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DESIGN AND ANALYSIS OF MULTI LAYERED PRESSURE VESSEL FOR VARIABLE THERMAL LOADING

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Abstract

The Multi-layer pressure vessel are used as an alternative solution to Mono-block pressure vessel, as this vessel will be design as per peak thermal loading cases where, chances of disaster and explosions are more. In order to avoid the same, the Multi-layered PV will be designed and analyzed using ANSYS and the results will be provided to the industry for fabrication. Along with the thermal stresses, maximum allowable working pressure (MAWP) and keeping the corrosive nature of the handling fluid in mind the analysis has been done. The pressure vessel has been analyzed for the operating fluid of Ethylene Oxide and taking into consideration its adverse effects, the pressure vessel i.e., mono block pressure has been multi-layered for optimizing the thickness and to prevent the same for and sort of failure. In high-pressure applications, multi-layering of pressure vessels is quite beneficial.

1.0 INTRODUCTION

1.1 About Pressure Vessels:

In a variety of sectors, pressure containers are useful. The following are some of the varieties of pressure vessels that are frequently used in various industries and are accessible on the market, according to the pressure vessel maker- Spherical Pressure Vessel , Cylindrical Pressure Vessel , Conical Pressure Vessel .

1.2 Geometrical definition:

The inner diameter of the equipment and the distance between tangent lines are utilized to establish the geometry of a pressure vessel. Because this is a process requirement, the inner diameter should be used.

- Welding line: the point where the head and shell are fused together.
- Tangent line: The point at which the curvature of the skull begins is known as the tangent line.

1.3 Multi-Layer Pressure Vessel:

This invention relates to a method of fabricating multi-layer pressure vessels, and more specifically to the pre-stressing of the vessel's cylindrical walls during fabrication. It is possible to pre-stress the vessel's cylindrical wall to compensate for the different rates of stress increase 1 (in the metal of the vessel's inner and outer sections under working load. This pre-stressing is achieved by squeezing the structure during the welding of each layer to the point where the metal of the layers flows elastically, resulting in initial residual compression stresses in the vessel's inner portion and first residual tension stresses in the outer part. During this procedure, it has been discovered that there is a tendency for the innermost layer or layers to be overly compressed, resulting in the vessel's overall efficiency being less than desirable. In other words, as additional layers are put to the structure, the stress tends to collect or concentrate in the innermost layers. Various strategies for reducing stress concentrations or leveling out the stress distribution in the structure have been presented. The majority of these solutions necessitate extra, costly stages. When produced in the way described, the primary goal of the current invention is to provide a method for preventing an excessive concentration of compressive stress in the structure's innermost layers. The innovation is based on the discovery that when pre-stressing cylinders, the stress differential between the first few layers was bigger than the stress differential between the outer layers, and that this stress differential was kept in the finished structure. The stress differential between each layer and the one before it is determined by the relative thickness of the current layer and the layers before it.

1.4 The invention claimed:

The process of pre-stressing a multi-layer high pressure cylinder during manufacture in order to enhance stress reduction when it is in operation. Multilayer pressure vessel consists of different layers joined together by the process of shrink fitting. Thermally expanded shell is fitted over the other shell which is at normal temperature. The system is cooled to achieve the desired interference. During the fabrication of Pressure vessel, we prefabricate an internal support structure made of channels or angles to order to minimize the local concentration of stress resulting in localized deformation. We can also utilize a circumferential external support structure for pre-stressing and achieving desired interference. Thus the vessel is built up by shrink fit with two or more concentric shell progressively shrunk together inside out. Thus by increasing the number of layers reduces the resulting working stresses during operation and range of stress between maximum and minimum value. Optimum number of layers considered in this case is three only, as it is not economically advised to go beyond it due to high fabrication cost as compared to a negligible reduction in hoop stress value.

1.5 Design Philosophy for this Current Study:

The current research is an industrial project that focuses on the necessity for multi-layered (High Pressure Vessels) due to its numerous benefits. It has provided a means for the process designer to efficiently handle a variety of processes while maintaining optimal control over the process parameters. This study compares single-layer (Mono-block Pressure Vessels) and multi-layer pressure vessels. The primary focus of research for a pressure vessel is to develop it to withstand the numerous stresses that occur during the processes. As a result, one of the primary challenges is the quantification of stress. The data for this study was gathered from the industry, and the FEA analysis was performed on a single-layer pressure vessel. The circumferential and hoop stresses obtained from this investigation have been investigated and studied. The vessel is then Multi-layered with the use of liner material at a later stage. a lining material with a thickness of 3mm was chosen for multi-layering purposes; it is essentially an Austenitic stainless steel.

1.6 Problem Definition:

Multi-layer pressure vessels are used as an alternative solution to Mono-block pressure vessel, as this vessel will be design as per peak thermal loading cases where, chances of disaster and explosions are more. In order to avoid the same, the Multi-layered PV will be designed and analyzed using ANSYS and the results will be provided to the industry for fabrication. Meanwhile the thickness of PV will be also optimized. Along with the thermal stresses, maximum allowable working pressure (MAWP) and keeping the corrosive nature of the handling fluid in mind the analysis has been done.

2.0 OBJECTIVE

i. To optimize the thickness of Multi-layer pressure vessel for the stresses generated due to peak thermal load.
ii. To provide the ANSYS analysis report to the industry to confirm the safety aspects of pressure vessel during its operating conditions & for peak thermal hike cases.
iii. To generate database for the industry for similar kind with the same type/capacity and volume of operating fluids.

3.0 METHODOLOGY

The design calculations are being done according to the code and the study is further taken for peak thermal analysis, for this purpose a finite element package such as ANSYS is used. The Pressure vessel is been analyzed for Steady-State Thermal loadings and for Static Structural are also been carried out for the same, the Results are then compared and if the Stresses are well defined under the permissible limit the Pressure vessel is taken NDT testing and final dispatch is been done.

4.0 DATA COLLECTION AND ANALYSIS OF DATA

4.1 Design Data from Industry:

This is an industrial project and hence the following basic engineering package will be taken from the industry, they are as follows, Operating Temperature and Pressure, Operating Volume Capacity, Percent Filling,Water Capacity,Nature of Service Fluid-Corrosive/Hazardous/abrasive/lethal, Required MOC,Type of Ends, Nozzle Schedule, Preference on Corrosion allowance, if not, will have to be calculated, Design Code, preferred/specified by Client / Process Department. If not, will have to select.

4.2 Design of PV by Process Engineer:

A Process Engineer/Process Designer is the one who lays the foundation of design code, ASME calculations and all the appropriate parameters that are usually required for the process. For our study all the data pertaining to the process is taken from the Process Designer for carrying out the further analysis.

4.3 Design of PV as per the ASME Code:

The Pressure Vessels are mostly designed as per following codes:

a. American Code : ASME Sec VIII Div.-I
b. Indian Code : IS: 2825
c. British Code : PD: 5500
d. German Code : AD-MerkBlatter

Design Basis / Input Data from Process Department:

During start of the Project or the design of the pressure vessel, following data is required from the Process Department / Personal / Client- Shape of Vessel, Position: Horizontal or Vertical, Process Fluid, Design Temperature, Design Pressure.

Table 4.3.1 Design Codes

Sr. No.	Country	Name of Design Code	F.O.S on	
			Tensile Stress	Yield Stress
1	American Code	ASME Sec VIII Div.-I	3.5	1.5
2	Indian Code	IS:2825	3	1.5
3	British Code	PD:5500	2.35	1.5
4	German Code	AD-Merkblatter	-	1.5

4.4 Basics for selection of Design Code:

The code which gives optimum thicknesses, more safety (Factor of Safety) and more life to the equipment's shall be selected. The decision for selecting the code may be broadly based on Factor of safety, but this may not yield optimum or lesser thicknesses. Hence the final decision will be primarily designer's preview. From the above table, it is observed that the F.O.S on yield stress of all the codes are same i.e., 1.5. Whereas on tensile stress, it is Maximum in ASME Sec VIII Div.-I & Lowest in PD: 5500.Hence, it is inferred that, using PD: 5500 will result in lower thicknesses, lesser safety and ASME Sec VIII Div.-I will result in higher thicknesses & more safety. Use ASME Sec. VIII Div-I as design code for pressure vessels if not specified / preferred by Client.

4.4.1 Design Pressure:

The design pressure is the differential between the internal and exterior pressures that is used to establish the minimum needed thickness of each vessel component. It has a sufficient safety buffer above the operational pressure. (Design pressure = 10 % of operating pressure or 10 psi minimum) plus the static head of the operating fluid.

Design Pressure = 1.1 x Operating Pressure + Static Head due to service liquid Minimum design pressure for a Code non-vacuum vessel is 15 Psig.

4.4.2 Maximum Allowable Working (Operating) Pressure:

At the required temperature, it is the maximum gauge pressure allowable at the top of the completed vessel in its operational configuration. It is calculated using the nominal vessel thickness, excluding corrosion allowance and thickness required for loads other than pressure. In most cases, it will be equal to or very close to the design pressure of the pressure vessel or component.

4.4.3 Design Temperature:

The style is unique. Temperature is the working fluid's maximum temperature +500F as a safety margin, or the operating fluid's minimum temperature if the vessel is built for low temperature service (below -200F) It has a sufficient safety buffer above the operational temperature (50 to 100 percent of operating Temperature)
Design Temperature = 1.5 x Operating Temperature
Design Temperature = 2.0 x Operating Temperature

4.5 Selection of Material of Construction (MOC) and Material Specifications:

Table 4.5.2 Acceptable Pressure Vessel Materials

Service Temperature (˚F)		Plate	Pipe	Forgings	Pressure Bolting	Structural Bolts, Nuts. Shape
Cryogenic	-425 to -321	SA 240 Type 304,304L,347	SA 312 Type 304,F304L,F347	SA182 Grades F304,F304L,F347	Bolts :SA 320Gr B8	Same as Pressure Parts
	-320 to -151	SA 240 Type 304,304L,316,316L,	SA 312 Type 304,304L,316,316L ,	SA182 Grades F304,F304L,F316, SA 522	Nuts: SA194 Gr. 8	
		SA 353	SA333 Gr.8			
Low Temperature	-150 to -76	SA 203Gr.D or E	SA333 Gr.3	SA350 Gr.LF3	Bolts :SA 320Gr L7	
	-75 to -51	SA 203Gr.A or B	SA333 Gr.3	SA350 Gr.LF3	Nuts: SA194 Gr. 4	
	-50 to -21	SA 516 all grades impact tested	SA333 Gr.1,6	SA350 Gr.LF3		
	-20 to -44	SA 516 all grades over 1” thk. impact tested				
	+5 to +32	SA 516 all grades over 2” thk. impact tested				
Intermediate Temperature	+61 to +775	SA 285 Gr. C (3/4” Max),	SA 53 (Seamless) or SA 106	SA 181 Gr. I or II		
		SA515 Gr.55,60,65 (1.5” max)	SA 335 P1	SA 105 Gr. I or II		
		SA516 all grades, all Thk.				
		SA 204 Gr. B all Thk.				

4.5.2 Corrosive Service: As protective lining materials, glass, rubber, enamel, lead, and Teflon have all shown to be effective. The use of such linings necessitates the use of specialized fabrication techniques. Stainless steel, on the other hand, is the most prevalent and commercially accessible corrosion-resistant material.Solid stainless-steel plates are most cost-effective for vessel shells up to 10mm thick; Carbon or low alloy carbon steels with applied corrosion resistant layers are utilized in general above this thickness. Attaching the protective layer to the carbon steel plate can be done in three ways.

4.5.3 About Stainless Steels: Stainless steels have a chromium concentration of 11 % or more, but less than 30 percent, and are noted for their exceptional corrosion resistance.They are heat resistant alloys if they include more than 30% chromium. Stainless steels are chromium-iron alloys.

They are used due following common reasons.
1) Necessary resistance to corrosive environment.
2) Increased service life.
3) Safety of working personnel.
4) Strength & oxidizing resistance at elevated temp.

5) Impact strength at low temperature.
6) Providing the cleaning of equipment.

Stainless steel is split into three classes based on the main alloying elements.

1) Straight chromium group (400series) with Chromium content upto 30%
2) Chromium-Nickel Group (300 series)- Standard 18% Chromium & 8% Nickel (percentage may vary)
3) Chromium-Nickel –Manganese Group (200 series)

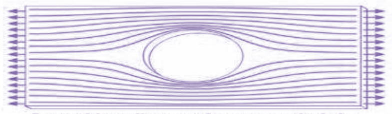
4.5.4 Type of Dish Ends:

Flat head or Covers Formula for Thickness calculation: $th = CD\sqrt{\frac{P}{f}} + CA$	**Tori spherical Head** Formula for Thickness calculation: $th' = \left[\frac{PRcW}{(2fJ - 0.2P)}\right] + CA$ $W = \frac{1}{4}\left[3 + \sqrt{\frac{Rc}{R1}}\right]$
Hemispherical Head Formula for Thickness calculation: $th' = \frac{PDi}{(4fj - 0.4P)} + CA$ **th = 1.06 th'**	**Elliptical Head** Formula for Thickness calculation: $th' = \frac{PDi}{(2fJ - 0.2\,P)} + CA$ **th = 1.06 th'**

4.6 Stress concentration:

A stress concentration (also known as a stress raiser) is a point in an item where stress is concentrated. Because, the force is spread uniformly over an object's area, a loss in area, such as that generated by a crack, results in a localized increase in stress.

Figure 4.6.0 Stress Concentration



4.6.1 WRC- Welding Research Council

Under Section 501 (C) (3) of the Internal Revenue Code, the Welding Research Council continues to operate as a non-profit research organization. In November 1967, WRC became an independent company. Its main goals are to: • In diverse laboratories, conduct needed joint research in welding and closely related topics.
• Information about welding research should be disseminated.
• Universities should support welding research.
• Work with comparable organizations in other countries..

Table 4.6.1 Comparison of FEA, WRC107, WRC 297 for SIF (Stress intensification Factor)

The Table below demonstrates an example of Output				
Source	Axial	In-Plane	Out Of Plane	Torsion
FEA	9.9	2.5	5.6	2.4
WRC 107	11.6	3.5	6.9	6.9
WRC 297	22.5	4.3	9.4	9.5

The following are examples of stress patterns in cylinders:
Circumferential stress or Hoop stress:
In the tangential direction, a normal stress applied.
Axial stress: Normal stress parallel to the axis of cylindrical symmetry
Radial stress: Stress in directions parallel to the symmetry axis but perpendicular to it.

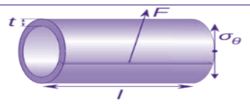


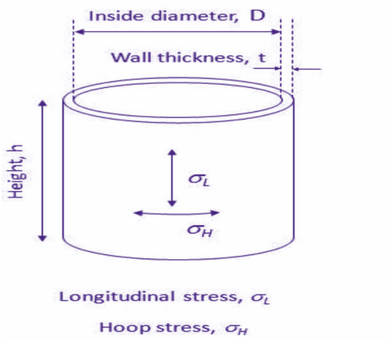
Figure 4.6.1 Stress in the Cylinder

The Circumferential Stress is the one which has the most adverse effect on the cylinder vessel.

Figure 4.6.2 Tee and Pipe Neck on the Shell

Stress concentration near Nozzle area

Figure 4.6.3 Stress Concentration near Nozzle Area



- Forces due to internal pressure are balanced by shear stresses in wall
- Horizontal section:

$$P\frac{\pi D^2}{4} \approx \sigma_L \pi D t$$

- Vertical section:

$$\sigma_L = \frac{PD}{4t}$$

- Similar equations can be derived for other geometries such as heads (see Ch 13)

$$PhD = \sigma_H 2ht$$

$$\sigma_H = \frac{PD}{2t}$$

The Process of designing & analyzing of Pressure vessel which is handling Ethylene oxide C2H4O also colourless flammable gas is mentioned.

ASME SEC VIII DIV. 1, EDITION 2010 & SMPV (U) RULE-1981 guidelines are used to consider the various criteria. Design pressure = 7.2 Kg/cm², Design temperature = 55ºC

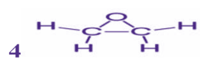


Figure 4.6.4 Chemical Bond Structure

Figure 4.6.5 IUPAC Structure

4.7 Design Calculation as per ASME Section VIII Division 1 and Selection of PV as per Process Requirements:

Design Code	:	ASME SEC VIII DIV.I ,Edition 2010, Addenda-2011 & SMPV(U) Rule-1981
Equipment	:	Ethylene Oxide Storage tank
Tank Capacity	:	19.9 M^3 WATER CAPACITY
Equipment No	:	T-101/102
Customer	:	MDP Solutions. Pvt. Ltd Kandivali, Mumbai
Project	:	P-1605

DESIGN DATA

1)	Design code	:	ASME SEC VIII DIV.I ,Edition 2010, Addenda-2011, & SMPV(U) Rule-1981		
2)	Service	:	Ethylene Oxide (EO)		
3)	Type	:	Horizontal, limpetted		
4)	Inside Díameter	:	1950	Millimeter	
5)	Shell Length	:	6000	Millimeter	
6)	End closures type	:	2:1 Ellipsoidal Dished Ends		
			Vessel Side		
7)	Design pressure	:	7.0 +	0.195	(Static Head)
		:	7.2	Kg/cm^2 (g) Incl. Static Head	
		:	102.3 Psi(g)		
8)	Design Temperature	:	-6 to +55 Deg.C =		131
9)	Operating Pressure	:	3.5	Kg/cm^2 (g)	
10)	Operating Temperature	:	35 Deg.C	=	95
11)	Gross Geometric Vol.	:	19.86 m ^ 3		
12)	Design Sp.gravity	:	1	Op. Sp. Gravity :	
13)	Filling Percentage	:	90%		
14)	Filing Ratio	:	0.72		
15)	E.O Capacity	:	14300.0 Kg.	=	14.3 MT

VOLUMETRIC , STATIC HEAD & HYDROTEST PRESSURE CALCULATIONS

1) Volumetric calculations:					
d,	Shell Inside diameter		=	1950 MM	
L,	Shell Length (T.L to T.L)		=	6000 MM	
	1) Vol.of the cylindrical shell		=	0.7854 x d ^ 2 x L	
			=	17.919 m^3	
	2) Volume of 2 Nos. 2:1 Ellip. Dished Ends				
			=	2 x 0.1309 x id^3	
			=	1.941 m^3	
V,	Total water filled capacity		=	19.860 m^3	
	Type of vessel		=	Horizontal, limpetted	
	Specific Gravity of Liquid		=	1	
D	I.D of vessel		=	Shell Diameter	
			=	1950 MM	
P	Pressure due to static head		=	D x Density of Liquid x % Filling	
			=	1950	
			=	0.195	
Hence, pressure due to static head			=	2.774 Psi	
	Total Design Pressure		=	Design Pressure + Static head	
			=	99.56 Psi(g) +	
			=	102.40 Psi(g)	
	Total Design Pressure		=	7.2	

SHELL THICKNESS FOR INTERNAL PRESSURE

D,	Inside shell diameter						=	1950	mm
L,	Length of shell						=	6000	mm
P,	Total design pressure	(Incl static head)		=	7.2	kg/cm^2(g)	=	102.4	Psi(g)
	Design temperature			=	55	ºC	=	131	°F
E,	Joint efficiency of shell						=	1	
	Material of construction for shell						=	**SA 240 Gr. 304L**	
R,	Inside radius of shell (corroded)						=	975	mm
	Max. allowable stress at ambient temperature.						=	16700	Psi.
S,	Max. allowable stress at design temperature for shell						=	16700	Psi.
	Corrosion allowance						=	0	mm

THICKNESS OF SHELL UNDER INTERNAL PRESSURE :				
(In terms of Inside Dimensions)				
For circumferential stress				
t_c	=	P x R		
		(S x E - 0.6 x P)		
	=	16700 x 1 - 0.6 x 102.4	=	6.001
Required thickness under circumferential stress			=	6.001
Minimum reqd. thickness of shell (incl c.a.)			=	6.001
Minimum Required Shell Thk as per Code			=	1.6
Provided thickness of shell			=	8
PROVIDED THICKNESS OF SHELL IS ' MORE' THAN THE MIN.REQD.THICKNESS				
HENCE SHELL THICKNESS IS SAFE FOR INTERNAL PRESSURE				

DISHED END THICKNESS FOR INTERNAL PRESSURE

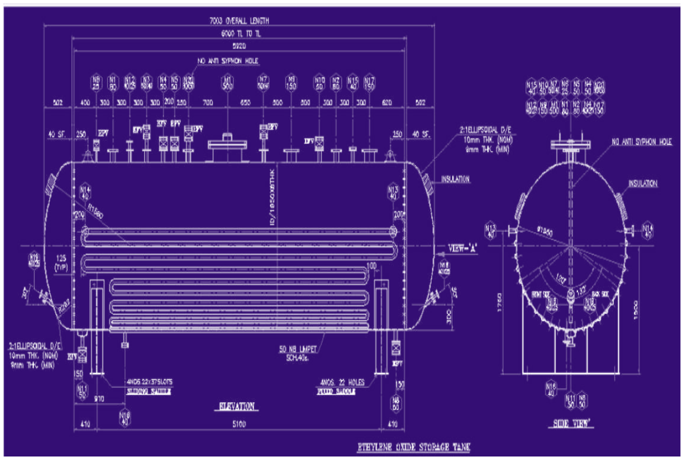
Dished End Type		=	2:1 Ellipsoidal Dished Ends				
Inside Dished End diameter					=	1950	mm
Inside Crown Radius					=	1755	mm
Inside Knuckle Radius					=	331.5	mm
Ratio of Dia. & Twice of Height					=	2	
Factor					=	1	
Total design pressure	(Incl static head)	=	7.2	kg/cm^2(g)	=	102.4	Psi(g)
Design temperature		=	55	ºC	=	131	°F
Joint efficiency of Dished End					=	1	
Material of construction for Dished End					=	**SA 240 Gr. 304L**	
Max. allowable stress at ambient temperature.					=	16700	Psi.
Max. allowable stress at design temperature for Dished End					=	16700	Psi.
Corrosion allowance					=	0	mm

THICKNESS OF Dished End UNDER INTERNAL PRESSURE :					
t	=	P x D x K			
		(2 S x E - 0.2 x P)			
		2 x 16700 x 1 - 0.2 x 102.4			
Required thickness under circumferential stress			=	5.98	mm
Minimum reqd.thickness of Dished End (inclC.A.)			=	5.98	mm
10% Thinning Allowance			=	1	mm
Required Thk. With C.A & T.A			=	6.98	mm
Minimum Required Dished End Thk as per Code			=	1.6	mm
Provided thickness of Dished End			=	10	mm
PROVIDED THICKNESS OF DISHED END IS ' MORE' THAN THE MIN.REQD.THICKNESS					
HENCE DISHED END THICKNESS IS SAFE FOR INTERNAL PRESSURE					

5.0 CAD MODEL AND FEA ANALYSIS

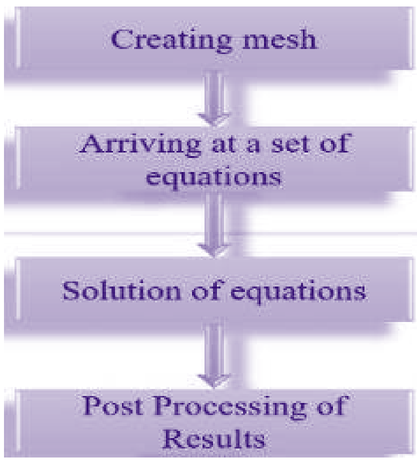
5.1.2 D model In AutoCAD:

Figure 5.1.1 AutoCAD Drawing



5.2 FEA Analysis by ANSYS

Figure 5.2.1 Steps in FEA



Gives comprehensive results, It is feasible to simulate potentially unsafe or unworkable load circumstances and failure scenarios in a safe manner, Allows rapid analysis of performance as various parameters can be calculated simultaneously, Low investment and rapid calculation time.

Shortcomings of the approach: The regulations didn't account for everything a designer might wish to consider, Thermal gradients, piping loads, nozzle loads, quickly varying loads, seismic loads, wind, and so on are all factors to consider. The stress was computed using the average membrane stress and did not include any other types of stresses such as thermal loads

Design by Analysis Approach:

Design by Analysis with Section III and Section VIII, Division 1 of the code addressed the flaws of the Design by Formula approach. The designers can estimate stresses anywhere in the vessel, not simply the membrane stresses in regular sections, according to Section III and Section VIII, Division I of the regulation. Finite Element Analysis is used in this Design by Analysis method.

6.0 RESULTS ANALYSIS

1Single Layer Pressure Vessel Analysis:
Material Data
SA240 GR304 (Base Material)

Density	7.2e-006 kg mm^-3
Thermal Conductivity	1.62e-002 W mm^-1 C^-1

Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa	Temperature C
2.e+005	0.3	1.6667e+005	76923	

Tensile Yield Strength MPa
170

Compressive Yield Strength MPa
170

Tensile Ultimate Strength MPa
285

Material Data- For Multi-Layering:

SA240 GR304 (Base Material): Stainless Steel Gr

Density	7.2e-009 kg mm^-3
Thermal Conductivity	1.62e-002 W mm^-1 C^-1

Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa
2.e+005	0.3	1.6667e+005	76923

Tensile Yield Strength MPa	Compressive Yield Strength MPa	Tensile Ultimate Strength MPa
170	170	285

Liner Material

6.1 Steady State Thermal Analysis:

The steady-state thermal analysis is used to determine the thermal equilibrium of a system in which the temperature remains constant across time.

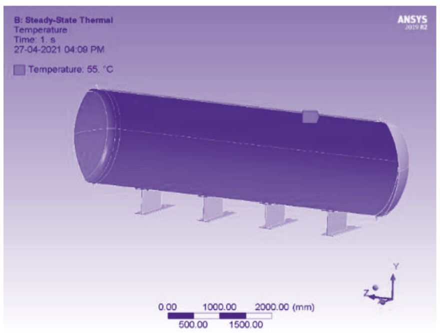


Figure 6.1.1 Temperature 55 Degree Celsius

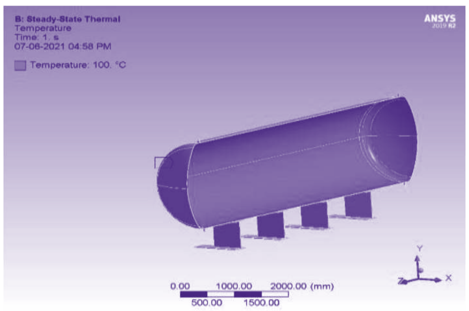


Figure 6.1.2 Temperature 100 Degree Celsius

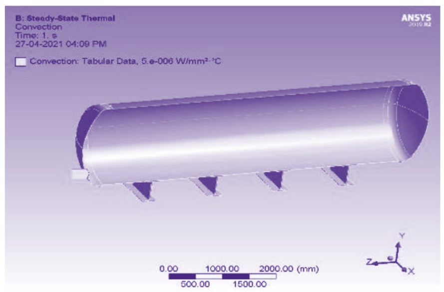


Figure 6.1.3 Convection

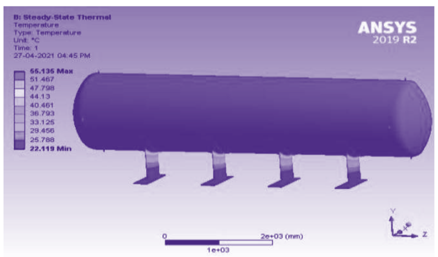


Figure 6.1.4 Temperature Distribution 55 Degree Celsius

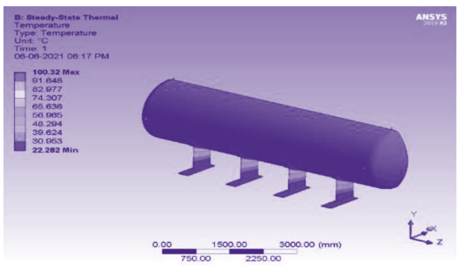


Figure 6.1.5 Temperature Distribution 100 Degree Celsius

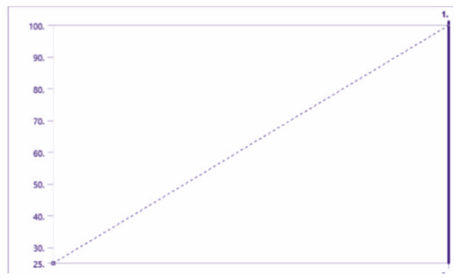


Figure 6.1.6 Temperature Distribution Graph for 100 Degree Celsius

A Gradual Steady-State Thermal Analysis is been carried out and a gradual temperature distribution is been shown, as one can see there is a gradual increase in temperature from 22.119 degree Celsius to 55.135 degree Celsius.A Temperature Distribution is shown Above as seen there is a linear rise in the temperature profile. Also, a gradual temperature distribution up to 100 degrees Celsius is been depicted

6.2 Static Structural Analysis:

The displacements, stresses, strains, and forces caused in structures or components by loads that do not cause significant inertia or damping effects are identified using a static structural analysis. The assumption is for stable loading and response circumstances, which indicates that the loads and the structure's response will change slowly over time. The ANSYS, Samcef, or ABAQUS solvers can be used to calculate a static structural load on structure. In a static analysis, the following forms of loading can be used:

- Forces and pressures exerted externally.
- Inertial forces in a steady condition (such as gravity or rotational velocity).
- Non-zero displacements imposed.
- Strain caused to due Temperature.

6.3 Load Constraints:

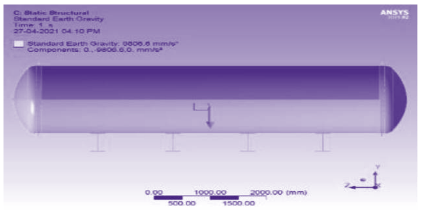


Figure 6.3.1 Standard Earth Gravity

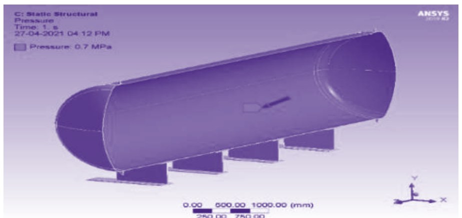


Figure 6.3.2 Pressure 0.7 Mpa

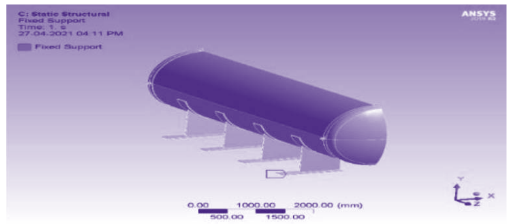


Figure 6.3.3 Fixed Support

6.4 Results for Static Structural Analysis:
6.4.1 Radial Stress Variation:

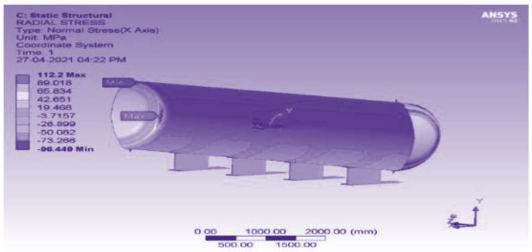


Figure 6.4.1 Radial Stress X-Axis Single- Layer, Max Value – 112.2 Mpa

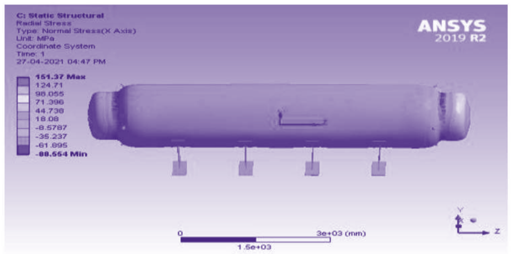


Figure 6.4.2 Radial Stress X-Axis Multi-Layer, Max Value – 151.37 Mpa

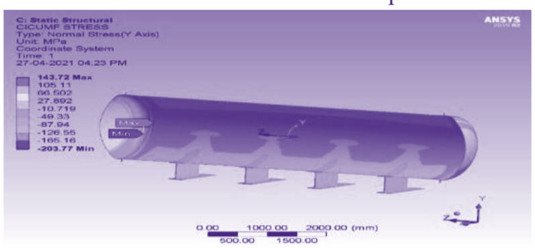


Figure 6.4.3 Circumferential Stress X-Axis Single- Layer, Max Value -143.72 Mpa

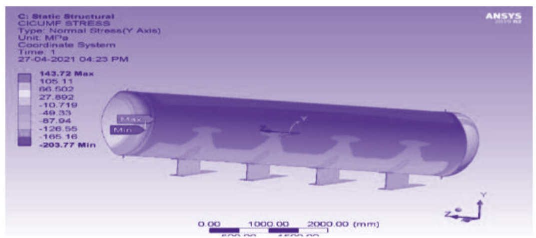


Figure-6.4.4 Circumferential Stress X-Axis Multi- Layer, Max Value -84.9282

6.4.2 Circumferential (Hoop) Stress Variation:

Directional Deformation (X-Axis)

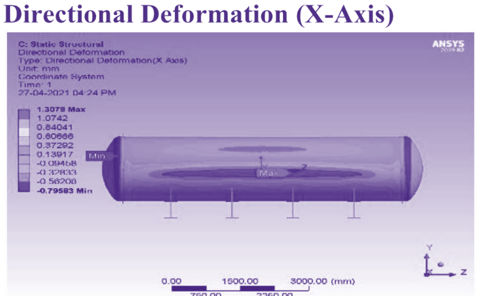


Figure 6.4.5 Single-Layer Max Value- 1.3079 mm

Directional Deformation (Y-Axis

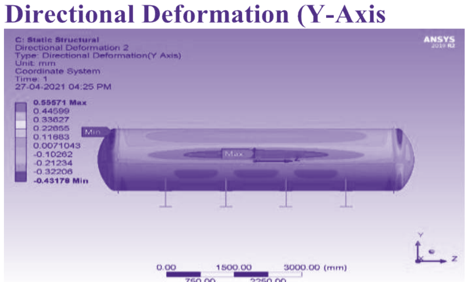


Figure 6.4.7 Single-Layer Max Value- 0.5557 mm

Directional Deformation Z-Axis

Figure 6.4.9 Single-Layer Max Value- 2.1051 mm

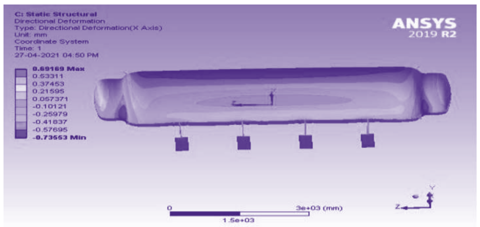


Figure 6.4.6 Multi-Layer Max Value- 0.6916 mm

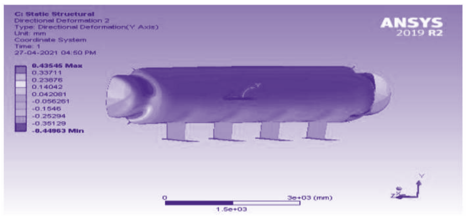


Figure 6.4.8 Multi-Layer Max Value- 0.4354 mm

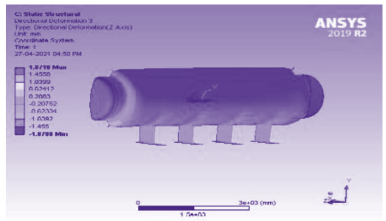


Figure 6.4.10 Multi-Layer Max Value- 1.8716 mm

6.5 Results with Comparison

6.5.1 Result for Single- Layered Pressure Vessel:

Table 6.5.1 Results for Single-Layer Pressure Vessel

Type	Normal Stress		Directional Deformation			Normal Elastic Strain	
Orientation	X Axis	Y Axis	X Axis	Y Axis	Z Axis	X Axis	Y Axis
MAX	112.2 MPa	143.72 MPa	1.3079 mm	0.55571 mm	2.1051 mm	6.015e-004 mm/mm	5.2856e-004 mm/mm
MIN	-96.449 MPa	-203.77 MPa	-0.795 mm	-0.4317 mm	-2.1079 mm	-5.6821e-004 mm/mm	-8.7183e-004 mm/mm

6.5.2 Result for Multi-layered Pressure Vessel:

Table 6.5.2 Results for Multi-Layer Pressure Vessel

Type	Radial Stress	Circumferential Stress	Normal Elastic Strain	Normal Elastic Strain 2	Directional Deformation	Directional Deformation 2	Directional Deformation 3
MAX	151.37 MPa	84.928 MPa	7.7683e-004 mm/mm	3.1411e-004 mm/mm	0.69169 mm	0.43545 mm	1.8716 mm
MIN	-88.55 MPa	-163.56 MPa	-2.533e-004 mm/mm	-7.181e-004 mm/mm	-0.73553 mm	-0.44963 mm	-1.8708 mm

7.0 CONCLUSION

The pressure vessel is been analyzed for the operating fluid of Ethylene Oxide, and taking into consideration its adverse effects the pressure vessel i.e., mono block pressure is been multi-layered for optimizing the thickness and to prevent the same for and sort of failure. In comparison, it has been discovered that increasing the number of layers in a pressure vessel reduces the hoop stresses. The analysis is against the failure modes that stress measurements must be compared and interpreted. A uniform Stress Distribution is been observed which is the indication of a suitable and optimal Analysis. From the above comparison its clear that Multi-Layer Pressure Vessel Provide a fine extension for thickness optimization and hoop stress reduction. As seen by Multi-Layering the Vessel there is a substantial Reduction in the stresses. Because of the advantages of multi-layered pressure vessels over mono-block pressure vessels, they are becoming increasingly popular, Multi-layered pressure vessels have been found to be superior in high-pressure and high-temperature operating environments.

8.0 FUTURE SCOPE

1. Material optimization by reducing thickness for the specified parameters.
2. Using stresses created in a monolayer vessel of the same size as acceptable stresses to determine permissible pressure in a multilayer vessel.
3. Evaluation of various layer materials in order to lower production costs.
4. Costing, operational life, and failure resistance, among other things, will be improved.
5. Material costs have increased, which can be mitigated by employing higher-quality materials.

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