hydrotest needs to be checked using FEA before doing the actual test and avoid the accidents. This paper's objective is to conduct a precise and reliable Finite Element (FE) analysis utilizing both design by analysis and design by rule ASME approaches.

Figure 1 displays an initial sketch of a Lock hopper vessel utilized in the oil and gas sector. This vertical vessel incorporates two lifting lugs and two lug supports, functioning as a hopper at the base with an added cone. Detailed nozzle specifications,

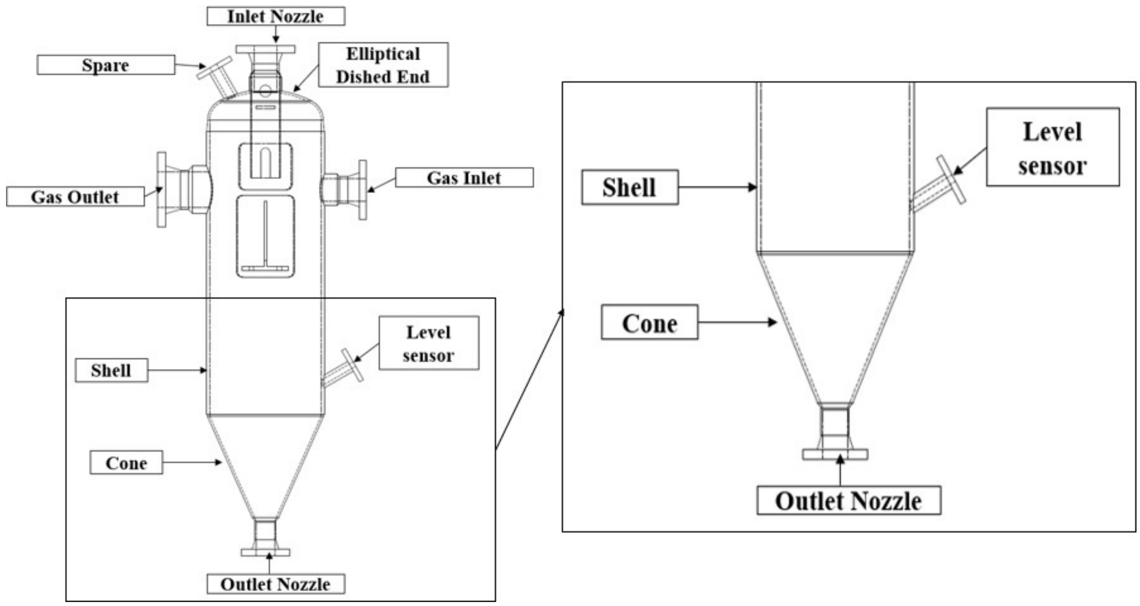


Fig 1. Lock hopper- vertical pressure vessel details

including the critical level sensor nozzle with its small size and angled installation, are outlined in Figure 1.

The validation efforts in this paper focus on the shell, cone, level sensor nozzle, and outlet nozzle, as illustrated in the zoomed-in view on the right side of Figure 1. Vessel design data like Design pressure, MAWP, Hydtrotest pressure, Design temperature and dimensions of vessel are outlined in Table 1, while comprehensive material information is presented in Table 2, referencing material properties like Young's Modulus, Poisson's ratio, Yield stress and Allowable stress from ASME Section II Part-D[5] at the design temperature.

Table 1. Preliminary Design data of lock hopper from industry

Parameter	Symbol	Value	Unit
Design pressure	P_d	1.471	MPa
MAWP	P	3.49	MPa
Hydrotest Pressure	P_t	6.636	MPa
Design temperature	T	200	0C
Inside diameter of shell	D_i	475	mm
Thickness of shell	t_s	16	mm
Joint efficiency	E	85	%
Cone thickness	t_c	16	mm
Corrosion allowance	CA	3	mm
Cone Angle	α	30	Degree

Table 2. Materials of pressure vessel parts

Component	Material	Young's Modulus (MPa)	Poisson's Ratio	Yield Stress (MPa)	Allowable Stress (MPa)
Shell, Cone	SA 516 Gr.70N	1.92E+05	0.3	225	138
Flange	SA 105	1.92E+05	0.3	213	138
Nozzle pipe	SA 106 Gr.B	1.92E+05	0.3	207	118

The analysis of the vessel encompasses both design and test conditions. Design conditions involve evaluating the Internal Maximum Allowable Working Pressure (MAWP) and Tensile nozzle loads, while test conditions include assessing the Hydrotest pressure. The ultimate goal is to enhance the understanding of the vessel's structural behavior and safety, employing an advanced approach that combines design by analysis and design by rule methodologies.

1.3 Objective

The paper analyzes and validates the design of a pressure vessel as per ASME Sec. VIII Div. 1 & 2 using CATIA V5R3 for CAD modeling, HYPERMESH 19.1 for meshing, and ANSYS 22R2 for analysis. The assessment covers both design and test conditions, aiming to enhance the understanding of the vessel's structural behavior and safety through a combined approach of design by analysis and design by rule methodologies.

2 Methodology

2.1 Analytical Calculations

2.1.1 Shell

The minimum required thickness of the shell part is determined using the method in Appendix 1 (Supplementary Design Formulas), 1-1(a), of ASME Section VIII Division 1 [(6)]:

$$t_s = \frac{PR_O}{SE + 0.4P}\, mm$$

Where, $R_O = D_i/2 + t_s$, is shell outside radius in mm, P is MAWP in MPa, S is Allowable stress limit MPa, and E is weld efficiency referred from Table 1.

$$t_s = \frac{3.49 * 253.5}{138 * 0.85 + 0.4 * 3.49}$$

$$t_s = 7.45\, mm$$

As per available preliminary design data, the corrosion allowance is 3mm as shown in the Table 1. Hence, in the above calculated thickness, a 3mm corrosion allowance is added.

$$t_s = 7.45 + 3 = 10.45\, mm$$

The calculated minimum shell thickness of 10.45mm is less than the preliminary thickness of the shell (16mm). Hence, the preliminary designed shell thickness is safe.

2.1.2 Cone

The minimum required thickness of the cone component is determined using the obligatory Appendix 1 (Supplementary design formulae), section 1-4(c), of ASME Section VIII Division 1 [(6)]:

$$t_c = \frac{PD_i}{2cos\alpha(SE - 0.6P)}\, mm$$

Where α is 30° from Table 1.

$$t_c = \frac{3.49 * 475}{2 * cos30(138 * 0.85 - 0.6 * 3.49)}$$

$$t_c = 8.31\, mm$$

As per available preliminary design data, the corrosion allowance is 3mm as shown in the Table 1. Hence, in the above calculated thickness, a 3mm corrosion allowance is added.

$$t_c = 8.31 + 3 = 11.31\, mm$$

The calculated minimum cone thickness of 11.31mm is less than the preliminary thickness of the cone (16mm). Hence, preliminary designed cone thickness is safe.

2.2 Finite Element Modelling

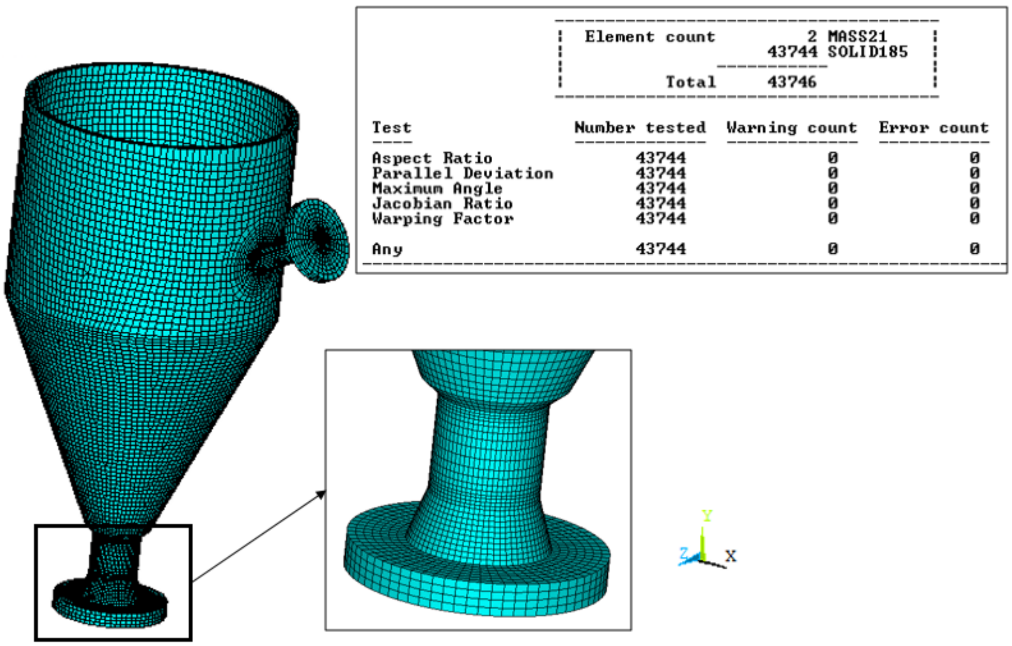
For different loading scenarios, linear elastic analysis as per DBA by clause 5.2.2 of ASME Sec. VIII Div. 2[7] is performed.

2.2.1 CAD Modelling

CAD modelling of the vessel is done in CATIA V5 R3. CATIA V5 R3 is Dassault Systems CAD Software. As vessels inner surface is in contact with fluid, the FEA of vessel is always carried out in corroded conditions. Hence, preliminary design dimensions were reduced by applying 3mm corrosion allowance. Nozzle dimensions are referred from ASME B16.5[8].

2.2.2 Meshing

As per standard practice, FEA is carried out for corroded conditions. The 3D model has been meshed with 3D hex SOLID185 elements for structural analysis. As per Grasp Engineering[9], minimum 3 elements are maintained across the thickness of each part, so the stress linearization can be done accurately and avoid errors in the results. 3D Hex elements give more accurate results than tetra elements as per Value Design ltd[10]. In Figure 2, the mesh model, created using Altair's Hypermesh 19.1 software, and meets all ANSYS APDL element check criteria without any warnings or errors (0%), ensuring optimal result accuracy.



Element count		
	2	MASS21
	43744	SOLID185
Total	43746	

Test	Number tested	Warning count	Error count
Aspect Ratio	43744	0	0
Parallel Deviation	43744	0	0
Maximum Angle	43744	0	0
Jacobian Ratio	43744	0	0
Warping Factor	43744	0	0
Any	43744	0	0

Fig 2. Mesh model of vessel Shell, Nozzle and Cone section with detailed element quality

2.3 Load Combinations

The load cases considered for the vessel in design by analysis (Linear elastic) approach as per ASME Sec. VIII Div. 2[7]. For design conditions, internal maximum allowable pressure + tensile nozzle loads are applied. For test condition, hydrotest pressure is applied.

2.3.1 Pressure loading

Internal pressure (MAWP = 3.49MPa & Hydrotest = 6.636MPa) is applied on the inner surfaces of the Shell, Cone and nozzles. Due to internal pressure, there will be a thrust acting at nozzle openings. An equivalent force is applied at the opening faces of the model in the form of compensating pressure in order to maintain the model's static equilibrium state. The formula for the same computation is provided below,

$$Pc = \frac{-Pi}{\left(\left(D_0/D_i\right)^2 - 1\right)}$$

Where, D_i = Inner Diameter Opening Face, Do = Outer Diameter Opening Face, P_i = Internal Pressure, Pc = Pressure Compensation.

Figure 3 illustrates the internal pressure applied to the inner surfaces of the shell, cone, and nozzle, including cylindrical and axial constraints. The compensating force (Thrust force) is applied at the nozzle opening, as depicted in Figure 3.

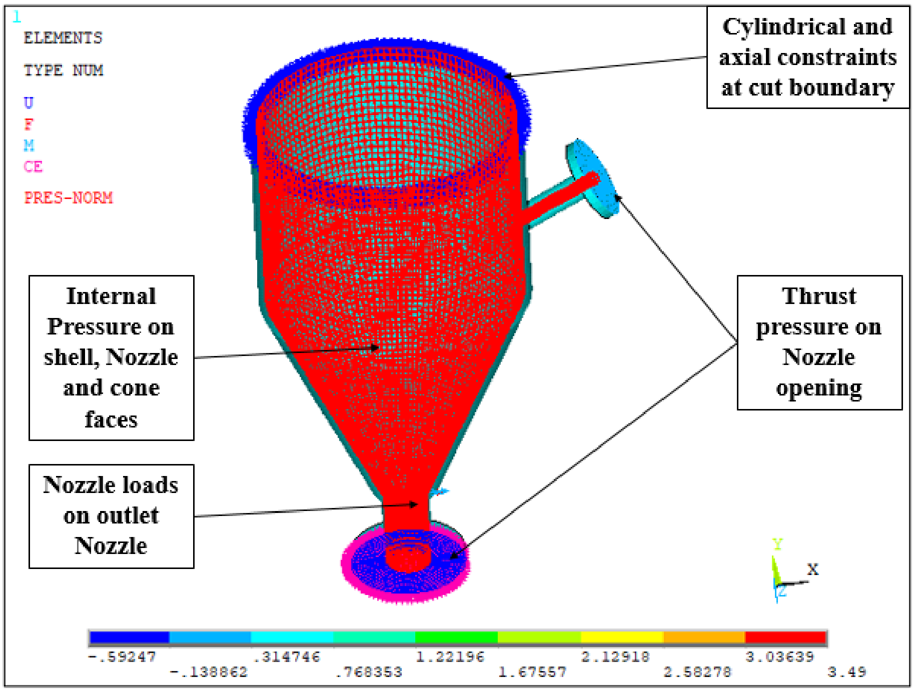


Fig 3. Boundary conditions on pressure vessel

2.3.2 Nozzle loading

Table 3 shows the nozzle forces and moments values which are referred from the API 660 [11] standard.

Table 3. Nozzle Loads details as per API 660

Nozzle	M_L (N-mm)	M_C (N-mm)	M_T (N-mm)	P (N)	V_L (N)	V_C (N)
Outlet	8.190E+05	6.300E+05	-9.450E+05	4200.0	4200.0	3150.0

Nozzle loads are applied at the nozzle to the shell/Dished End junction and transferred to Nozzle flange face with the help of RBE3 and CERIG constraints equations. The applied nozzle loads are shown in Figure 3.

2.4 Acceptance criteria

Acceptance criteria are the stress limits according to ASME Section VIII, Division 2 [7] which are used to qualify the vessel under applied loading.

a. For design condition:

P_m < S;

P_m+Pb < 1.5S;

P_L < Maximum (1.5S or Sy);

P_L+P_b < Maximum (1.5S or Sy)

b. For test condition:

P_m < 0.95Sy;

P_m+Pb < 1.425Sy

Where,

P_m=General primary membrane stress (MPa),
P_L= Local primary membrane stress (MPa),
P_b= Bending stress (MPa),
S = Allowable stress (MPa),
S_y= Yield stress (MPa).

3 Results and Discussion

A linear static analysis is solved using ANSYS APDL. A direct solver type used for solution. In linear analysis, convergence criteria are typically not required for direct solvers. Direct solvers are used to solve systems of linear equations directly, without the need for iterative methods that involve convergence criteria.

3.1 Stress Classification lines

SCLs are important to classify the stresses in different stress components as per Grasp Engineering[9]. Maintaining at least 2-3 elements across thickness gives accurate stress results for SCL. Straight lines connecting the inside and outside of a vessel are known as stress classification lines or SCLs. Both the vessel's interior and exterior surfaces are parallel to it. SCL passes through nodes where stresses in FEA are calculated, and these nodes are where stresses are calculated. As a result, the SCL tool divides the stresses into several stress components such as membrane stress, bending stress, membrane plus bending, peak, and total stresses. Cross sections of components with varying component thicknesses create membrane and bending loads. The definition of SCLs lines may be found in table 5.6 of ASME Sec. VIII Div. 2[7]. In Figure 4, SCL are shown at different locations on the vessel. SCLs shown in Figure 4 covers discontinuity (Junction) and away from discontinuity regions. SCL-1 and SCL-2 are at nozzle junctions while SCL-3 is away from the discontinuity.

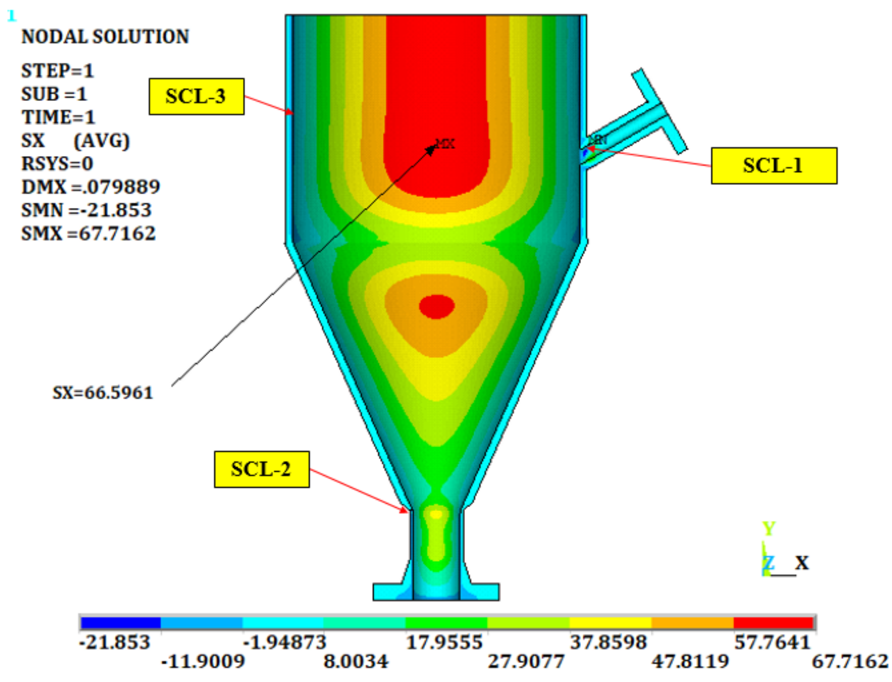


Fig 4. Hoop Stress (MPa) and Stress Classification lines (SCL) locations

3.2 Hoop stress check

A sanity check of the FEA model ensures the correct application of loading. Hence, a hoop stress is analytically calculated and compared with the value from FEA. Results are verified for stress due to MAWP as follows.

Away from discontinuities, hoop stress in the shell,

$$\sigma = (P * a^2 / (b^2 - a^2)) * (1 + b^2/a^2)$$

Where, P=3.49 MPa and a=481 mm for the ID and 507 mm for the OD of the shell

σ = 66.351 MPa

In Figure 4, the hoop stress plot reveals that the tangential stress 66.59 (SX) closely match the above estimated hoop stress value, with a minimal percentage deviation of 0.0% between the analytical and ANSYS results. This validation ensures the accuracy of the FEA model for applied loadings.

3.3 Load case 1 results

For load case 1 Maximum stress of 208.132 MPa is observed at the level sensor nozzle to shell junction as shown in Figure 5**(a)**. A total deformation of 0.18mm is observed at the outlet nozzle as shown in Figure 5**(b)**. The stress value obtained cannot be directly compared to any allowable limits. In existing literature[2], stress values are compared directly to the allowable limit without considering stress classification (SCL). A direct comparison of the stress value (208.132 MPa) with the yield (207 MPa) indicates that the allowable limit is exceeded. This implies that the vessel component may fail to withstand the loading, necessitating modifications or an increase in thickness in the design. However, it's important to note that the maximum stress value is localized. In such cases, ASME[7] recommends the use of stress classification for a more accurate assessment. Stress linearization is carried out at multiple locations to see the membrane and bending stress variation across the thickness. A total 3 SCLs are considered at discontinuity region and away from discontinuity region as per ASME[7]. Figure 6 **(a) (b), (c)** and Table 4 show the different stress categories variation across thickness at multiple locations and those stresses are within allowable limits set by ASME[7].

Table 4. Stress result summary for design and test condition

SCL	Stress category	Induced Stress Value (MPa)	Allowable Limit (MPa)	Pass/Fail
		Load case 1: Design Condition		
SCL-1	P_L	81.53	S_{PL} = 207	Pass
	P_L+P_b	149	Sps = 207	Pass
SCL-2	P_L	61.51	S_{PL} = 207	Pass
	P_L+P_b	68.68	Sps = 207	Pass
SCL-3	Pm	58.12	S =138	Pass
	P_m+P_b	60.77	1.5S = 207	Pass
		Load case 2: Test Condition		
SCL-3	Pm	110.5	0.95Sy = 213.75	Pass
	P_m+P_b	115.5	1.425Sy = 320.625	Pass

3.4 Load case 2 results

For load case 2 maximum stress observed is 394.48 MPa and this is the total stress as shown in the Figure 5**(c)**. A total deformation of 0.152mm is observed at the level sensor nozzle junction as shown in Figure 5**(d)**. As per part 4 of ASME[7], for test conditions stress classifications away from discontinuity need to be checked. Figure 6 **(d)** and Table 4 show the different stress categories variation across thickness at SCL-3 and those stresses are within allowable limits set by ASME[7].

3.5 Discussion

Within the existing literature, meshing emerges as a critical factor influencing stress results substantially. Notably, the use of tetra mesh, as indicated by Value Design Ltd[10], is considered less accurate compared to hex mesh. Grasp Engineering[9] emphasizes the significance of employing Stress Classification Lines (SCLs) and maintaining a minimum of 2-3 elements across the thickness of each part to obtain precise results. Beyond meshing considerations, the validation of applied boundary conditions is crucial, involving a thorough comparison of analytical results with Finite Element Analysis (FEA) outcomes. Hydrotest, emphasized by Henry Froats[4], is an essential hydrostatic testing method ensuring pressure vessel safety. It is conducted on each vessel prior to operation and validated through Finite Element Analysis (FEA). This paper addresses a

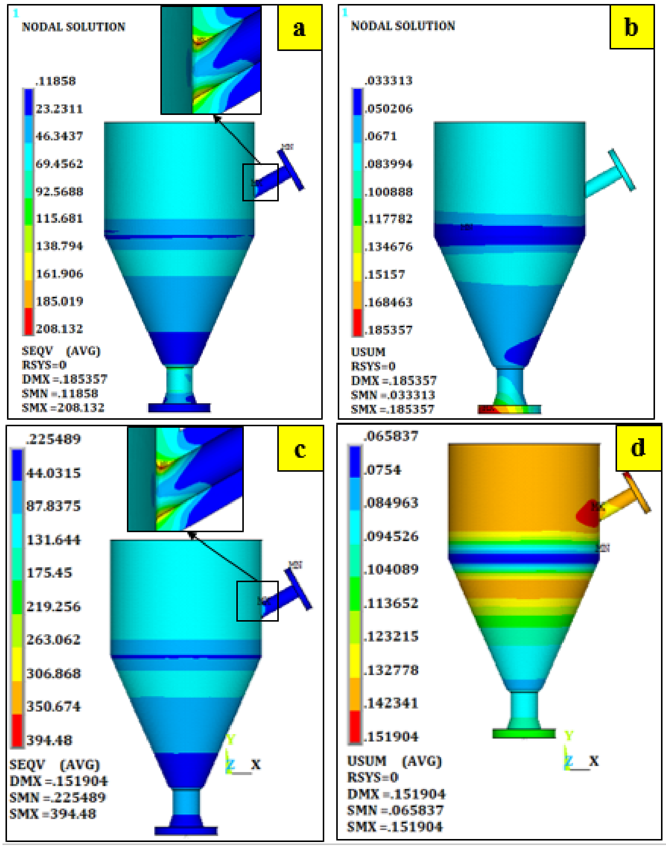


Fig 5. (a) - Equivalent stress plot (MPa) for LC-1, (b) - Total deformation plot (mm) for LC-1 & (c) - Equivalent stress plot (MPa) for LC-2, (d) - Total deformation plot (mm) for LC-2

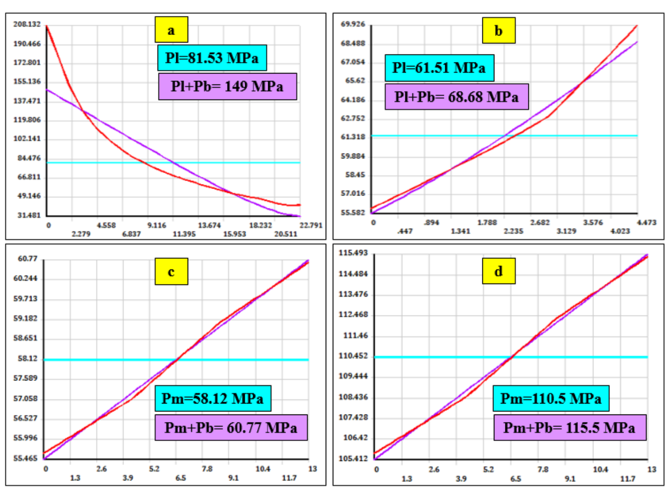


Fig 6. (a), (b), (c)- Stress linearization for LC-1 & (d) - Stress linearization for LC-2

comprehensive FEA modeling approach, encompassing boundary condition validation and accurate SCL classification for enhanced accuracy in stress assessments for design and test conditions.

Table 4 shows the stress classification results summary, and those stresses are within the allowable limits as per ASME[7] acceptance criteria.

Achieving improved accuracy in stress assessments is vital for designing vessels in a more precise and optimized way, ultimately leading to cost savings in manufacturing.

4 Conclusion

In conclusion, this study has successfully investigated the strength and performance of a pressure vessel under various loading conditions, including internal pressure, nozzle loads, and hydro-test, using Ansys APDL. The validation process aligned the vessel design with ASME Section VIII standards, employing both Design by Rule (Analytical) and Design by Analysis (FEA) with an accurate elastic analysis approach.

The novelty of this work lies in its reliability as a tool for engineers designing vessels, especially in safety-critical industries like oil and gas. The use of hex mesh with a minimum of 3 elements across thickness enhances accuracy, providing more precise guidelines for vessel design. Additionally, the method validates applied boundary conditions by comparing hoop stress from analytical calculations with Finite Element Analysis (FEA), ensuring the accuracy of the FEA model.

The introduction of Stress Classification Lines (SCLs) is another innovative aspect of this study, offering a detailed classification of stresses into components such as membrane stress, bending stress, and total stresses. The application of SCLs at critical points and away from discontinuity regions enhances the understanding of stress distribution, providing valuable insights for design optimization.

The hoop stress check serves as a sanity check for the FEA model, further confirming the correct application of loading. The agreement between analytically calculated hoop stress and FEA results, with a minimal percentage deviation, adds a layer of confidence to the accuracy of the findings.

The results from Load Case 1 and Load Case 2 demonstrate the safety and reliability of the vessel design under various conditions. Despite the localized exceedance of allowable limits in Load Case 1, the study emphasizes the importance of stress classification, as recommended by ASME[7], for a more accurate assessment.

In comparison to existing literature, this study addresses the critical influence of meshing on stress results, emphasizing the superiority of hex mesh over tetra mesh for accuracy. The validation of boundary conditions through a comprehensive comparison of analytical and FEA results adds another dimension to the method's credibility.

Overall, the innovative aspects of this study, including the use of hex mesh, stress classification lines, and boundary condition validation, contribute significantly to the advancement of accurate and reliable methods for designing pressure vessels. The achieved improvement in stress assessment accuracy is pivotal for designing vessels in a more precise and optimized manner, ultimately leading to cost savings in the manufacturing process.

For further future enhancement, the meshing, SCLs creation, and other parametric set can be automated using Python or Matlab. The meshing and SCLs are manually created which is time consuming.

5 Abbreviations

ASME: American Society of Mechanical Engineers; APDL: Ansys Parametric Design Language; BVPC: Boiler and Pressure Vessel Code; API: American Petroleum Institute; FEA: Finite Element Analysis; DBR: Design By Rule; DBA: Design By Analysis; MAWP: Maximum Allowable Working Pressure

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