FINITE ELEMENT ANALYSIS OF NOZZLE FOR VERTICAL PRESSURE VESSEL (MB-CLC401-S012026)

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Abstract
This paper covers 3D modeling of all parts of vertical pressure vessel (MB-CLC401-S012026) using CREO 2.0 parametric software as per ASME code section-VIII division-I. The design of NOZZLE A9 has been modified with introducing reinforcement pad at the connection of nozzle and shell portion. The design has been modified as per ASME code. Considering design data and calculated dimensions of pressure vessel, 3D CAD model has been generated using CREO Parametric modeling software. The static structural analysis has been carried out using ANSYS software for checking the design of NOZZLE A9 of the pressure vessel. The suggested design modification and accuracy of its FEA result has been checked by performing hydro static test at M/s Vijay Tanks And Vessel Pvt Ltd.
Keywords: Vertical Pressure Vessel, CREO 2.0, ANSYS.

I. Introduction

Industrial pressure vessels are usually structures with complex geometry containing number of geometrical discontinuities and are often required to perform under complex loading conditions (internal pressure, external forces, thermal loads, etc.).[4] The design and manufacturing of these products are governed by mandatory national standards, codes and guidelines that ensure high safety performance. Most pressure vessel design codes (e.g. EN13445, BS550, ASME Sec-VIII Div-I) assume a membrane stress state condition for the determination of the minimum shell thickness and large safety factors at areas of geometric discontinuities such as openings, change of curvatures, nozzle intersections, thickness reduction, etc.[14] It should be noted that large safety factors lead to increasing the material thickness, while safety is not necessarily increased; fracture toughness decreases with increasing thickness, and stress corrosion cracking at components operating in corrosive environments is expected to be higher in thicker parts.[3]

Design of pressure vessels is governed by the ASME pressure vessel code. The code gives for thickness and stress of basic components, it is up to the designer to select appropriate analytical as procedure for determining stress due to other loadings. [5]

The pressure vessels are designed with great care because the failure of vessel in service may cause loss of life and properties. The material of pressure vessels may be brittle such as cast iron or ductile as plain carbon steel and alloy steel.

The main component of pressure vessel are,

(1) Shell, (2) Head, (3) Nozzle, (4) Support and the type of pressure vessel A) Horizontal Pressure Vessels, B) Vertical Pressure Vessels, C) Spherical Pressure vessels.

A. Selection of Material for Pressure Vessel

The pressure vessel withstand with

- High or very low temperatures
- High pressure High flow rate
- Sometime corrosive fluid

Material Used for Pressure Vessels

Cast Irons, Plain Carbon Steel, Alloy Steels, Aluminium Alloys, Copper and Copper Alloys, Nickel and Nickel alloy etc. as per application of pressure vessel.

B. Categories of Failures

1. Material- improper selection of material, defects in material 2. Design – incorrect design data, inaccurate or in correct design method 3. Fabrication – poor quality control, improper or insufficient fabrication procedure including welding, heat treatment or forming method. 4. Services- change of services condition by the user, in experienced operation or maintenance personnel. Some types of services require special attention both for selection of material, design detail, and fabrication methods.

Types of failures.

Elastic deformation - Elastic instability or elastic buckling, vessel geometry, and stiffness as well as property of materials are protection against buckling.

Brittle fracture - Fracture can occur at low or intermediate temperatures. Brittle fractures have occurred in vessel made of low carbon steel in the 40°-50° F range during hydro test where minor flaws exist.

Stress rupture - Creep deformation as a result fatigue or cyclic loading.

Excessive plastic deformation - The primary and secondary stress limit as in ASME section VIII, division 2 are intended to prevent excessive plastic deformation.

High strain - Low cycle fatigue is strain governed and occurred mainly in lower strength/high ductile material. Stress corrosion - Chlorides cause stress corrosion cracking in stainless steel.

Corrosion fatigue - Corrosion can reduce the fatigue life by pitting the surface and propagating crack. Material selection and fatigue properties are the major consideration.

II Design and Solid Modeling of Vertical Pressure Vessel.

Design data In Accordance with ASME Section VIII Division 1

Version: 2010 Edition,

Design Internal Pressure =0.745 N/mm²

Design Internal Temperature =115 °C

Projection of Nozzle from Vessel Top =0.0 mm

From Vessel Bottom =500mm

Minimum Design Metal Temperature =-100 °C

Type of Construction =Welded

TABLE I: Summary Of Required Thickness Calculation.

Item detail	Design pressure Kgf/cm^2	Min Thickness mm	Required Thickness mm
Skirt bottom	…	8	…
Bottom Disc	7.6	8	2.43337
Shell	7.6	8	2.43809
Top disc end	7.6	8	2.23727

TABLE II: Summary of Required Weld Sizes.

Required Base ring to Skirt Double Fillet Weld Size	4.7 mm
Required Gusset to Skirt Double Fillet Weld Size	6.3 mm
Required Top Plate to Skirt Weld Size	6.3 mm
Required Gusset to Top Plate Double Fillet Weld Size	0.73 mm

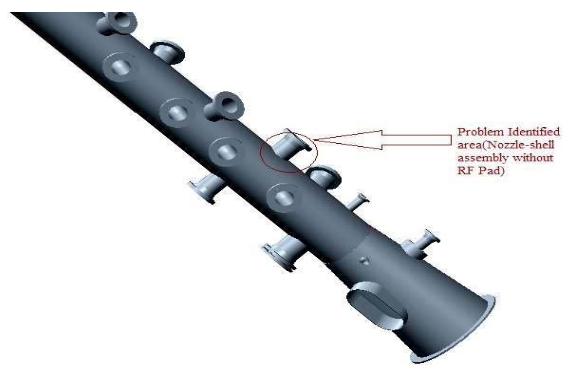


Figure 1. Existing design 3-D model of vertical Pressure Vessel

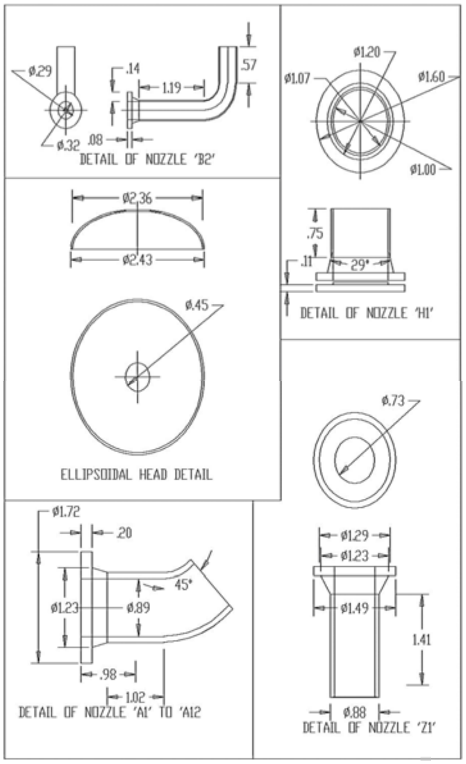


Figure 2. Detail Drawing of Pressure Vessel

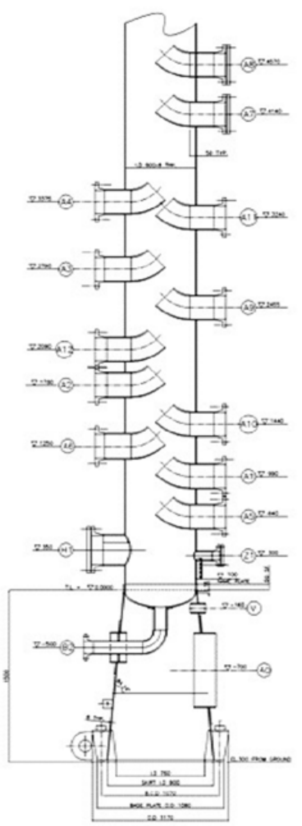


Figure 3. 2D-Drawing of Pressure Vessel.

III Calculation of Nozzle-A9

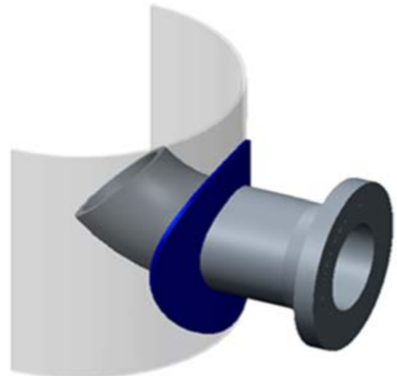


Figure 4. 3D Assembly of NOZZLE A9.

Pressure for Reinforcement Calculation (P) =1.327 N/mm²

Temperature for Internal Pressure (Temp) =115 °C

Maximum Allowable Pressure =1.9 N/mm²

Shell Material=SA-240 TP-304

Shell Allowable Stress at Temperature(S) =134.93 N/mm²

Shell Finished (Minimum) Thickness (t) =8mm

Shell Allowable stress at ambient (Sa) =137.89 N/mm²

Inside Diameter of Cylindrical Shell (D) =600.00 mm

Figure 5. NOZZLE A9

TABLE III: Information about Nozzle A9

Layout Angle	45o
Diameter	203.2 mm
Flange Type	Weld Neck Flange

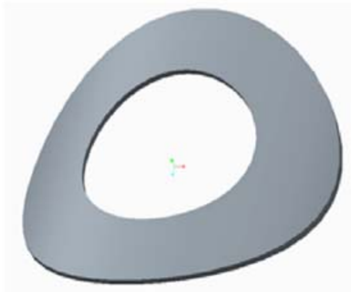


Figure 6. Reinforcement Pad.

Required thickness per UG-37(a)
= (P*R)/(S*E-0.6*P)
= 2.9689 mm
Here Available Nozzle Neck Thickness
= 7.1564 mm (Which is acceptable for safe design)

A. Summary of Nozzle Pressure/Stress Results.
Allowed Local Primary Membrane Stress =206.83N/mm²
Local Primary Membrane Stress (PL) =77.28 N/mm²
Maximum Allowable Working Pressure (Pmax) =3.67 N/mm²

Weld Size Calculations of Nozzle A9
Intermediate Calculation for nozzle/shell Welds (Tmin)
=7.1564 mm
Intermediate Calculation for pad/shell Welds (TminPad)
=8.0000 mm

IV FEA RESULT OF NOZZLE A9 (STATIC STRUCTURAL) without R.F pad

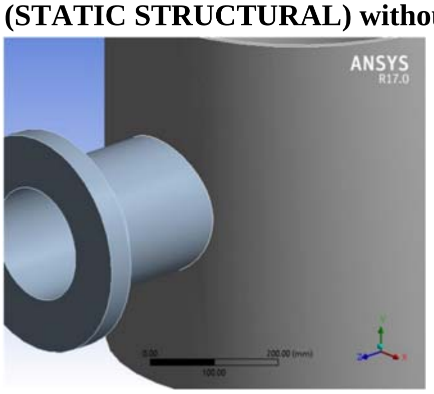


Figure 7. Nozzle A9 detail view from assembly.
TABLE IV: Model Geometry detail of Parts.

Object Name	SHELL BODY	NOZZLE A9
Material		
Assignment	Structural Steel SA 240 TYPE 304	
Bounding Box		
Length X	616. mm	350. mm
Length Y	600. mm	350. mm
Length Z	616. mm	558.24 mm
Properties		
Volume	8.8617e+006 mm³	8.5687e+006 mm³
Mass	69.564 kg	67.265 kg
Statistics		
Nodes	21076	14416
Elements	5937	4555

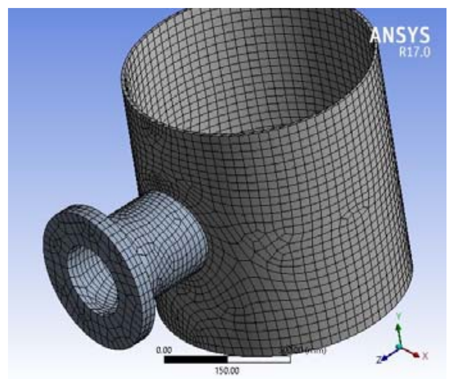


Figure 8. Model Mesh Figure.

TABLE V: Model (A9) Meshing detail.

Object Name	NOZZLE A9
Defaults	
Physics Preference	Mechanical
Relevance	100
Shape Checking	Standard Mechanical
Element Mid-side Nodes	Program Controlled
Sizing	
Relevance Center	Coarse
Element Size	Default
Statistics	
Nodes	35492

Elements	10492
Method	Hex Dominant
Free Face Mesh Type	Quad/Tri

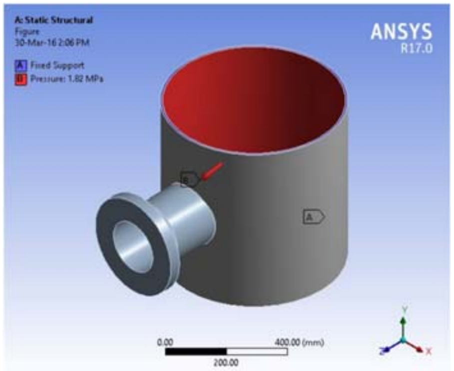


Figure 9. Model Static Structural loading condition.

A. Solution

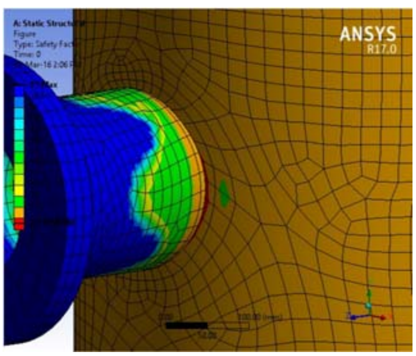


Figure 10. Model Static Structural Solution for Fatigue Tool Safety Factor.

TABLE VI: Model Static Structural Solution Results.

	NOZZLE A9		
Type	Equivalent Elastic Strain	Equivalent (vonMises) Stress	Total Deformation
	Results		
Minimum	1.0738e-006 mm/mm	7.0292e-002 MPa	0. mm
Maximum	8.0364e-004 mm/mm	159.75 MPa	0.25214 mm
Minimum Occurs On	NOZZLE A9		SHELL BODY
Maximum Occurs On	SHELL BODY		

After FEA analysis of Nozzle A9 result shows that Von-Mises stress generated at junction of Nozzle and Shell is 159.75 MPa. And total deformation occurred at same position is 0.25214mm.Figure: indicates that welded joint of NozzleA9 is most critical area where failure can be occurred.

V FEA result of NOZZLE A9 (STATIC STRUCTURAL) with R.F pad.

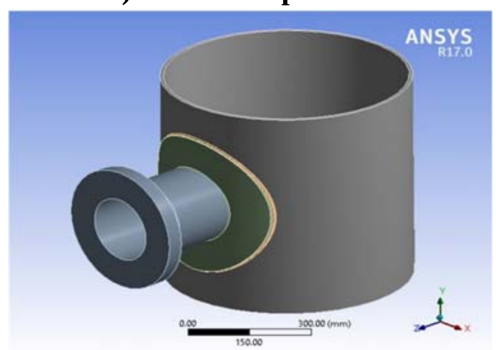


Figure 11. NOZZLE A9 3-D model (With R.F pad).

TABLE VII: Model (A9) Geometry Parts.

Object Name	SHELL BODY	NOZZLE A9	R.F Pad_A9
	Material		
Assignment	Stru ctural Steel SA240 TYPE 304		
	Bo unding Box		
Length X	616. mm	350. mm	400. mm
Length Y	600. mm	350. mm	400. mm
Length Z	616. mm	558.24 mm	81.769 mm
	P roperties		
Mass	69.564 kg	67.265 kg	5.9686 kg
	Statistics		
Nodes	21084	14416	6205
Elements	5956	4555	1788

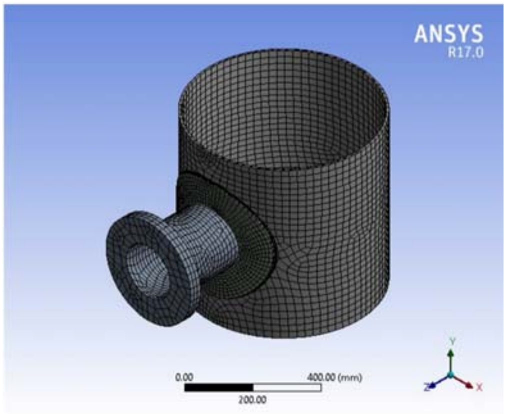


Figure 12. Model (A9) Meshing detail.

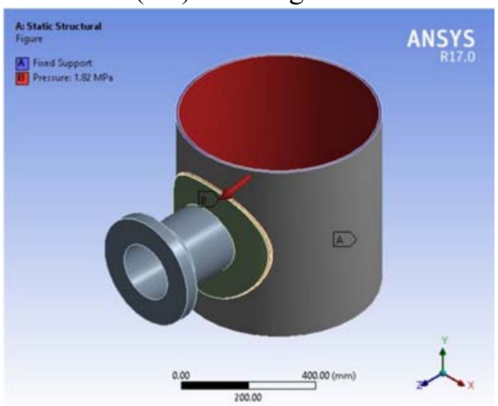


Figure 13. Model Static Structural loading condition.

A. Solution

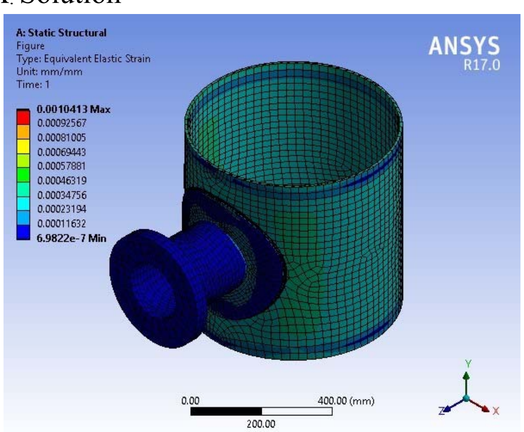


Figure 14. Model Static Structural Solution Equivalent Elastic Strain Figure.

TABLE VIII: Model Static Structural Solution Results.

Object Name	Equivalent Elastic Strain	Equivalent Stress	Total Deformation
Definition			
Type	Equivalent Elastic Strain	Equivalent (von-Mises) Stress	Total Deformation
Results			
Minimum	6.9822e-007 mm/mm	5.9392e-002 MPa	0. mm
Maximum	1.0413e-003 mm/mm	208.26 MPa	0.23601 mm
Minimum Occurs On	NOZZLE_A9		SHELL_BODY
Maximum Occurs On	RAIN_PAD_A9		SHELL_BODY

VI Validation of work with Experimental results.

A. Hydro Static Test.

The pressure vessel to be hydro static tested in vertical position using additional support at test pressure and hold for 2Hr minimum. The supports are to be positioned so as not to impose undue stress in the shell and to minimized deflection. Minimum hydro static test pressure shall be same as test pressure specified on the approved drawing. Inspection shall be made at all joint and connections at the test pressure. After success of testing, pressure is reduced gradually to zero and water is to be completely removed and the inside thoroughly drained of and dried in natural way. Bolt tightening is prohibited during hydro static test.

Test pressure to be applied as follows

Increase pressure gradually at 1/3 of M.A.W.P and hold for about 15 min for inspection. Then increase the pressure to the test pressure as mentioned in table and hold for 2 hr. minimum for inspection. After holding 2 hr. in test pressure, decrease pressure back to M.A.W.P and hold for 15min. finally decrease pressure to atmospheric pressure.

Recording

Pressure and temperature shall be recorded every 15min for 2hr at test pressure and attached on pressure recorder report for their final justification/approval.

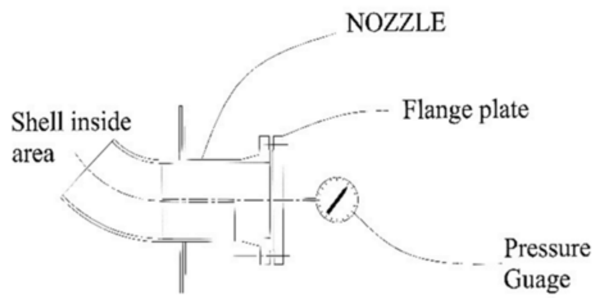


Figure 15. Hydro static test setup line diagram.

B. Experimental setup at VTV pvt ltd.

(A)

VII RESULTS AND DISCUSSION.

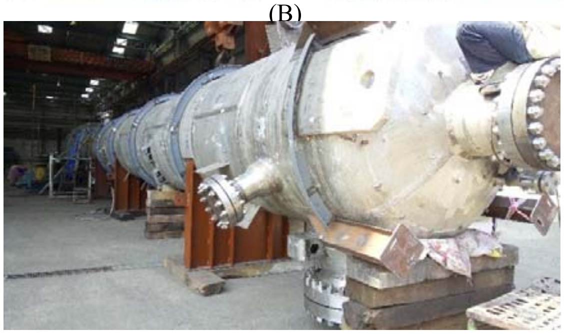
- Design modification of vertical pressure vessel has been done as per ASME code Section VIII Division-I.
- 3D-part and assembly model of vertical pressure vessel has been done as per design dimensions.
- Modified suggested geometry design of NOZZLE A9 gives more accurate and safe design as per the comparative result outcome from the analysis of component.
- Analytical solution in ANSYS 17.0 gives more accurate result obtained that structural design with R.F pad is more safer and with more factor of safety.
- An Analysis result data gives result in deformation that with R.F pad total deformation is 0.23601mm, while total deformation without R.F pad is 0.25214mm. Which gives difference of 0.228539mm. From that we can conclude that with R.F pad assembling of NOZZLE to pressure vessel shell reduce deformation ratio against applied pressure.

TABLE IX: Results outcome from HYDRO Test Performance.

Comp.	MAWP Kgf/cm²	Pressure generated at test Kgf/cm²	[tr] mm	Weld path	Inspection of Areas or Stresses
NOZZLE B2	29.23	31.79	3.73	OK	Passed
NOZZLE Z1	28.84	31.37	4.17	OK	---

- Different load types which are generated while Hydro test are having lower value then the Analytical value as mentioned in Table: which shows that design is more sustainable.
- Table IX: shows the value of stress at different component of pressure vessel while Hydro testing, that values are lower than allowable stress calculated from ASME CODE. Which shows that Design of Pressure Vessel is safest from failure.

(B)



(C)

NOZZLE H1	28.84	31.37	---	OK	Passed
NOZZLE A5	28.84	31.37	4.17	OK	Passed
NOZZLE A1	28.84	31.37	4.17	OK	Passed
NOZZLE A6	28.84	31.37	4.17	OK	Passed
NOZZLE A10	28.84	31.37	4.17	OK	Passed
NOZZLE A2	28.84	31.37	4.17	OK	Passed
NOZZLE A12	28.84	31.37	4.17	OK	Passed
NOZZLE A9	28.84	31.37	4.17	OK	Passed
NOZZLE A3	28.84	31.37	4.17	OK	Passed
NOZZLE A11	28.84	31.37	4.17	OK	Passed
NOZZLE A4	28.84	31.37	4.17	OK	Passed
NOZZLE A7	28.84	31.37	4.17	OK	Passed
NOZZLE A8	28.84	31.37	4.17	OK	Passed

CONCLUSIONS

- Design modification of vertical pressure vessel has been done as per ASME code Section VIII Division- I.
- 3D-part and assembly model of vertical pressure vessel has been done as per design dimensions.
- The comparison of Analytical results and FEA result using ANSYS 17.0 software gives the satisfactory of the suggested modified design of model.
- The FEA result shows that the deformation of the Nozzle A9 with R.F pad is 0.23601mm, and without R.F pad 0.25214mm. Which conclude that Nozzle with R. F pad Pressure vessel shell reduced the deformation ratio against applied pressure.
- The result obtained through Static Structural with R.F pad and without R.F pad are validated with result of HYDRO test performance. The validation shows the reliability of modified design of Nozzle A9.

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