FINITE ELEMENT ANALYSIS OF COMPOSITE REACTOR PRESSURE VESSEL

By

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Abstract

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Reactor Pressure Vessel is the container designed to hold gases or liquids above atmospheric pressure. The focus of this study is to develop Finite Element Model to predict the performance of composite reactor pressure vessel subjected to extreme pressure and temperature and compare with pressure vessel manufactured using conventional materials. In the first part of the study, a pressure vessel system with Carbon steel material under design pressure and temperature have been analyzed for stresses, deformation and Safety Factor. Various loading cases have been discussed during the analysis of pressure vessel.

Composite materials have been a suitable replacement to the conventional material due to high strength to weight ratio, Better chemical resistance, and Good Insulating properties. In the Second part of this study, finite element analysis of internally pressurized cylindrical vessel with closed ends metallic liner overwrapped with composite has been presented. The structure was examined using various composite materials like E-glass, S-glass and Carbon Fibre or combination of these materials placed with different ply angles under same boundary conditions. It was observed that pressure vessel with metallic liner and composite wrapping is safer and lighter than carbon steel under applied loading conditions.

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Chapter 1: Introduction

Reactor Pressure vessel is a container that holds liquids/ gases at pressure and temperature above atmospheric pressure and temperature. In general terms, it is the vessel used to carry out a chemical reaction. Thus, it is necessary that pressure vessels are designed for a good factor of safety, maximum safe operating pressure and temperature, and corrosion allowance. In industry, pressure vessels are usually in cylindrical or spherical shape, with different head configurations. Pressure vessels are used in wide range of industries based on the operating pressure, shape, size and construction material. Pressure vessels are used in Oil and Gas, Petrochemical, Pharmaceutical, Aerospace industries and Nuclear plants.

Traditionally, pressure vessels are manufactured using different alloys of steel. Due to the high density of steel material, pressure vessels are heavy. Composite materials can be suitable replacements for these materials due to high strength to weight ratio, high stiffness. In this study shell and dish end of the pressure vessel is overwrapped with low density and high stiffness composite material in both helical and hoop direction. This research focusses on a comparative study between Carbon steel, Composite materials and investigates the cause of failure for a pressure vessel metallic liner overwrapped with composite material.

1.1 Pressure Vessel

Pressure vessels are defined in American Society of Mechanical Engineer section VIII, Div 1 introduction as “Pressure vessels are containers for containment of pressure either external or internal. The pressure may be from an external source, or by application of heat from a direct or indirect source, or any combination thereof.[1]” Pressure vessels consist of different components such as dish ends, Shell, Jacket, Limpet/ Half pipe coil, and support. Figure 1.1 shows different components of reactor pressure vessel.

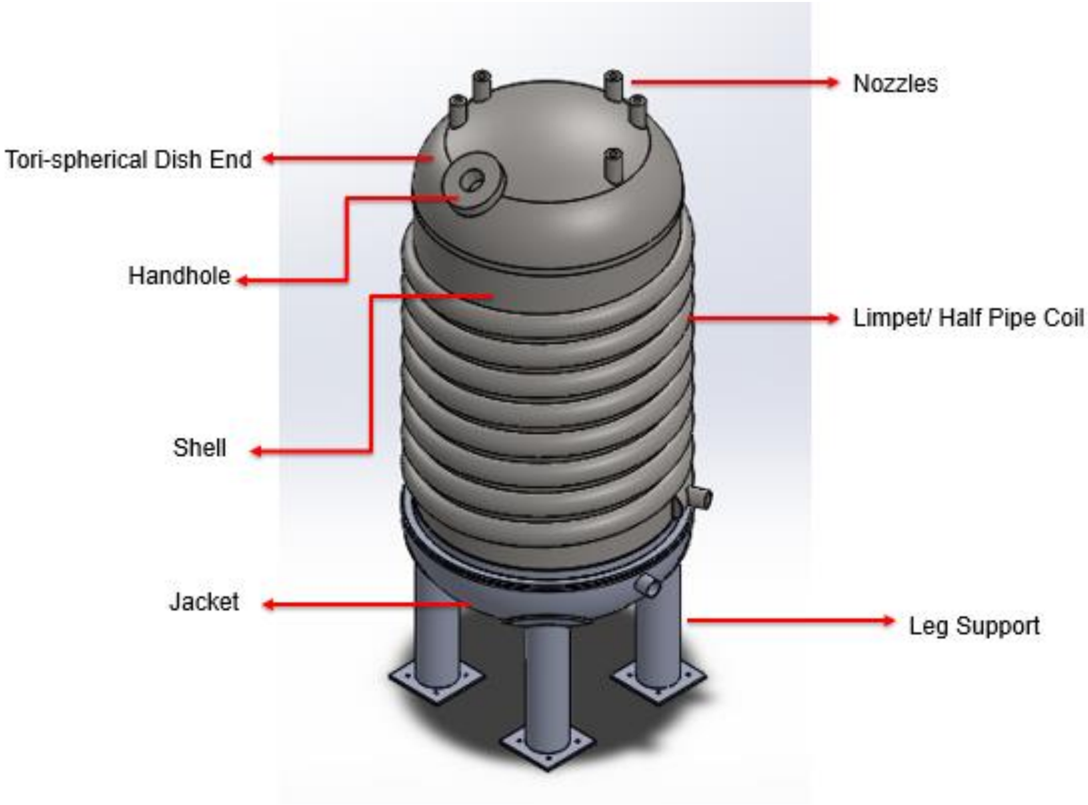

Figure 1.1: - Components of Pressure Vessel.

Pressure vessels are generally classified based on manufacturing material, based on geometric shape, based on installation methods, based on pressure bearing situation, based on wall thickness and based on technological processes. Figure 1.3 shows the classification of the pressure vessel.

Figure 1.2: - Types of the pressure vessel

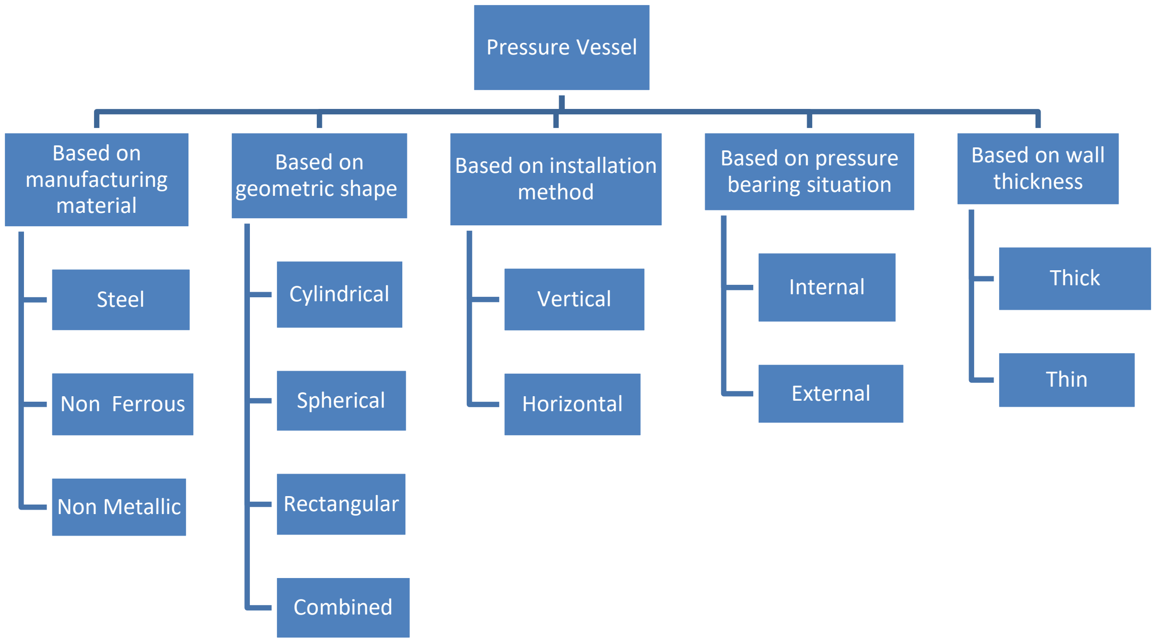

Figure 1.3: - Classification of Pressure Vessel

Pressure vessels are generally used as a storage vessel, Heat Exchangers, and Process Vessels.

- Storage Vessel: - These are the most prolific of all types of the pressure vessel. Depending upon the exact product to be stored they are manufactured using different materials.
- Heat Exchangers: - Heat exchanger is a device used to transfer heat from solid to liquid or between two or more liquids. They are typically used in refrigeration, air conditioning, petrochemical plants and power stations.
- Process Vessels: - Process vessels are components in which various processes like breaking down product, combining product are performed.

1.2 Composite Material

A structural composite is a material system consisting of two or more phases on a macroscopic scale, whose mechanical performance and properties are designed to be superior to those of the constituent materials acting independent [2]. One of the phases is usually discontinuous, stiffer, and stronger and is called the reinforcement, whereas the less stiff and weaker phase is continuous and is called the matrix [2]. The advantages of composite include High stiffness, High strength, Low density, Low thermal expansion and Design flexibility.

1.2.1 Matrices

The important function of matrices is to support fibers and transfer local stresses from one fiber to another. Four types of matrices used in composites are polymeric, metallic, ceramic, and carbon. Commonly used matrix among these types of matrices is polymeric, which can be thermosets or thermoplastic. The matrix determines maximum service temperature of a prepreg. Therefore, maximum service temperature is one of the key criteria for choosing the appropriate prepreg matrix. Table 1.1 below shows the operating range and common applications of different kinds of matrices[3].

Type of Matrix	**Operating Temperature**	**Application**
Epoxy	150 to 170°C	Aerospace, Sport, Marine, Automotive
Phenolic	65 to 105°C	Aerospace, Marine, Rail
Polyimide	-269 to 400°C	High-temperature components

Table 1.1: - Operating temperature and applications of Matrices

1.2.2 Reinforcements

The reinforcements/ fiber is the principal component of the composite material. It is major load carrying component and hence, occupy a large volume of the composite. Generally, fibers occupy 60% of the volume of composite material. The desirable characteristics of most reinforcing fibers are high strength, high stiffness, and relatively low density [2]. Most commonly used fibers are Glass fibers, Carbon fiber, Aramid (Kevlar) and Boron fibers.

Glass fibers have high strength and low stiffness as compared to other fibers. Therefore, glass fibers are used in low to medium performance composite applications. Glass fibers comparatively have low cost than other fibers. Glass fibers are sensitive to high temperatures. E-glass, S-glass, T- glass are commonly used in glass fiber materials. Some of the advantages of carbon fiber are high strength and high stiffness. But, the cost of carbon fiber is very high. A wide range of carbon fibers is available in the industry based on stiffness and strength. Kevlar or Aramid fiber has high tensile strength, low density, low compressive strength and High moisture absorption.

1.3 Motivation and Objective of Study

The focus of this study is to develop Finite Element Model to predict the performance of reactor pressure vessel metallic liner overwrapped with composite material. The pressure vessel should withstand the operating conditions. To predict the performance of reactor pressure vessel, composite shell and the top-dish end were designed and overwrapped over the metallic liner to withstand internal pressure (6.89 MPa) and operating temperature of 280°C. The pressure vessel was analyzed using given operating conditions to show the cause of failure for the composite Reactor Pressure Vessel using finite element simulations and by means of analytical calculations. Alternative designs and compatible material changes have been suggested in the later part of the sections.

Chapter 2: Literature Survey

In the previous chapter, component of the pressure vessel, classification, advantages and its applications were discussed. Composite material and its applications and advantages were also discussed in the previous chapter. In this chapter information from different papers and patents that was referred to complete this work has been presented.

Sumit V. Dubal and Dr. S.Y. Gajjal studied finite element analysis of reactor pressure vessel under different loading conditions. Appropriate stress value in cylindrical pressure vessel was calculated under different boundary conditions using finite element tool[4]. R. Kitching and H.H. Better carried out theoretical analysis to determine the distribution of stresses and radial deflection due to internal pressure in the limpet of a limpet coil pressure vessel [5]. According to Rao Yarrapragada K.S.S, et al., the performance of pressure vessel depends upon the weight of pressure vessel. The use of composite materials improves the performance of the vessel and offers a significant amount of material savings. Moreover, the stacking sequence is very crucial to the strength of the composite material [6]. Jianbing Hu developed Finite Element Model of composite cylinder considering various mechanical and thermal loading. Failure model based on Hashin's theory has been implemented to detect various failure modes of composite material [7]. According to Patil S, et al., Numerical inverse analysis is used to predict properties of the heat generating material by measuring the temperature at outer boundary. Accuracy and efficiency of the method are enhanced by using accurate sensitivity information by use of Semi-Analytical Complex Variable Method (CVSAM). Sensitivity information is beneficial in determining the reliability of the system. [13,14]. Q. G Wu, et al., performed stress and damage analysis on composite pressure vessel aluminum liner under internal pressure through numerical simulations. The variational wound angle and thickness of the head is reflected in finite element model in accordance with actual structure. A progressive damage model considering fiber tension and compression, matrix tension and compression are adopted for composite material [8]. Q. Zhang, et al., determined thermo-mechanical stresses in a multi-layered composite pressure vessel when the influence of its closed ends is considered. The analytical solution was derived for determining the stress distribution of multi-layered composite pressure vessel subjected to internal pressure and thermal load [9].

Chapter 3: Geometry and Boundary Condition

In this chapter, the study of geometry selected for the analysis has been presented. Various cases considering different materials, loading conditions, and boundary conditions have been discussed.

3.1 Geometry

The Reactor Pressure Vessel Consists of various components viz., Top Dish End, Cylindrical Shell, Bottom Dish End, Jacket Dish End, Limpet/Half Pipe Coil and Leg support. The CAD model of reactor pressure vessel was developed referring to as-built drawing and Half pipe/Limpet coil layout. The total height of the Reactor pressure vessel including the height of the nozzle and Leg Support is approximately 3000mm. The thickness of the Bottom and Top dish end as per design data is 75mm. The thickness of the Cylindrical Shell is 63mm and thickness of Jacket Dish end is 14 mm. The Inside radius of Reactor Pressure Vessel is 1525 mm. Fig 3.1 shows all the important dimensions of reactor pressure vessel along with all the components. Fig 3.2 shows the nozzle orientation on the various components of the pressure vessel. Fig 3.3 shows the layout of Half pipe/ Limpet coil on the shell.

Conventionally carbon steel, stainless steel, titanium and their alloys have been used to manufacture pressure vessels. Composite materials are suitable replacements for these conventionally used materials. To study the effect of composite material following cases have been discussed.

Case 1: - Carbon steel Model as per drawing and design data.

Case 2: - Carbon steel metallic liner of 40 mm overwrapped with Polyimide/ E-glass.

Case 3: - Carbon steel metallic liner of 30 mm overwrapped with a combination of Polyimide/E-glass, Polyimide/ S-glass, and Polyimide/carbon fiber.

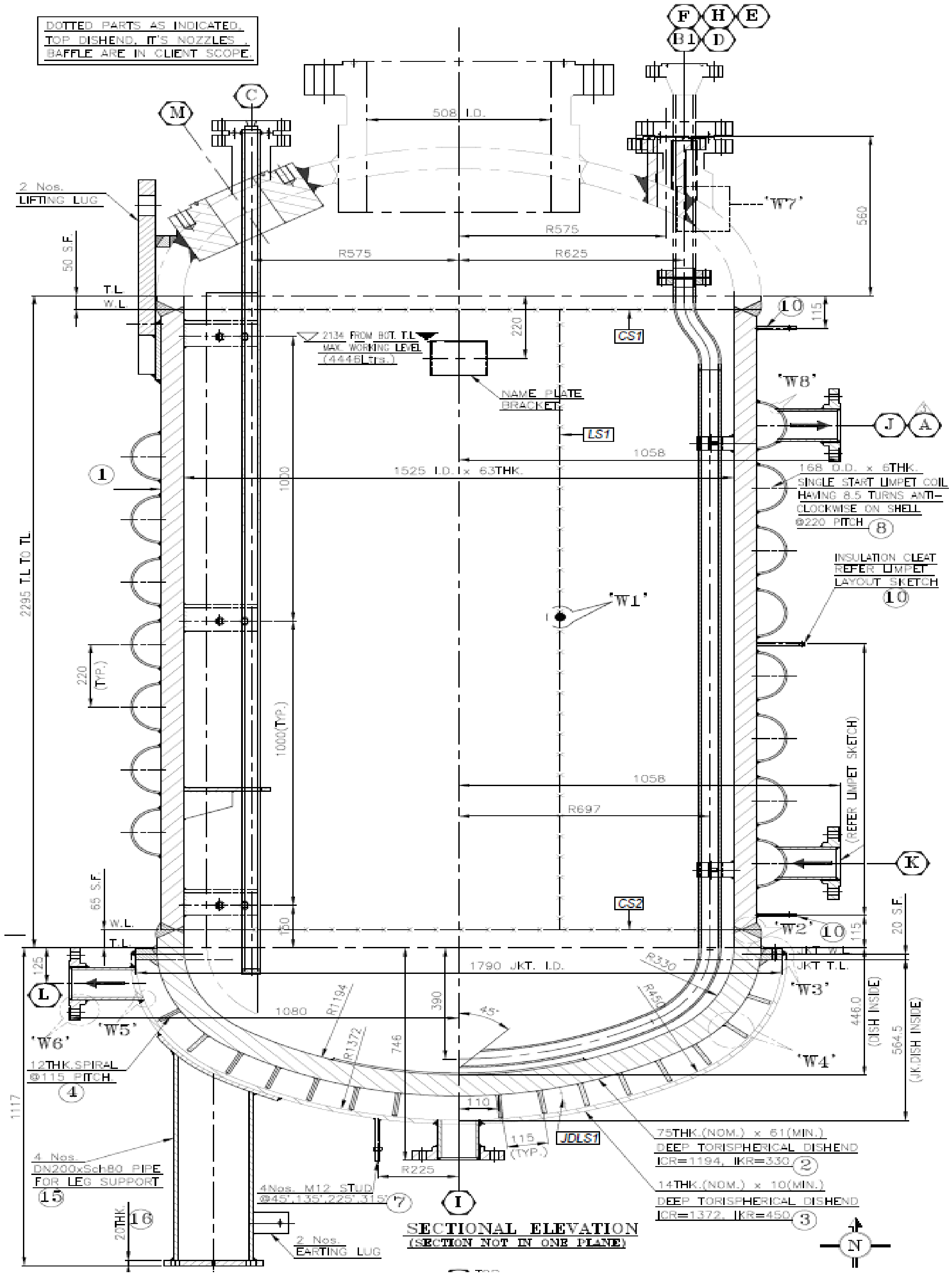

Figure 3.1: - As built drawing of Reactor Pressure Vessel.

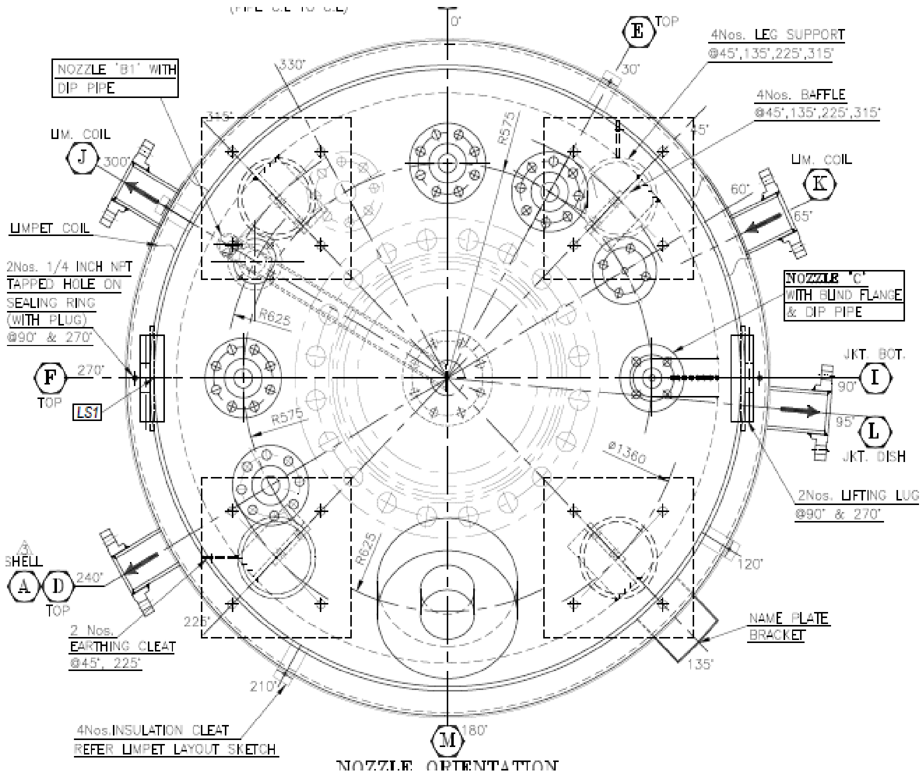

Figure 3.2: - Nozzle Orientation

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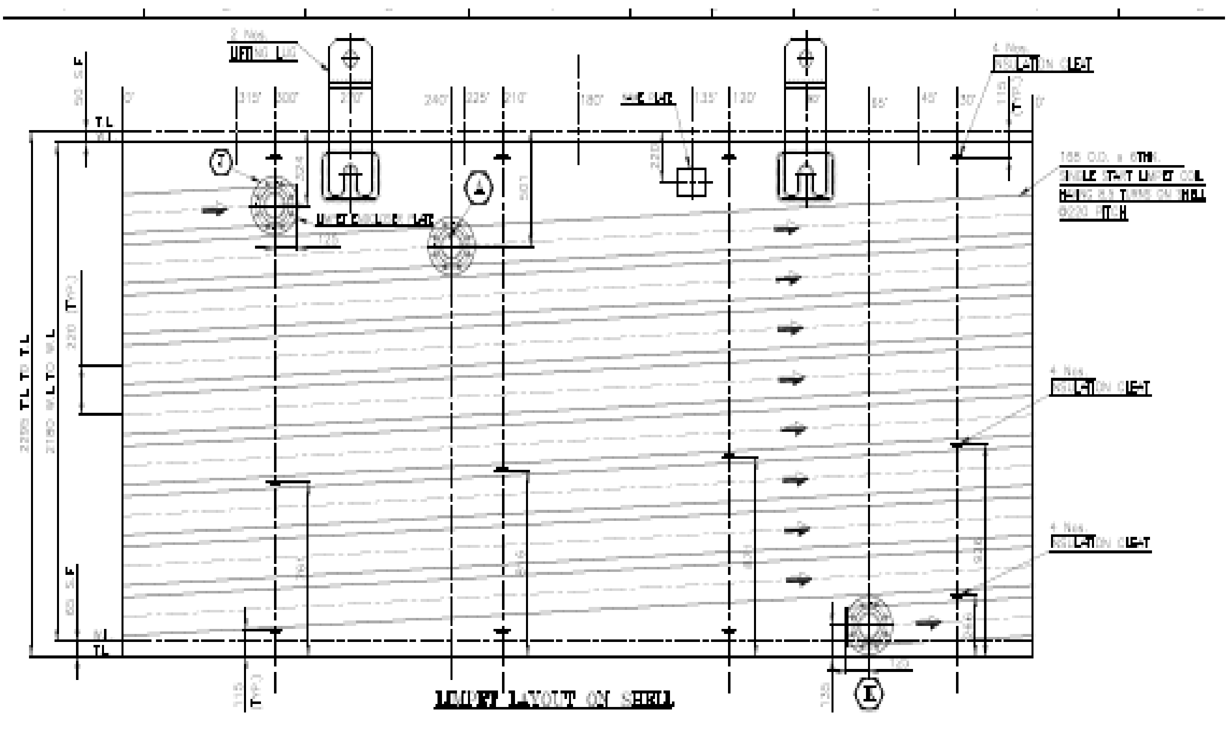


Figure 3.3: - Half Pipe/ Limpet coil layout on the shell

3.2 Meshing

The CAD model was meshed using Solid brick elements. Solid 186 is a higher-order 3-D 20 nodes solid element that exhibits quadratic displacement behavior. The element supports plasticity, hyperelasticity, creep, stress stiffening, large deflection, and large strain capabilities [15]. It also has mixed formulation capability for simulating deformations of nearly incompressible elastoplastic materials, and fully incompressible hyperplastic materials [15]. Meshing was done using Body sizing tool using Tetrahedron elements. Table 3.1 shows mesh size, number of nodes and number of elements in each case. Fig 3.4 shows Meshed CAD model in each case.

	Body Size	**Number of Nodes**	**Number of elements**
CASE 1	80 mm	98974	45468
CASE 2	50 mm	350117	292093
CASE 3	50 mm	460983	379574

Table 3.1: - Mesh Data

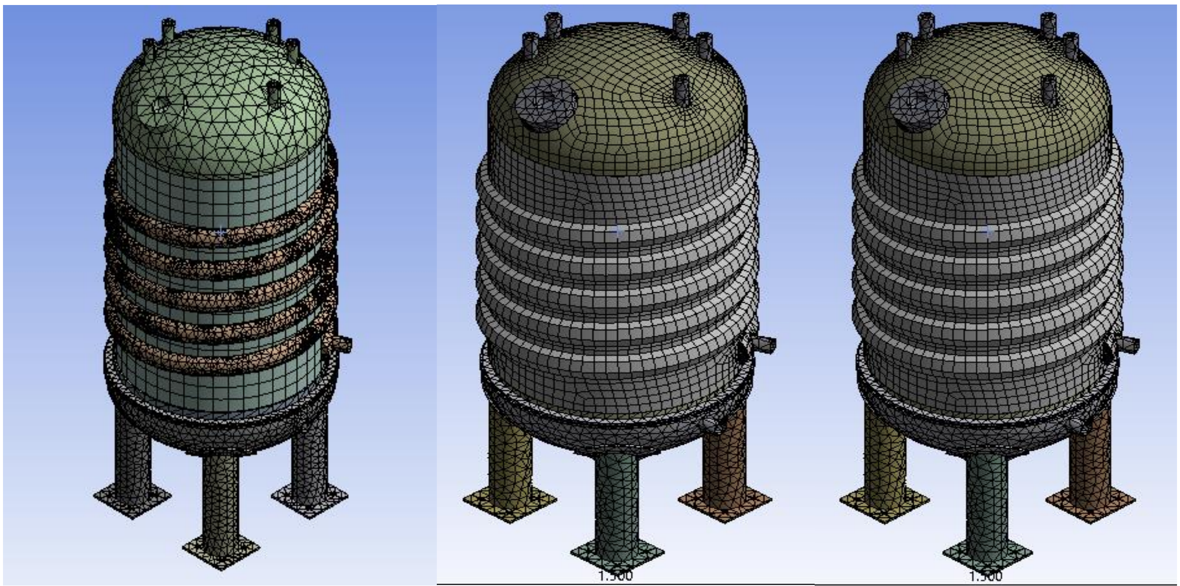


Figure 3.4: - Meshed CAD model

3.3 Boundary Conditions

The boundary conditions in each of the cases discussed in the earlier discussion are based on various loading conditions under which pressure vessel operates. These loading conditions are discussed below. Fig 3.5 shows the application of various boundary conditions as per various load cases.

Load Case 1: Internal Pressure.

Under this Load case, Reactor Pressure Vessel is subjected to design internal pressure of 70Kgf/cm^2 on shell side and 0.7Kgf/cm^2 on Jacket/Limpet coil side. Considering the effect of static head and design pressure, total internal pressure on each component of pressure vessel will be as shown in Table 3.2.

Component	**Internal Pressure + Static Head (MPa)**
Top Dish End	6.89
Shell	7.00
Bottom Dish End	7.02
Jacket Dish End	0.68
Limpet/ Half-Pipe Coil	0.68

Table 3.2: - Internal pressure on each component of the pressure vessel

Load Case 2: Steady-state Thermal conduction

As per this loading condition, the pressure vessel is subjected to operating temperature of 230°C on shell side and 280°C on Jacket/ Limpet coil side. Convection boundary condition is applied to the outer surface of the pressure vessel. Convection film coefficient is taken as 22 W/m^2 °C.

Load Case 3: External Pressure + Empty Weight.

In this loading condition, the assembly is analyzed for empty weight i.e. the weight of the vessel. This analysis is done to verify if the vessel is stable under empty condition. To do this analysis, supports are applied, and weight is considered by applying standard earth gravity of 9.80 m/s^2 and External Pressure equivalent to atmospheric pressure is applied.

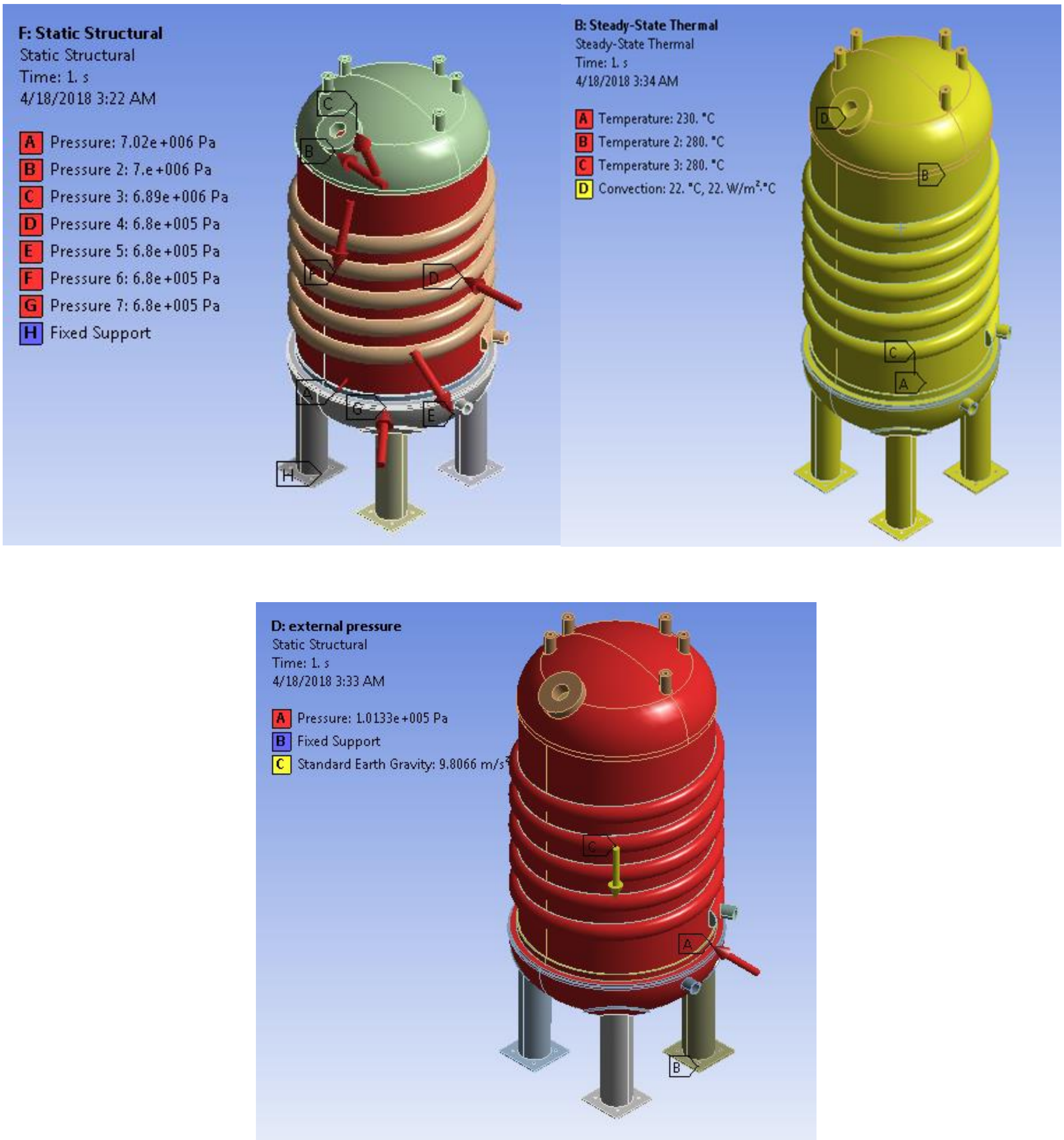


Figure 3.5: - Boundary conditions as per various Load Case.

Chapter 4: Material Properties

The focus of this study is to develop Finite Element Model to predict the performance of reactor pressure vessel metallic liner overwrapped with composite material. The metallic liner is manufactured using carbon steel (SA 516 Gr 70). The composite materials used in the study are Polyimide/ E-glass, Polyimide/ S-glass, Polyimide/YS90A60S carbon fiber. The mechanical and thermal properties of these materials are discussed in this chapter.

4.1 Properties of carbon steel.

Carbon steel is most commonly used as the material in the manufacturing of Pressure vessels. Carbon steel is the steel which contains carbon as their alloying element. Based on the content of the carbon it is categorized as Low-carbon (also known as mild steel) contains 0.3% carbon. Medium carbon steel and High carbon steel contains up to 1.5% and 2.1% of carbon respectively. Mechanical and Thermal properties of carbon steel (SA516 Gr70) are given in Table 4.1[16]

Property	**Carbon steel (SA516 Gr70)**
Density (Kg/m^3)	7800
Young’s Modulus (GPa)	200
Poisson’s ratio	0.29
Ultimate Tensile Strength (MPa)	450
Yield Strength(MPa)	260
The thermal coefficient of expansion (°C^{-1})	12e-6
Specific Heat (J/Kg. °C)	470

Table 4.1: - Mechanical and Thermal Properties of carbon steel (SA516 Gr70)

4.2 Properties of Composite Materials.

The composite materials used in this study have polyimide matrix because polyimide films retain their physical properties over a wide temperature range of as low as -269°C (-52°F) and as high as 400°C (752°F) [10]. Due to this unique property of polyimide matrix, it is used in high-temperature components. The fibers E-glass, S-glass, and Carbon fiber are used as reinforcements. Important properties to be considered for structural and thermal analysis are Longitudinal and Transverse Modulus, In-plane shear modulus, density, Poisson's ratio, the coefficient of thermal expansion and Thermal conductivity. The mechanical and thermal properties of Kapton polyimide film are given in Table 4.2 [10].

Ultimate Tensile Strength (MPa)	231
Tensile Modulus (GPa)	2.5
Thermal coefficient of linear expansion ppm/°C	20
Coefficient of Thermal Conductivity (W/mK)	0.12
Specific Heat (J/gK)	1.09

Table 4.2: - Properties of Kapton Polyimide film

Based on the study of mechanical properties of polyimide and epoxy, it is observed that mechanical properties of polyimide and epoxy resin are comparable. Thus, Epoxy resin is used in the analysis. Epoxy resins have high strength and modulus, low shrinkage, brilliant adhesive properties. The properties of Epoxy/E-glass, Epoxy/S-glass, and Epoxy/ YS90A60S carbon fiber are as given in Table 4.3 [11]. These properties are calculated using MATHCAD program. Mechanical properties of Epoxy/E- glass UD, Epoxy/S- glass UD are taken as defined in ANSYS.

Property	Epoxy/E-glass	Epoxy/S-glass	Epoxy/ YS90A60S carbon fibre
Longitudinal Modulus (GPa), E_1	45	50	529.2
Transverse Modulus (GPa), E_2	10	8	8.74
Major Poisson's ratio, υ_{12}	0.3	0.3	0.324
Minor Poisson's ratio, υ_{23}	0.4	0.4	0.421
In-plane shear modulus (GPa), G_{12}	5	5	3.687
Out-Plane shear modulus (GPa), G_{23}	3.846	3.846	3.076
In-plane coefficient of thermal expansion, α_1 (ppm/°C)	7	7.1	-0.99
Out-plane coefficient of thermal expansion, α_2 (ppm/°C)	27	30	44.8
Density (Kg/m^3)	2000	2000	1510

Table 4.2: - Mechanical properties of composite materials.

Chapter 5: Finite Element Analysis

Finite Element Analysis procedure is divided into three steps: Pre-Processing, Solution, and Post-Processing. In this study Finite element analysis is carried out for each of the case discussed in Chapter 3 and results are compared. Finite element Analysis is carried out using ANSYS 17.2. Each of the cases is analyzed under the effect of internal pressure, temperature and external pressure. Boundary conditions are applied as discussed in Chapter 3. Material properties discussed in Chapter 4 are entered in pre-processing.

- Pre-Processing- In this step of the analysis, input data for analysis is given. Model is created, Mesh and Element type and material data are defined.
- Solution: - In this step, boundary conditions in terms of loads, supports are given and the solution is obtained
- Post-Processing: - In this step, results are viewed and required results are evaluated and studied.

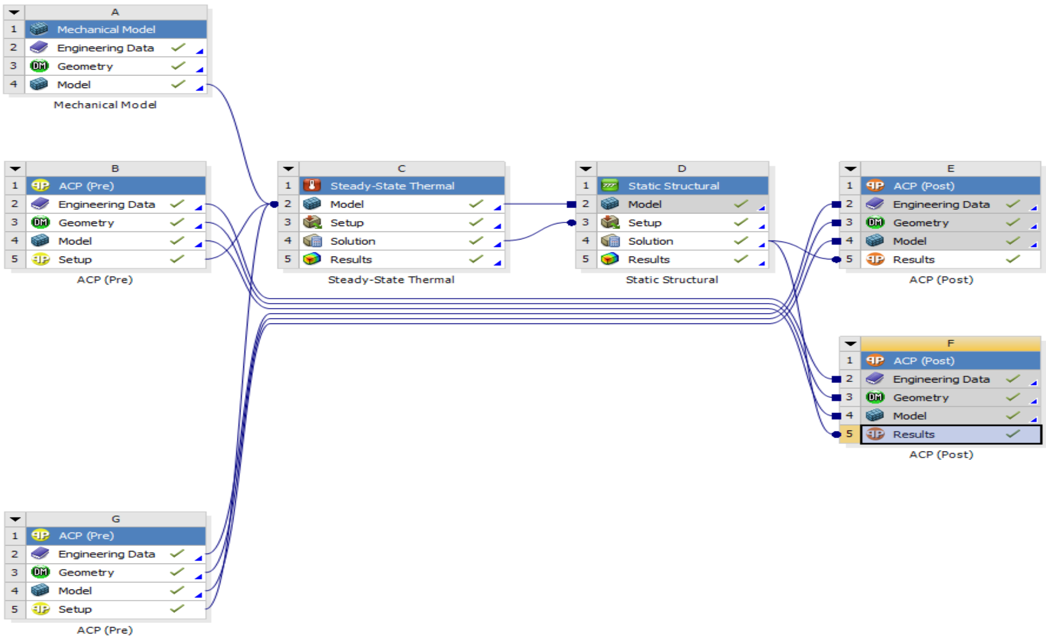


Figure 5.1: - ANSYS set up

ANSYS composite Pre-Post is used for the analysis using the composite material. Figure 5.1 shows the set up for ACP pre-post. A surface model is imported to ACP (pre) and ply modeling is done to create a solid model. To create a ply model, stack up sequence is defined. In a pressure vessel, circumferential stress is twice that of axial stress. Therefore, stack up is created along helical, axial and hoop direction. Helical and axial layers bear axial loading and Hoop layers carry circumferential loading [7]. Therefore, stacking sequence considered for the analysis in all the cases is [90/90/0/0/90/90] and [45/-45]. In case 2, the metallic liner of 40mm is overwrapped with Polyimide/ E-glass ply of 0.4 mm thickness containing 56 number of plies. In case 3, the metallic liner of 30mm is overwrapped with a combination of polyimide/ E-glass, polyimide/S-glass and polyimide/carbon fiber plies of 0.4 mm thickness, containing 76 number of plies. Figure 5.2 shows the direction of 0° ply and direction of thickness. Figure 5.3 shows solid model for shell and top dish.

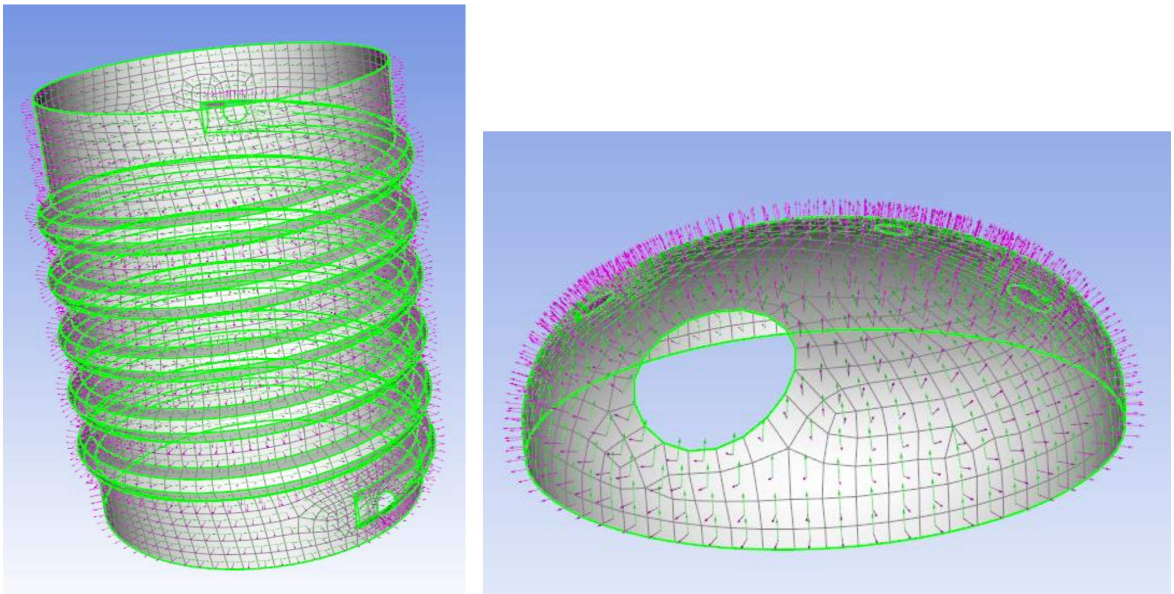
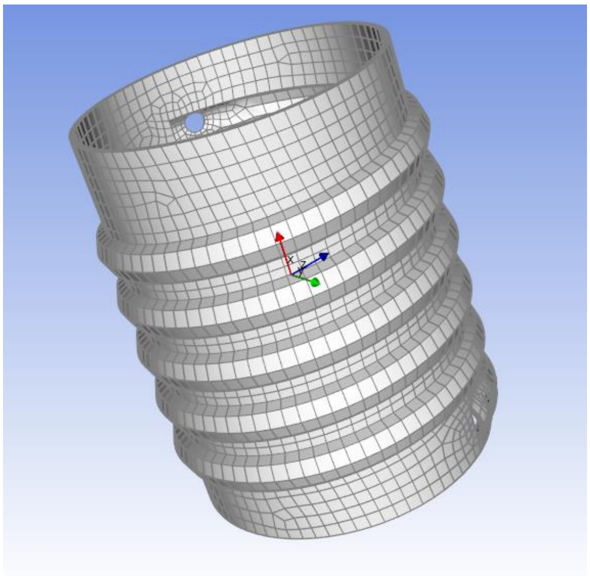


Figure 5.2: - Direction of plies and thickness



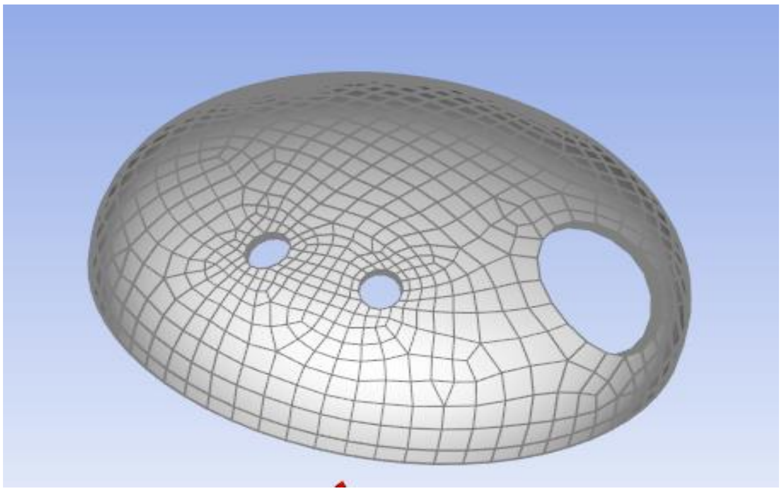


Figure 5.3: - Solid Models

These solid models are transferred to static structural analysis and predefined boundary conditions are applied. Results like Maximum Normal stress, Factor of safety and Temperature are viewed in ACP-post.

Chapter 6: Results

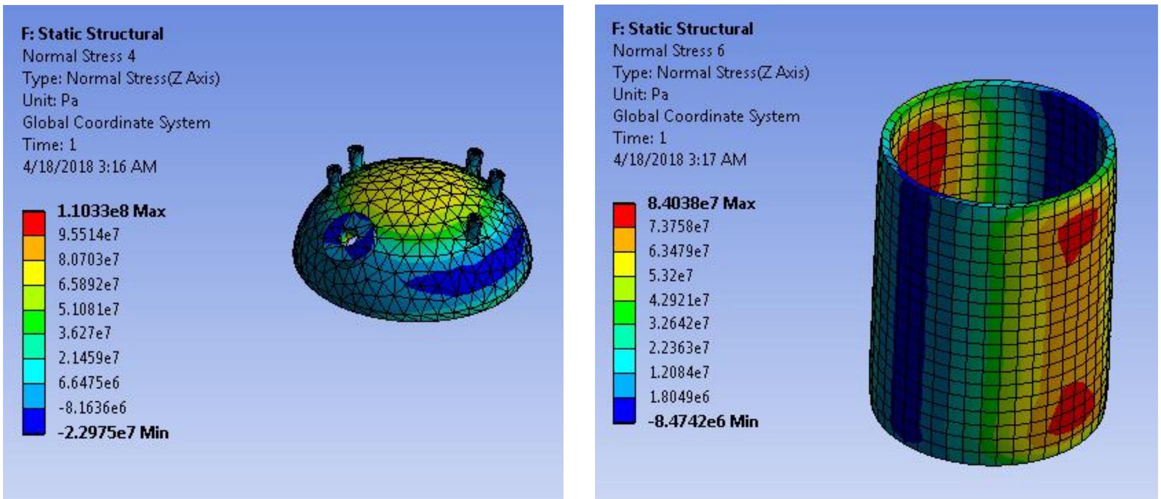
The simulations are run for all the cases discussed earlier according to material data, boundary conditions, and loading conditions. The results of each case for different loading conditions are discussed in this chapter.

6.1 Carbon Steel model as per design data

This model is analyzed under internal pressure, external pressure, empty weight, and thermal load. According to American Society of Mechanical Engineers (ASME) design criteria, the allowable stress in each of the component should not exceed 138 MPa. The results in each of the load cases are as discussed below.

6.1.1 Carbon steel model under Internal Pressure.

Results for maximum normal stress under internal pressure in each component of pressure vessel viz., Top dish end, Shell, Bottom dish end, jacket dish end and Limpet/half pipe coil are shown in Figure 6.1. Maximum stress in each of the component does not exceed 138 MPa. Hence, Design of pressure vessel is safe.



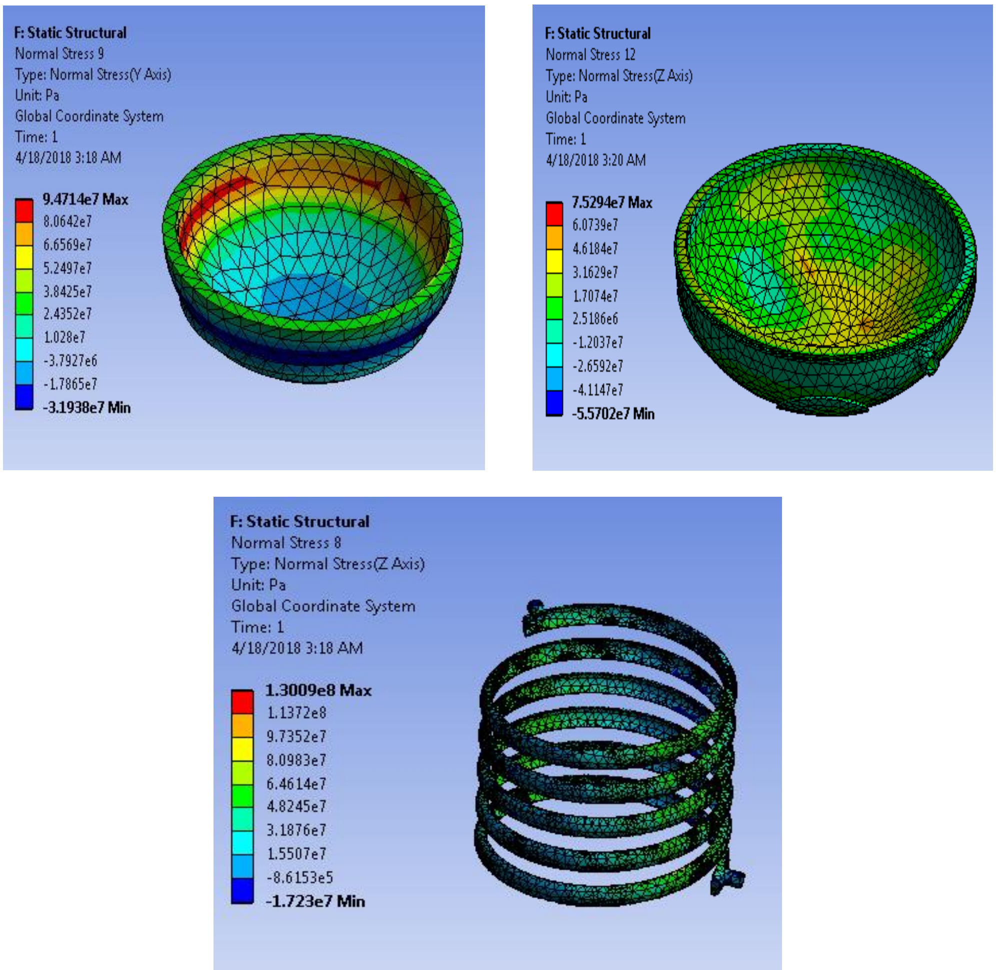


Figure 6.1: - Normal stress for case 1 under Internal pressure.

6.1.2 Carbon steel model under External Pressure and Empty Weight.

Results for maximum displacement and maximum equivalent stress under a combination of external pressure and empty weight in each component of pressure vessel are shown in Figure 6.2. Maximum equivalent stress on a pressure vessel is 20.92 MPa and displacement is 0.3mm.

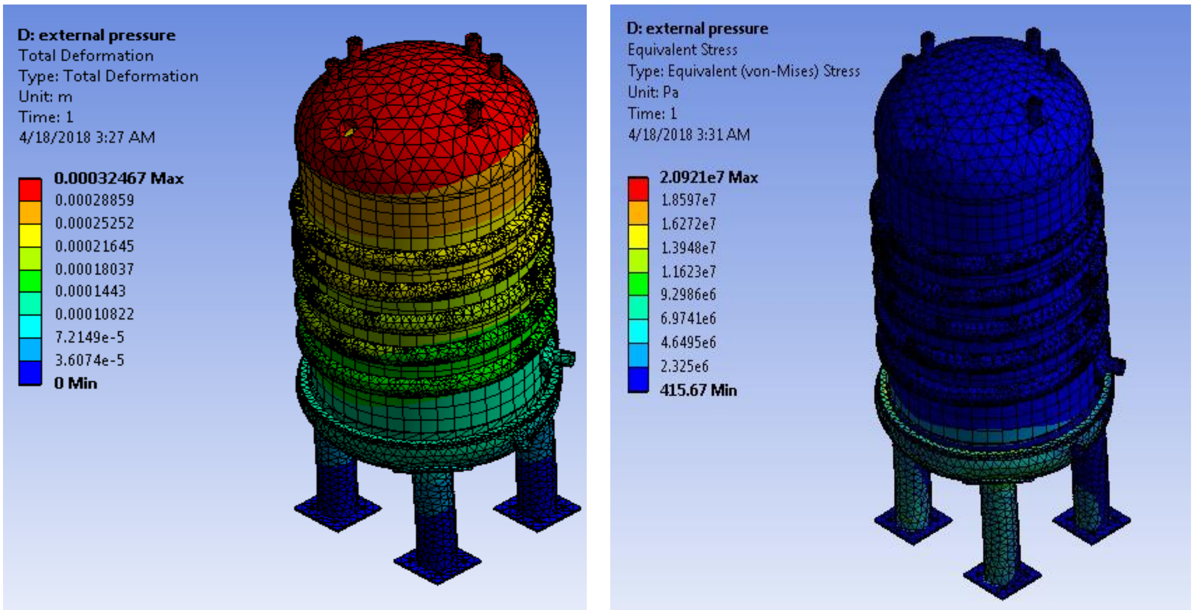
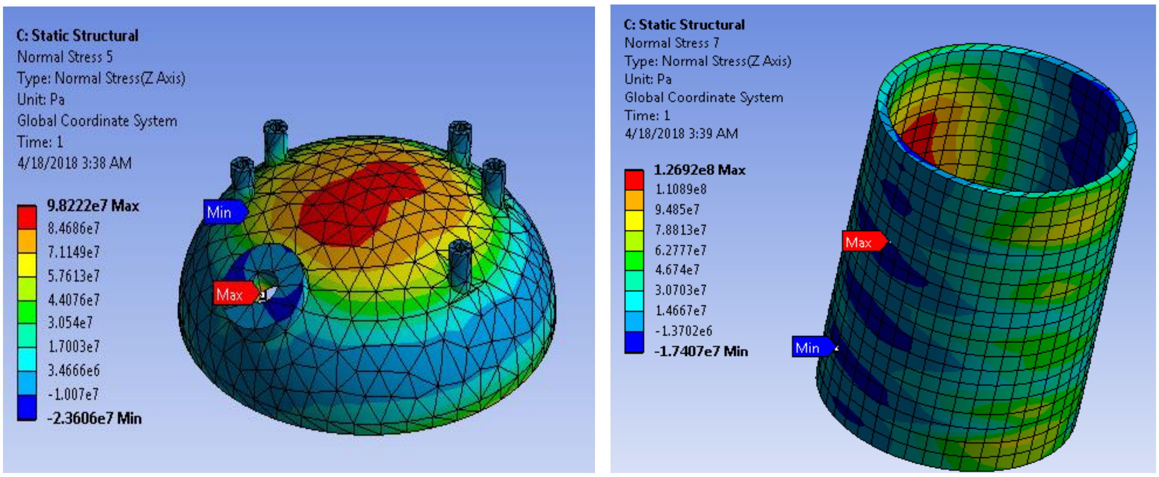


Figure 6.2: - Equivalent stress and deformation

6.1.3 Carbon steel model under Thermal load and Internal pressure.

Results for displacement, maximum equivalent stress under a combination of internal pressure and thermal load in each component of the pressure vessel (top dish end, shell, bottom dish end, jacket dish end, limpet coil) are shown in Figure 6.3.



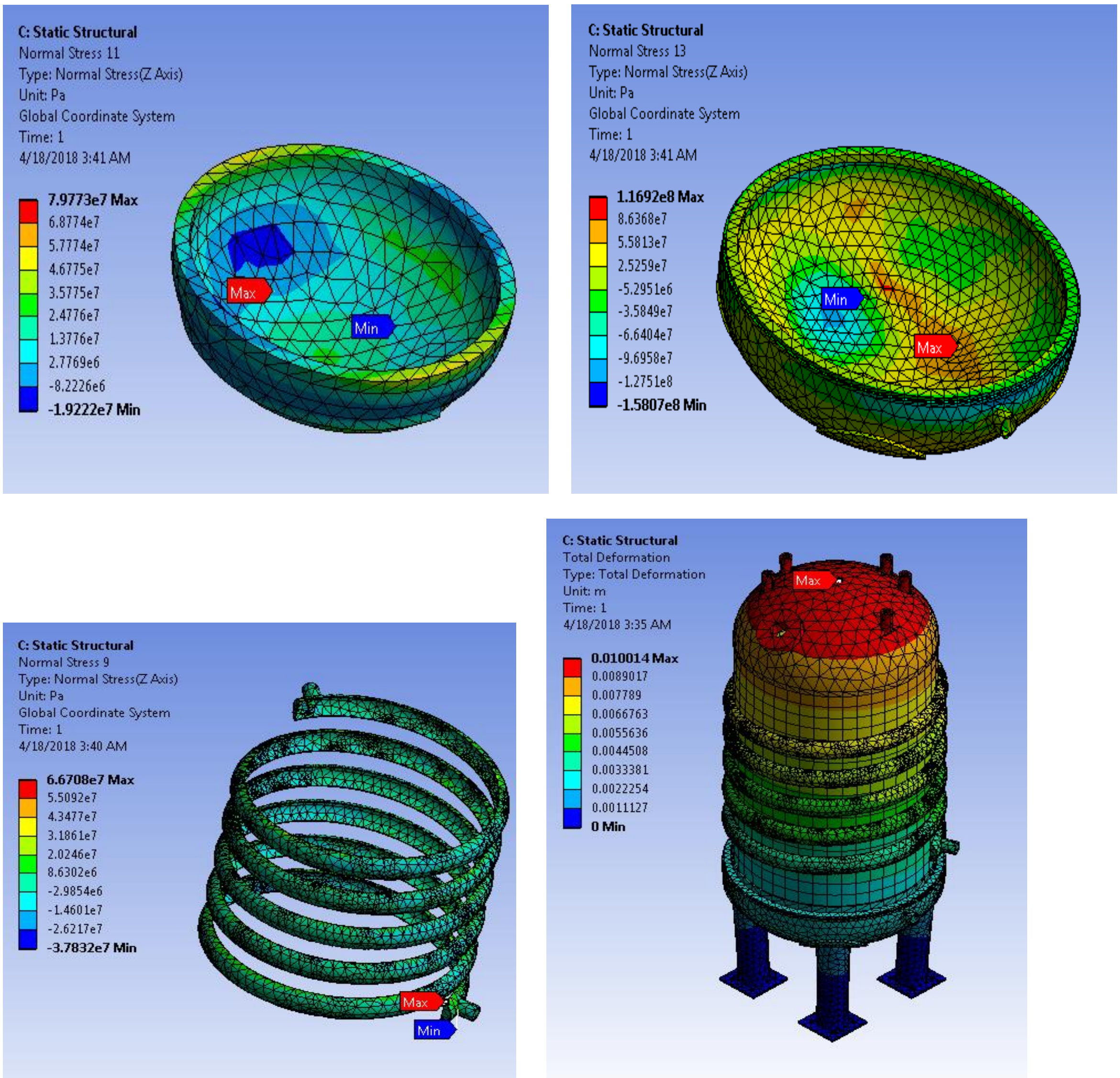


Figure 6.3: - Carbon steel model under Thermal load and Internal pressure

6.2 Carbon steel metallic liner (40mm) overwrapped with composite.

In this case, the metallic liner of 40 mm is overwrapped with polyimide/ E-glass composite material. This model is analyzed under internal pressure, external pressure, empty weight. The results in each of the load cases are as discussed below.

6.2.1 Case 2 under Internal pressure.

In this loading case, the pressure vessel is analyzed under the effect of internal pressure and stresses on the metallic liner and all the plies are studied. Figure 6.4 shows the maximum normal stress on the metallic liner. Figure 6.5 shows the stresses on inner and outer ply of the shell and top dish. Figure 6.6 shows a graphical representation of the variation of stresses along several plies due to internal pressure.

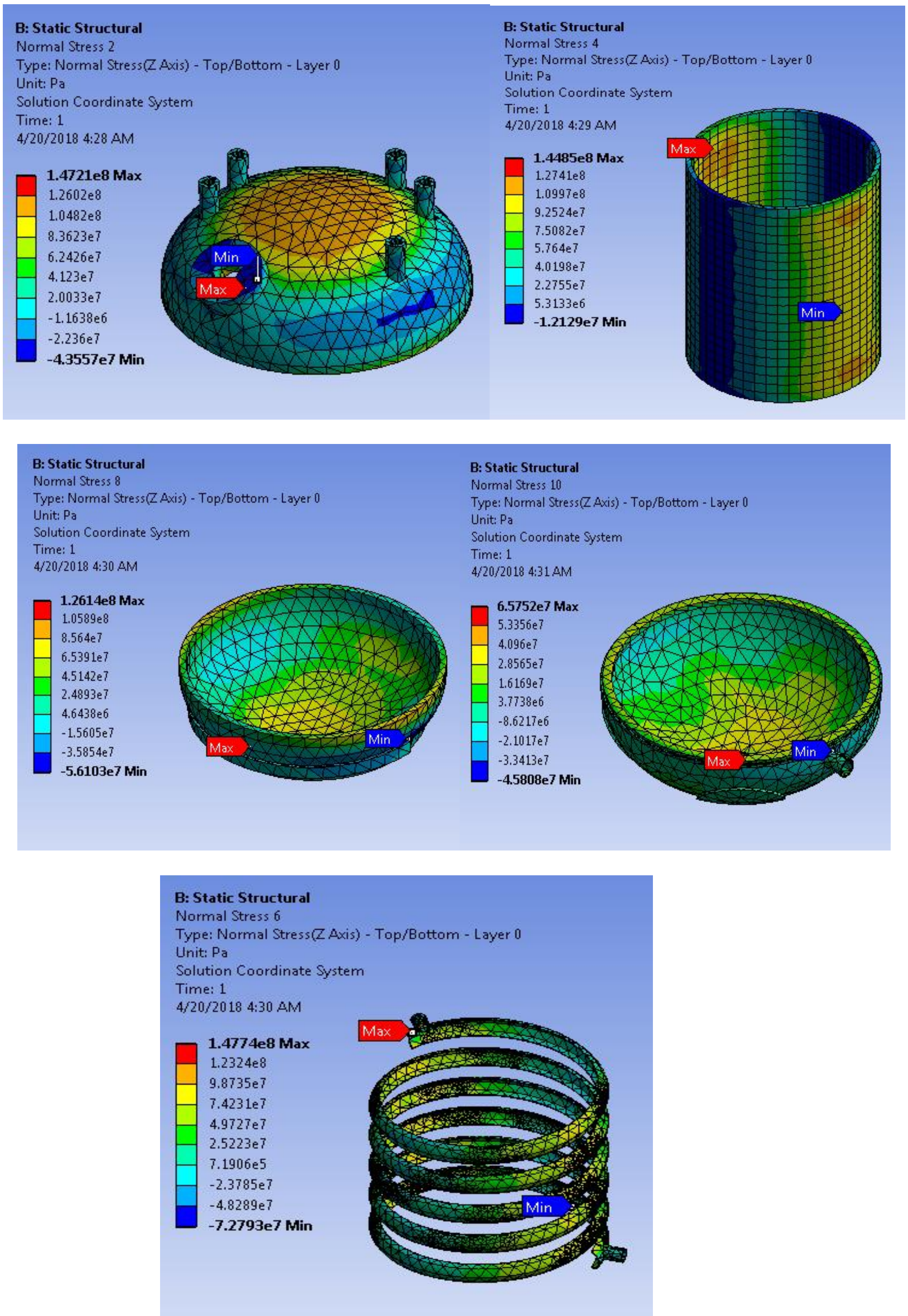


Figure 6.4: - Normal stress for Case 2 under internal pressure.

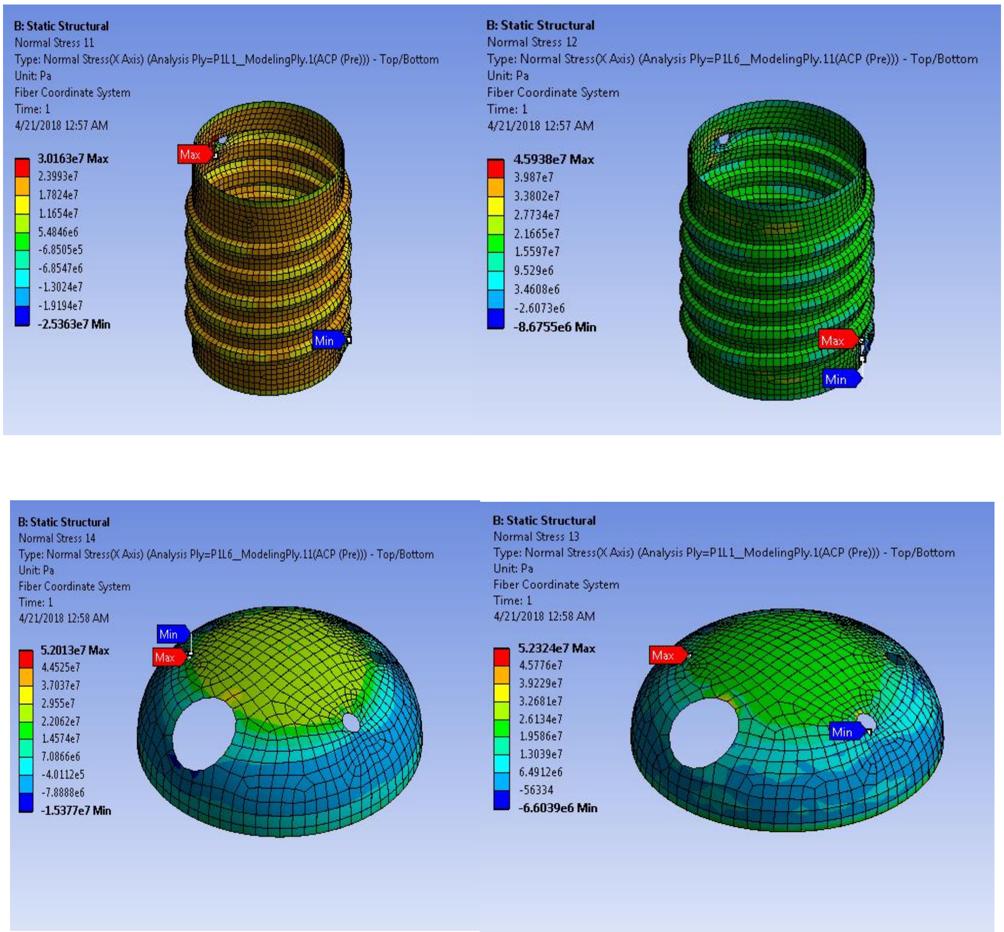


Figure 6.5: - Stress on the inner and outer ply of composite shell and top dish.

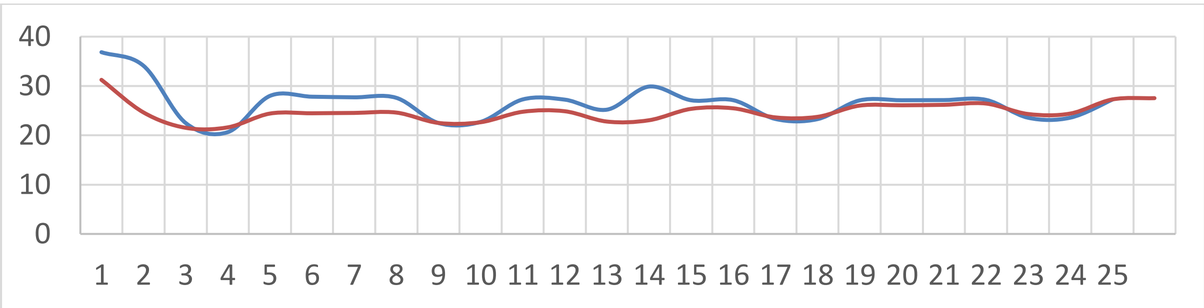


Figure 6.6: - Variation of stress due to internal pressure in various plies.

6.2.2 Case-2 under external pressure and Empty weight.

In this loading case, the pressure vessel is analyzed under the effect of external pressure and empty weight. Stresses on the metallic liner and all the plies are studied. Figure 6.7 shows the maximum equivalent stress on the metallic liner. Figure 6.8 shows the stresses on inner and outer plies of the shell and top dish. Figure 6.9 shows a graphical representation of the variation of stresses along several plies due to external pressure.

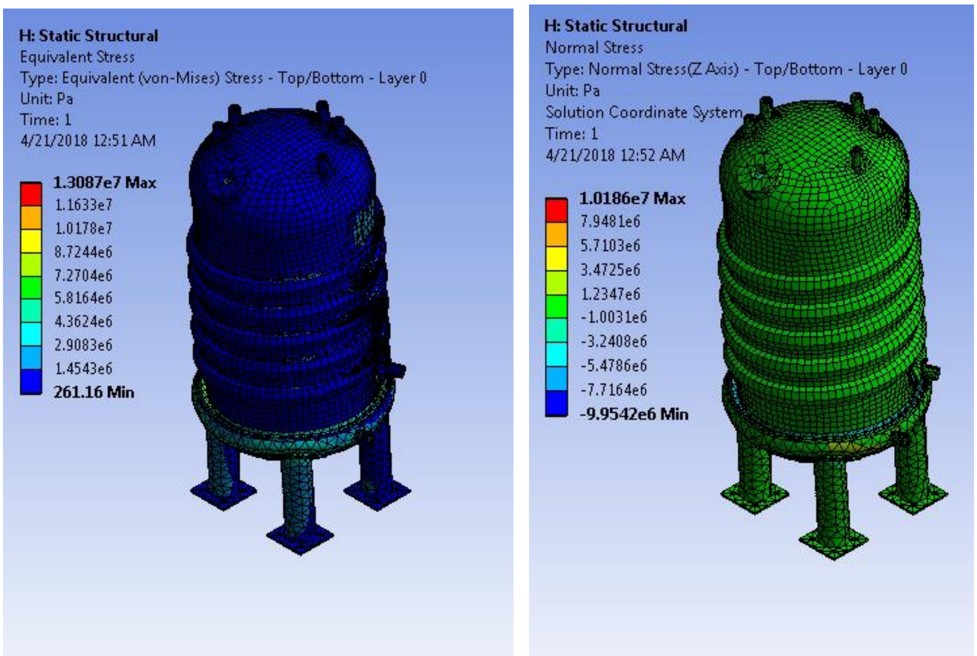
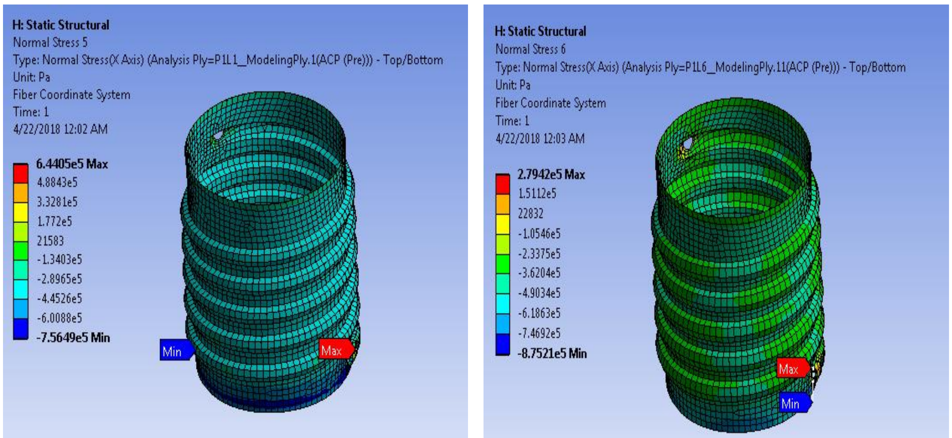


Figure 6.7: - Maximum Equivalent stress on metallic liner under external pressure.



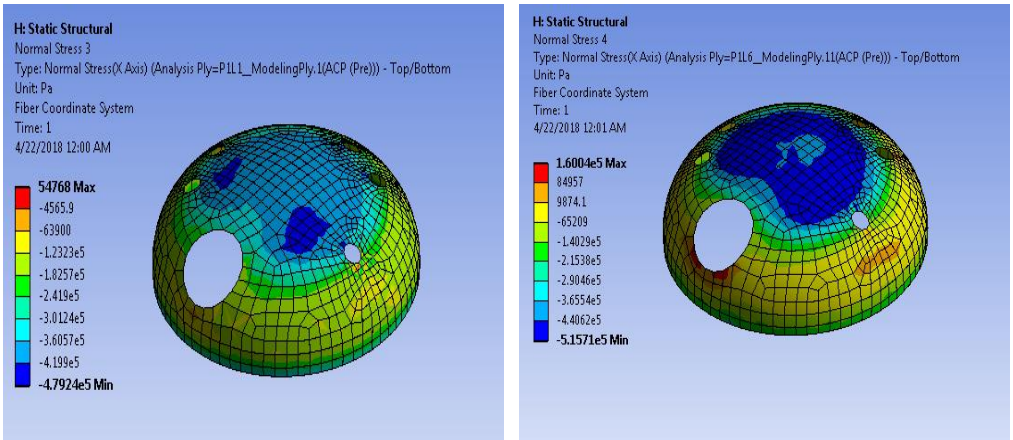


Figure 6.8: - Stress on the inner and outer ply of composite shell and top dish

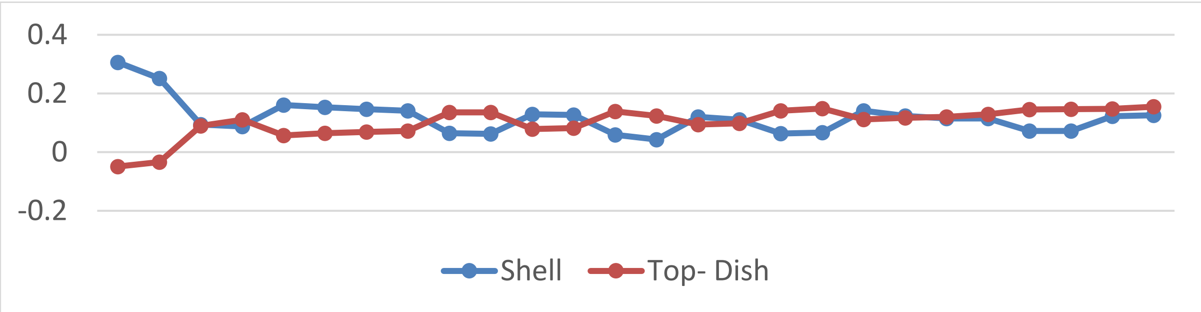


Figure 6.9: - Variation of stress in various plies.

6.2.3 Case-2 under steady state thermal conduction

In this analysis, the pressure vessel is analyzed for the thermal condition. Operating temperature condition is applied on the shell side and limpet/jacket side. The temperature inside the shell is 230°C and temperature inside jacket/limpet is 280°C. The temperature on the outer surface is calculated after the analysis. Figure 6.10 shows temperature on the inner and outer ply of composite shell and top dish. Figure 6.11 shows a variation of temperature in all the plies.

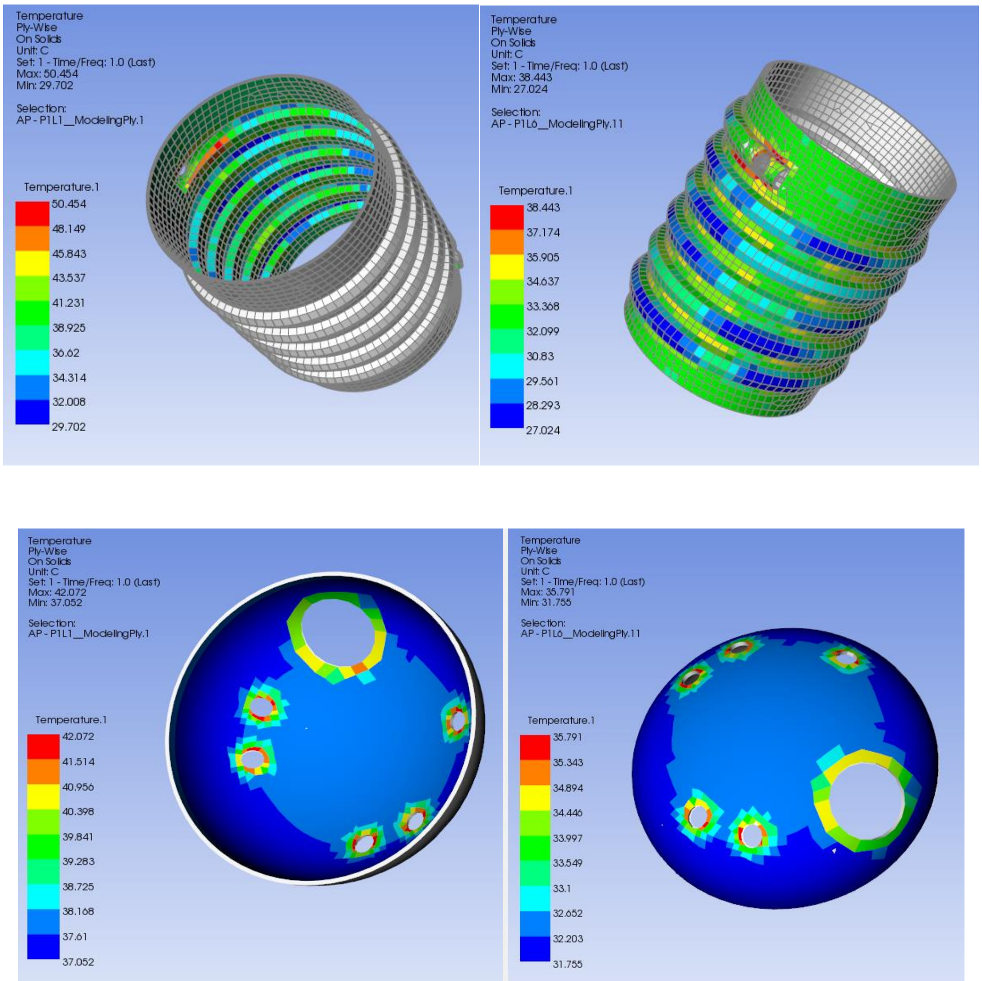


Figure 6.10: - Temperature distribution on inner and outer ply.

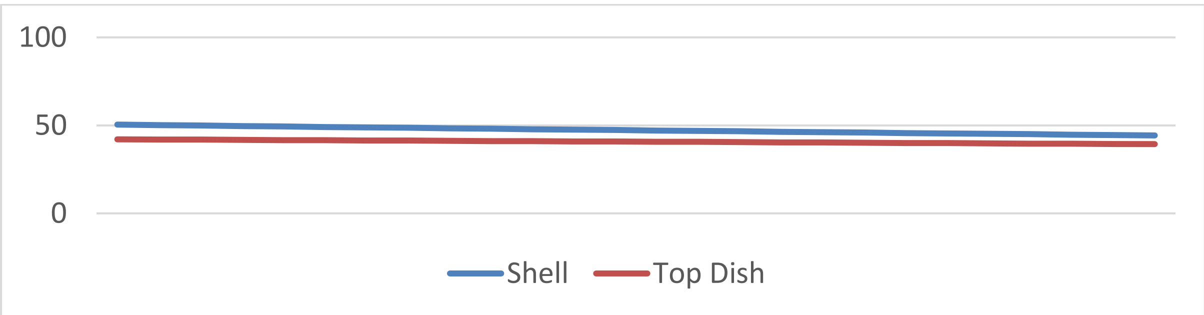


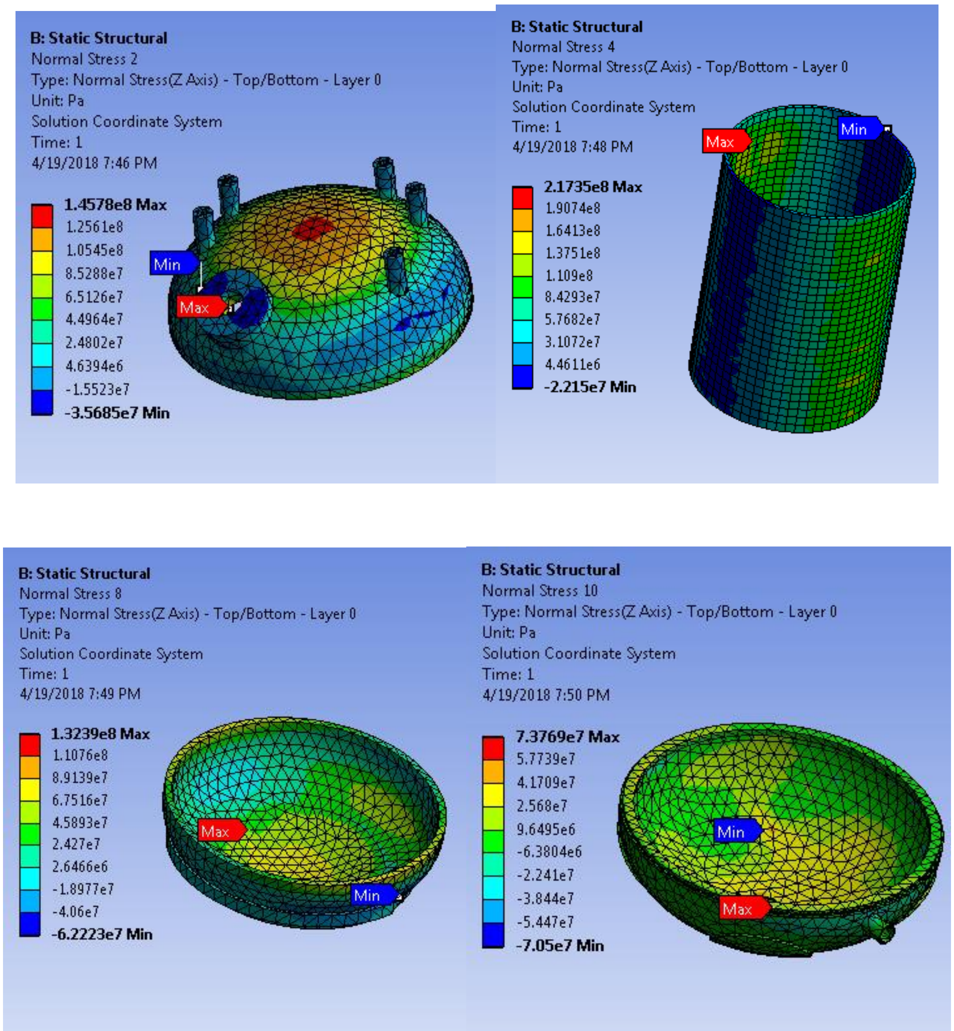
Figure 6.11: - Variation of temperature in plies

6.3 Metallic liner (30mm) overwrapped with composite material.

In this case, the metallic liner of 30 mm is overwrapped with polyimide/ E-glass composite material. This model is analyzed under internal pressure, external pressure, empty weight. The results in each of the load cases are as discussed below.

6.3.1 Case 3 under Internal pressure.

In this loading case, the pressure vessel is analyzed under the effect of internal pressure and stresses on the metallic liner and all the plies are studied. Figure 6.12 shows the maximum normal stress on the metallic liner. Figure 6.13 shows the stresses on inner and outer plies of the shell and top dish. Figure 6.14 shows a graphical representation of the variation of stresses along several plies due to internal pressure.



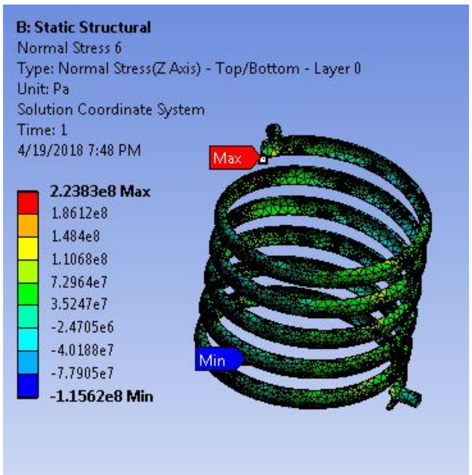


Figure 6.12: - Maximum normal stress on the metallic liner (30mm)

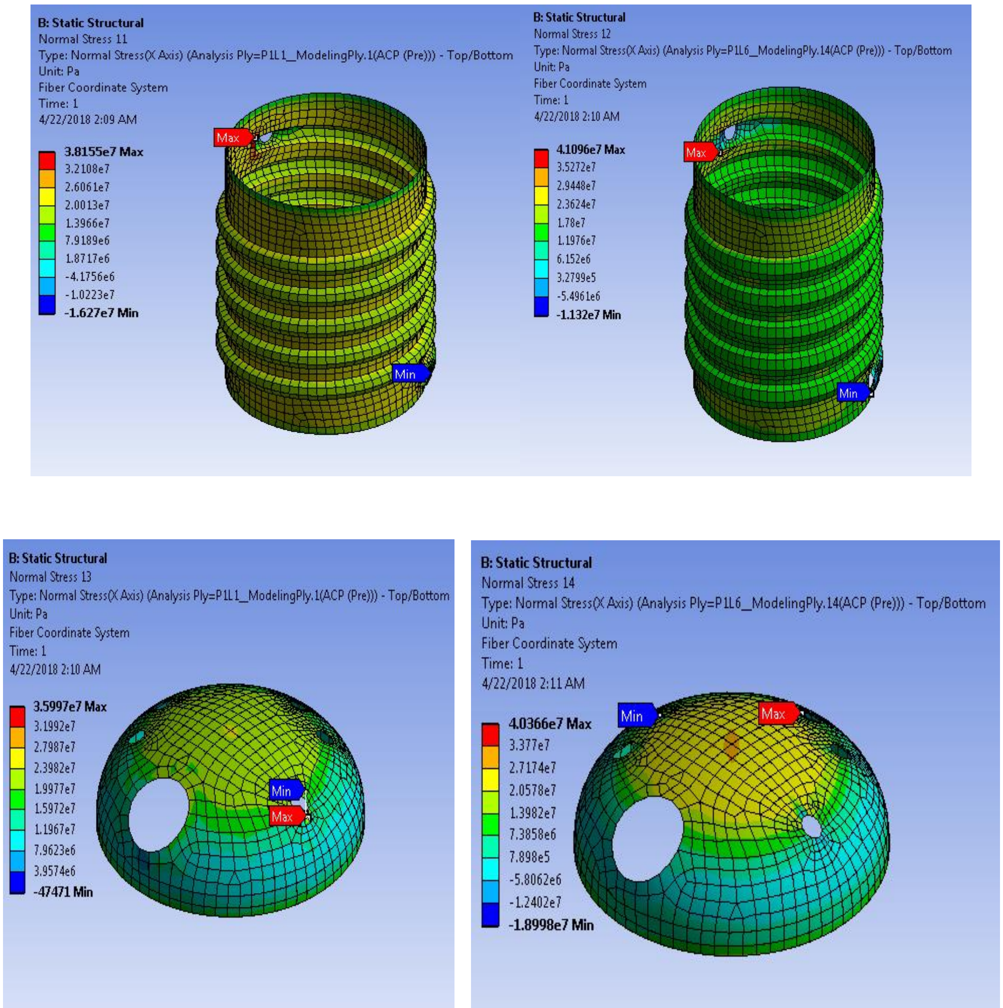


Figure 6.13: - Stresses on inner and outer plies of the shell and top dish.

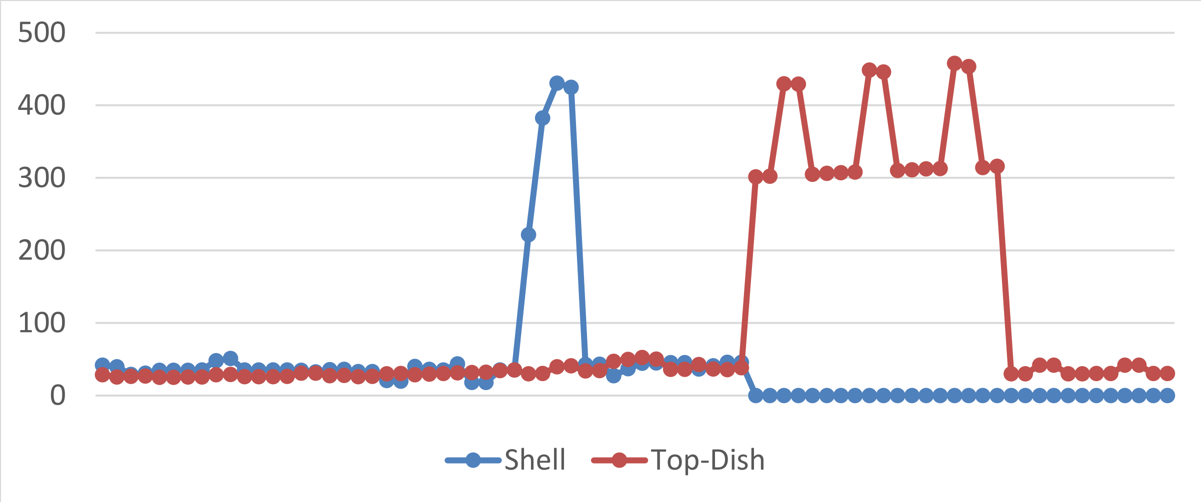


Figure 6.14: - Variation of stresses along several plies due to internal pressure.

6.3.2 Case-3 under external pressure and Empty weight.

In this loading case, the pressure vessel is analyzed under the effect of external pressure and empty weight. Stresses on the metallic liner and all the plies are studied. Figure 6.15 shows the maximum equivalent stress on the metallic liner. Figure 6.16 shows the stresses on inner and outer plies of the shell and top dish. Figure 6.17 shows a graphical representation of the variation of stresses along a number of plies due to external pressure.

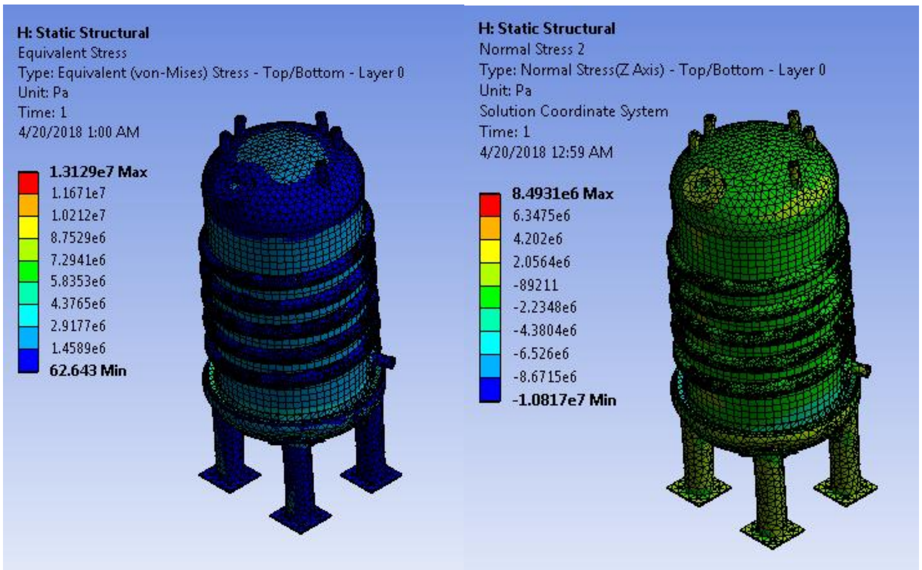


Figure 6.15: - Maximum equivalent stress on the metallic liner (30mm)

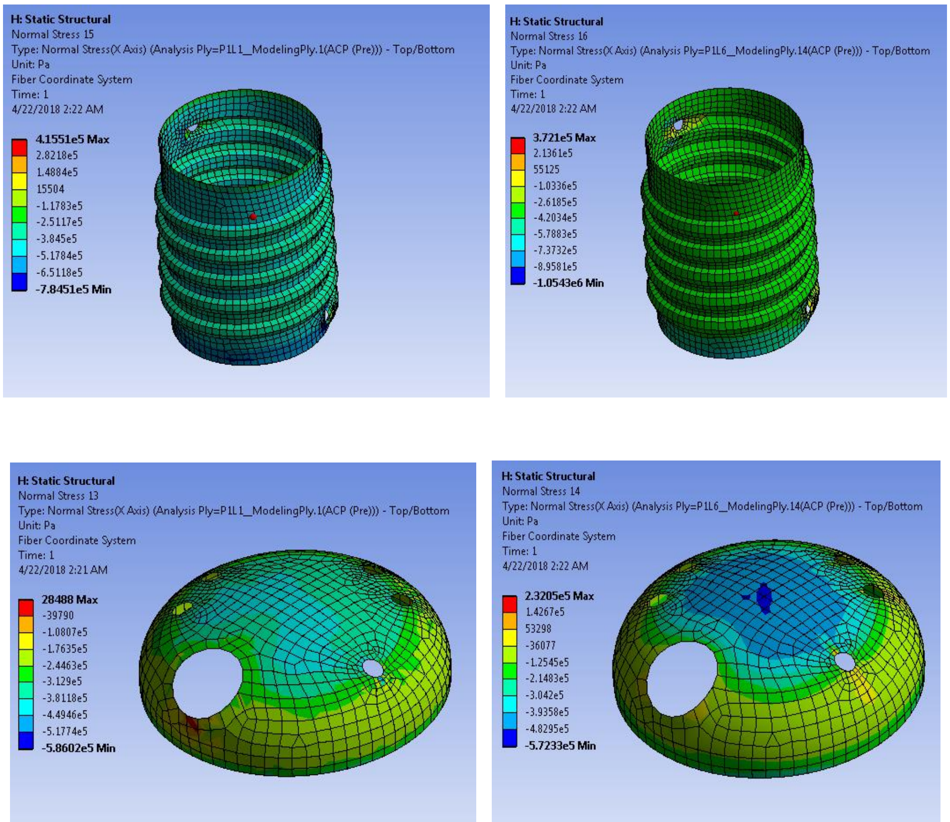


Figure 6.16: - Stresses on inner and outer plies of the shell and top dish

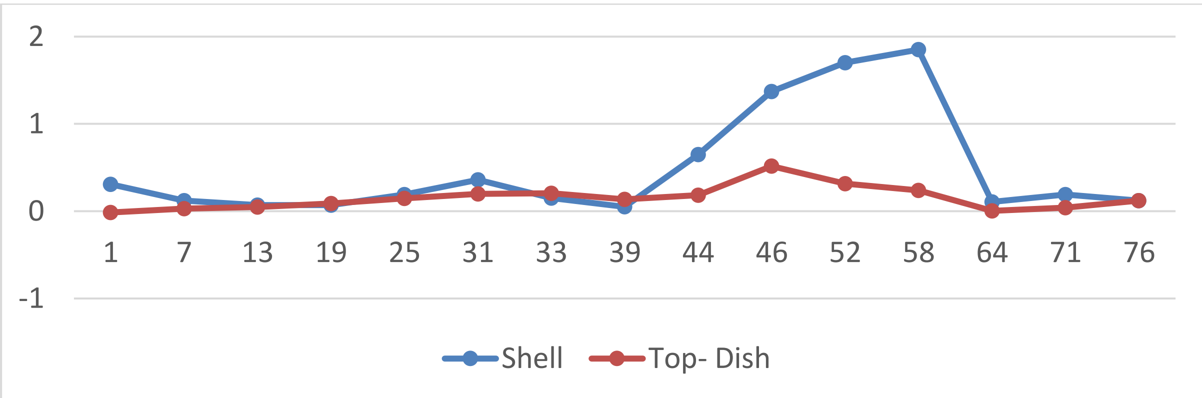


Figure 6.17: - Variation of stresses along several plies due to external pressure.

6.3.3 Case-3 under steady state thermal conduction

In this analysis, the pressure vessel is analyzed for the thermal condition. Operating temperature condition is applied on the shell side and limpet/jacket side. The temperature inside the shell is 230°C and temperature inside jacket/limpet is 280°C. The temperature on the outer surface is calculated after the analysis. Figure 6.18 shows temperature on the inner and outer ply of composite shell and top dish. Figure 6.19 shows a variation of temperature in all the plies.

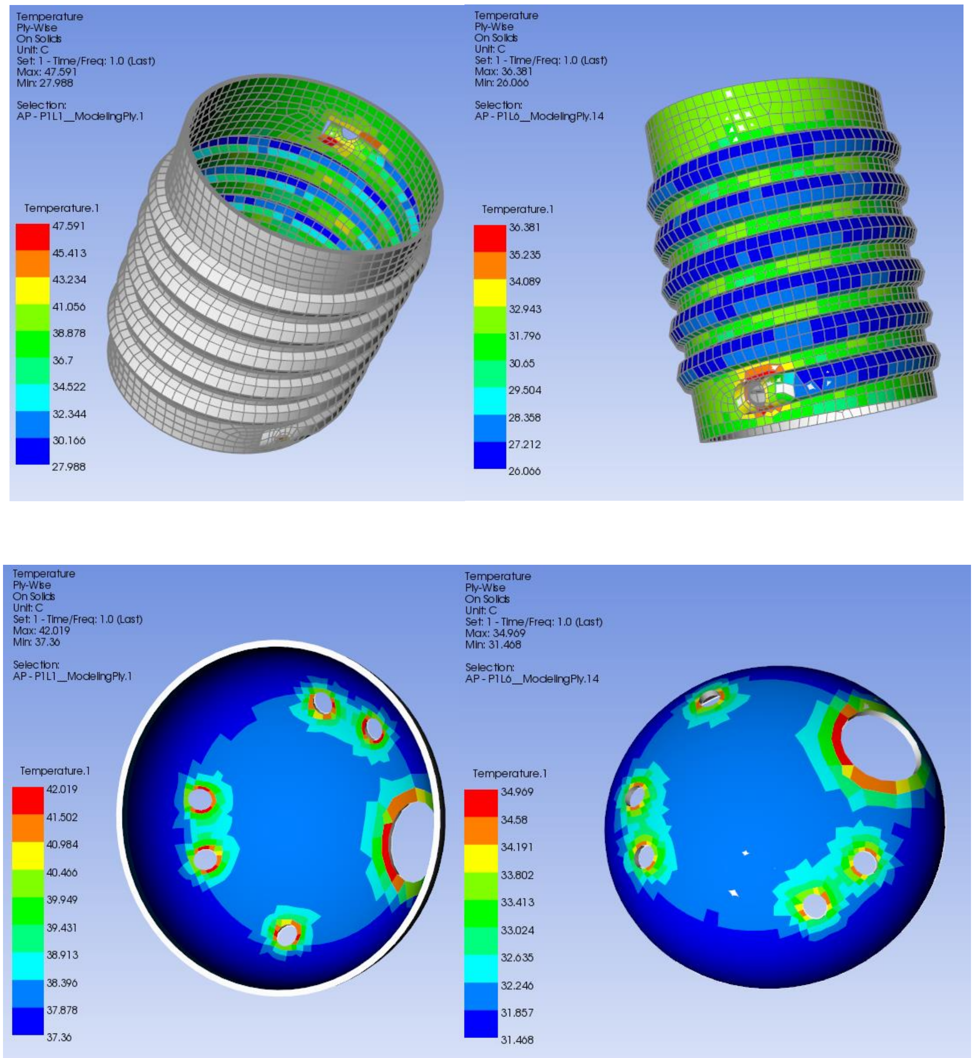


Figure 6.18: - Temperature on the inner and outer ply of composite shell and top dish

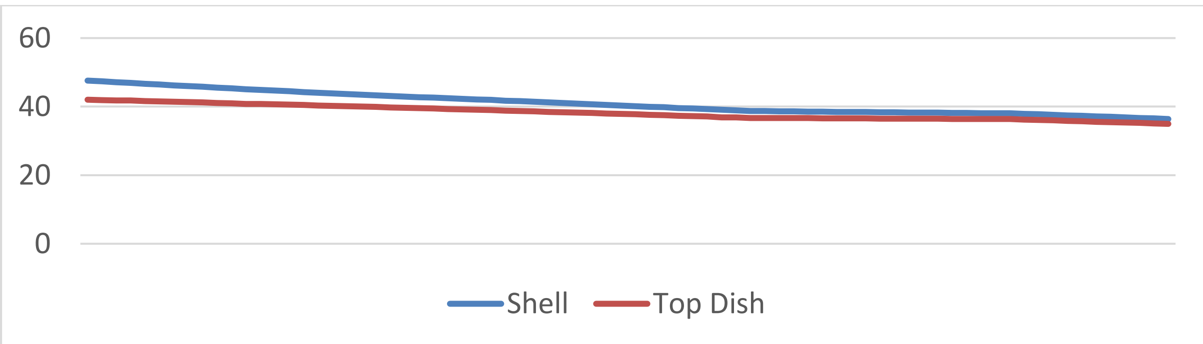


Figure 6.19: - Variation of temperature in all the plies

Chapter 7: Analytical Calculations

Theoretical calculations are done to verify Finite element results are correct. Analytical calculations are done to find values of minimum thickness required for pressure vessel, Allowable stress and maximum stress in each component of the pressure vessel. These calculations will also help in understanding change in the stress in each component of pressure vessel because of the composite material. In this chapter design of each component of a pressure vessel according to American Society of Mechanical Engineer Boiler and Pressure vessel code is discussed. Analytical calculations as per thin cylinder theory are also discussed in this chapter.

7.1 Design of Shell under internal pressure as per ASME

According to UG-27 (thickness of shell under internal pressure), the minimum required a thickness of shells under internal pressure shall not be less than that computed by the following formulas. The symbols defined below are used in the formulas of this paragraph [1]

E = joint efficiency for, or the efficiency of, appropriate joint in cylindrical or spherical shells, or the efficiency of ligaments between openings, whichever is less.

P = internal design pressure (see UG-21)

R = inside radius of the shell course under consideration

S = maximum allowable stress value.

t = minimum required thickness of the shell.

The minimum thickness or maximum allowable working pressure of cylindrical shells shall be the greater thickness or lesser pressure is given by

- Circumferential Stress (Longitudinal Joints): - When the thickness does not exceed one-half of the inside radius, or P does not exceed 0.385SE, the following formulas shall apply [1]:

$$t = \frac{PR}{SE-0.6P} \quad \text{or} \quad S = \frac{P(R+0.6t)}{Et}$$

- Longitudinal Stress (Circumferential Joints): -When the thickness does not exceed one-half of the inside radius, or P does not exceed 1.25SE, the following formulas shall apply [1]

$$t = \frac{PR}{SE+0.4P} \quad \text{or} \quad S = \frac{P(R-0.4t)}{2Et}$$

7.2 Design of Dish End under internal pressure as per ASME

According to mandatory appendix 1-4 (Design of formed heads under internal pressure), the minimum required a thickness of shells under internal pressure shall not be less than that computed by the following formulas. The symbols defined below are used in the formulas of this paragraph[1].

t = minimum required thickness of head after forming

P = internal design pressure

S = maximum allowable working stress,

E = lowest efficiency of any Category A joint in the head (for hemispherical heads this includes head-to-shell joint). For welded vessels, use the efficiency specified in UW-12

r = inside knuckle radius

L = inside spherical or crown radius for tori spherical and hemispherical heads

M = factor in the equations for tori spherical heads depending on the head proportion L/r.

$$t = \frac{PLM}{2SE-0.2P} \quad \text{or} \quad S = \frac{P(LM+0.2t)}{2Et}$$

Where, $M = \frac{1}{4}\left(3 + \sqrt{\frac{L}{r}}\right)$

7.3 Design of Half pipe coil under internal pressure as per ASME

The maximum permissible pressure P′ in half-pipe jackets shall be determined from the following formula [1]:

$$P' = \frac{F}{K}$$

where

P′ = permissible jacket pressure, psi

F = 1.5S - S′ (F shall not exceed 1.5 S)

S = maximum allowable tensile stress at a design temperature of shell or head material

S′ = actual longitudinal tensile stress in shell or head due to internal pressure and other axial forces. When axial forces are negligible, S′ shall be taken as PR/2t. When the combination of axial forces and pressure stress (PR/2t) is such that S′ would be a negative number, then S′ shall be taken as zero.

K = factor obtained from Figure EE-1, Figure EE-2, or Figure EE-3

P = internal design pressure (see UG-21) in vessel

R = inside shell or head radius, in.

D = 2R

The minimum thickness of a half-pipe jacket, when the thickness does not exceed one-half of the inside pipe radius or P does not exceed 0.385S1, is given by

$$t = \frac{P1R}{(0.85S1 - 0.6P1)}$$

where

T = minimum thickness of half-pipe jacket

r = inside radius of jacket defined in Figure EE-4

S_1 = allowable tensile stress of jacket material at design temperature

P_1 = design pressure in jacket (P1 shall not exceed P′.)

Based on the design calculations discussed above, results for minimum thickness required for the shell, dish end and Half pipe coil under internal pressure and static head are shown in Table 7.1

Component	Internal Pressure + Static Head (MPa)	Req Thk (mm)	CA(mm)	Actual thk (mm)	Remark
Top Dish	6.89	36.54	11.35	75	PASS
Cylindrical Shell	7.00	39.8	11.35	63	PASS
Bottom dish End	7.02	37.23	11.35	75	PASS
Jacket Dish End	0.68	4.01	0.0	14	PASS
Half pipe coil	0.68	0.7	0.0	14	PASS

Table 7.1: - Minimum thickness required for components of the pressure vessel.

7.4 Stress on thin-walled cylinders

If the wall thickness does not exceed the inner radius by more than approximately 10% the cylinder is classified as thin-walled cylinder[12]. A thin-walled cylinder is defined as one in which the tangential stress may, within certain prescribed limits, be regarded as constant with thickness. Following expression applies to the case of thin-walled cylinders subjected to internal pressure[12].

$$\sigma = \frac{PR}{t}$$

Chapter 8: Comparison Study

In this chapter, finite element analysis results are compared with analytical and design calculations as per ASME. Material properties of different materials used in the study are also compared to draw the appropriate conclusion.

8.1 Comparison study of Material Properties

In this study, carbon steel, E-glass, S-glass and carbon fiber has been used as materials. Figure 8.1 shows strength and density of these materials. It is evident from the figure that composite materials (E-glass, S-glass and carbon fiber) have high strength to weight ratio, low density, and high strength. The density of composite materials is three to four times less than that of carbon steel and strength is three folds more than carbon steel.

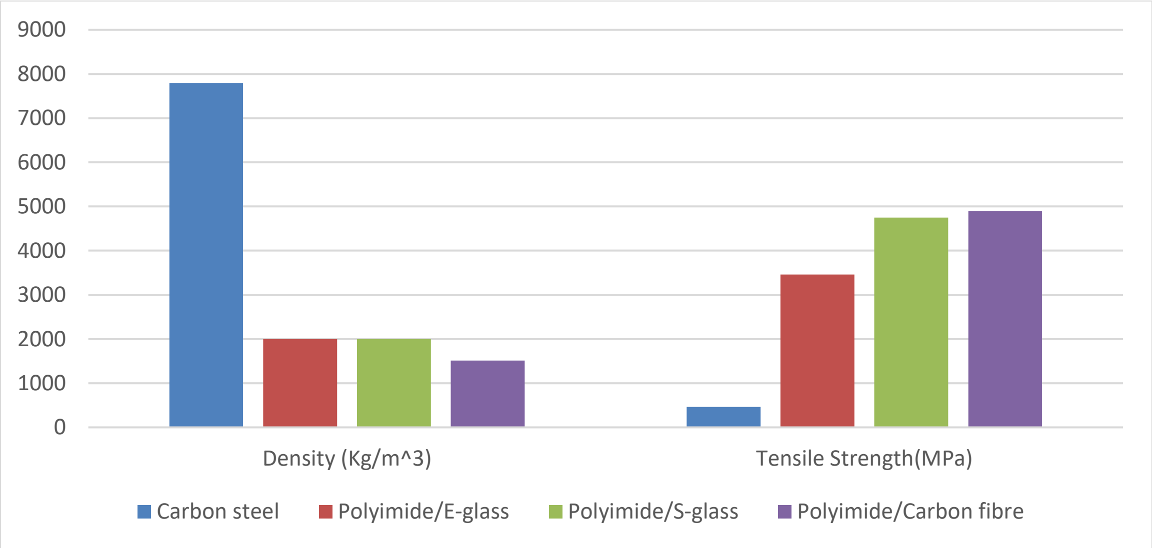


Figure 8.1: - Material properties of materials

8.2 Comparison of results for Case 1

In this section, results for maximum normal stress, allowable stress under internal pressure are discussed. The results obtained from the analytical calculation, ASME design calculations and Finite Element method have been compared. Table 8.1 shows the comparison of results. Figure 8.2 shows the graphical representation of maximum stress on each component of pressure vessel under internal pressure.

Component	Internal Pressure (MPa)	Actual thickness (mm)	Stress as per ASME (MPa)	Analytical (MPa)		FEM (MPa)	
				Hoop	Axial	Hoop	Axial
Top Dish	6.89	75	67.58	73.94	36.97	110.3	77.05
Shell	7.00	63	88.92	88.37	44.18	84.03	73.3
Bottom Dish	7.02	75	68.92	75.65	37.82	94.71	61
Jacket Dish	0.68	14	39.97	44.06	22.03	75.29	54.8

Table 8.1: - Comparison of stress under internal pressure for case 1

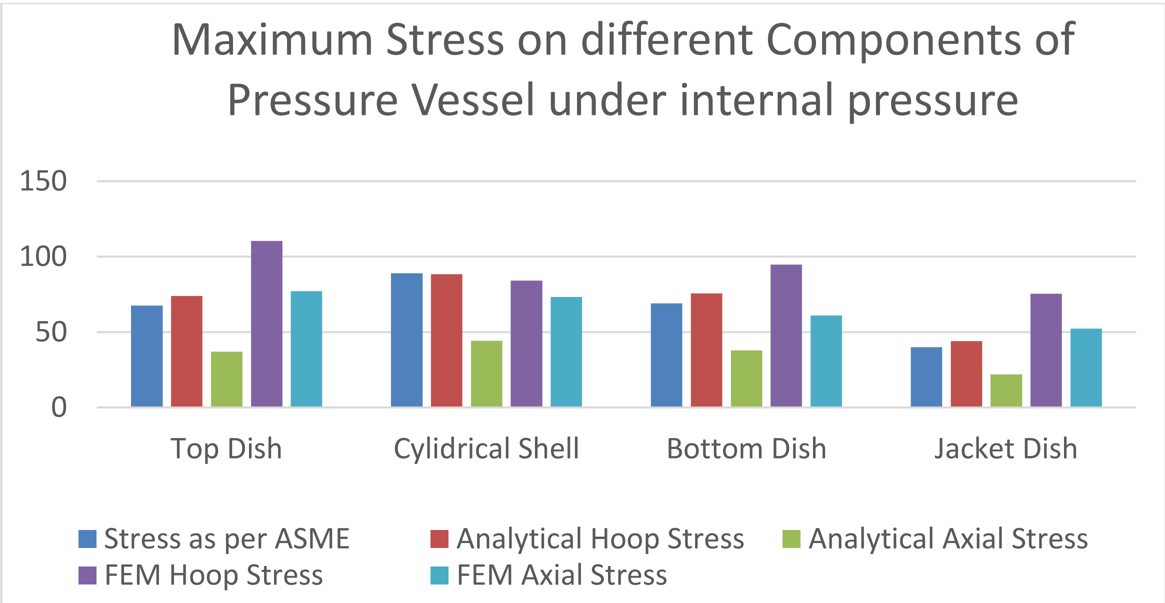


Figure 8.2: - Variation of stress on each component in Case 1

Figure 8.3 shows a variation of stress in each component of the pressure vessel. Variation of stress as per design calculation according to ASME guidance and Analytical formulae when the thickness is varied is shown in the figure.

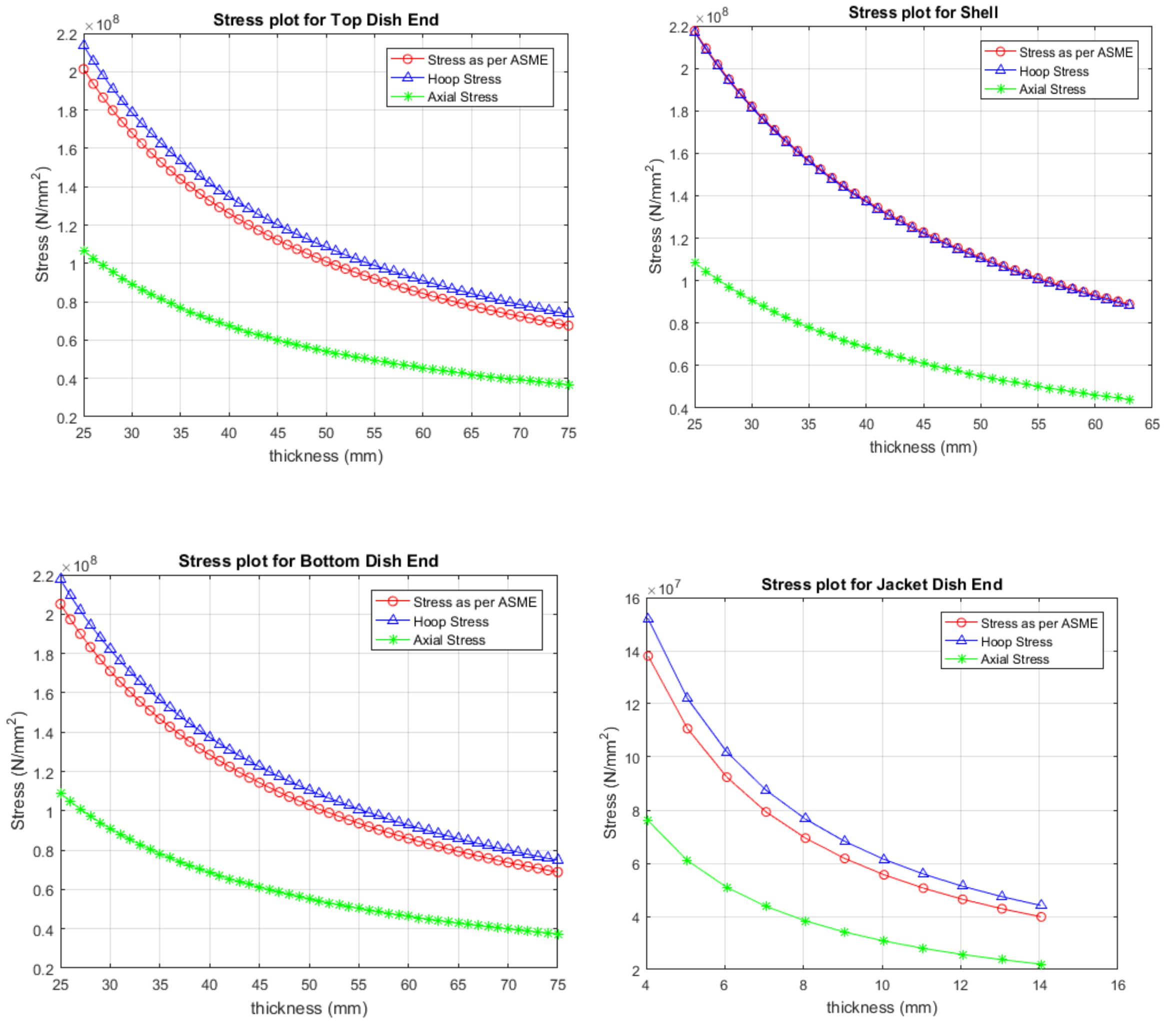


Figure 8.3: - Variation of stress if the thickness is varied

8.3 Comparison of results for Case 2

In this section, results for maximum normal stress, allowable stress under internal pressure for Case 2 are discussed. The results obtained from the analytical calculation, ASME design calculations and Finite Element method for the metallic liner of 40 mm have been discussed. Table 8.2 shows a comparison of stress using different methods. Figure 8.4 shows the graphical representation of maximum stress on each component of pressure vessel under internal pressure in Case 2.

Component	Internal Pressure (MPa)	Actual thickness (mm)	Stress as per ASME (MPa)	Analytical (MPa)		FEM (MPa)	
				Hoop	Axial	Hoop	Axial
Top Dish	6.89	40	126	137	68.9	147.6	110.6
Shell	7.00	40	137	138	69.4	144.2	130
Bottom Dish	7.02	40	128.5	140	70.2	126.1	144.7
Jacket Dish	0.68	14	39.8	44.06	22.03	65.75	84.41

Table 8.2: - Comparison of stress on each component of the metallic liner in Case 2

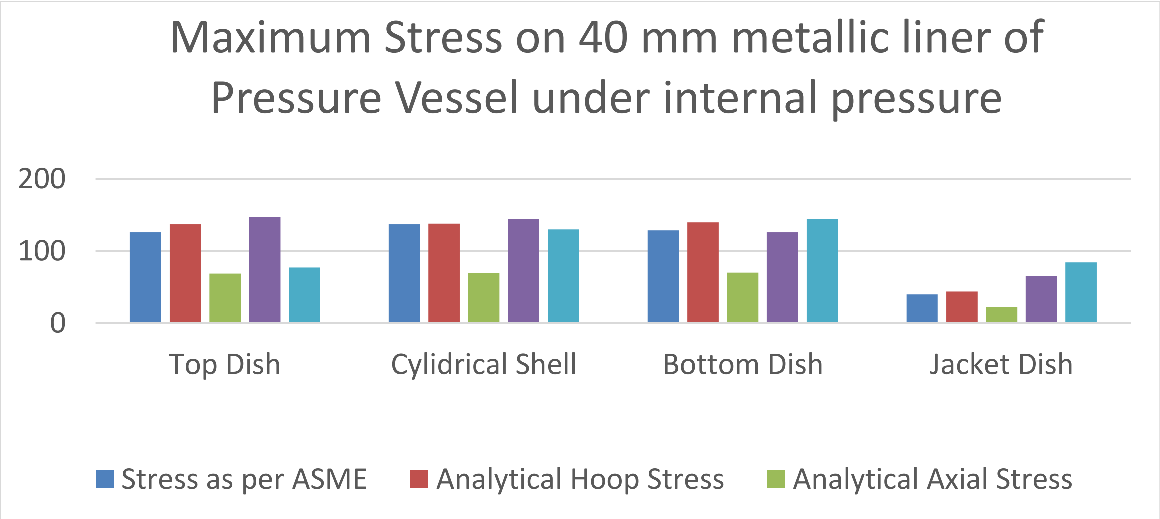


Figure 8.4: - Variation of stress on each component in Case 2

8.4 Comparison of results for Case 3

In this section, results for maximum normal stress, allowable stress under internal pressure for Case 3 are discussed. The results obtained from the analytical calculation, ASME design calculations and Finite Element Method for the metallic liner of 30 mm have been discussed. Table 8.3 shows a comparison of stress using different methods. Figure 8.5 shows the graphical representation of maximum stress on each component of pressure vessel under internal pressure in Case 3.

Component	Internal Pressure (MPa)	Actual thickness (mm)	Stress as per ASME (MPa)	Analytical (MPa)		FEM (MPa)	
				Hoop	Axial	Hoop	Axial
Top Dish	6.89	30	167	183	91.8	145.7	122.2
Shell	7.00	30	182	185	92.6	217.3	212.2
Bottom Dish	7.02	40	128.5	140	70.2	132.3	147.1
Jacket Dish	0.68	14	39.8	44.06	22.03	73.7	81.4

Table 8.3: - Comparison of stress in Case 3

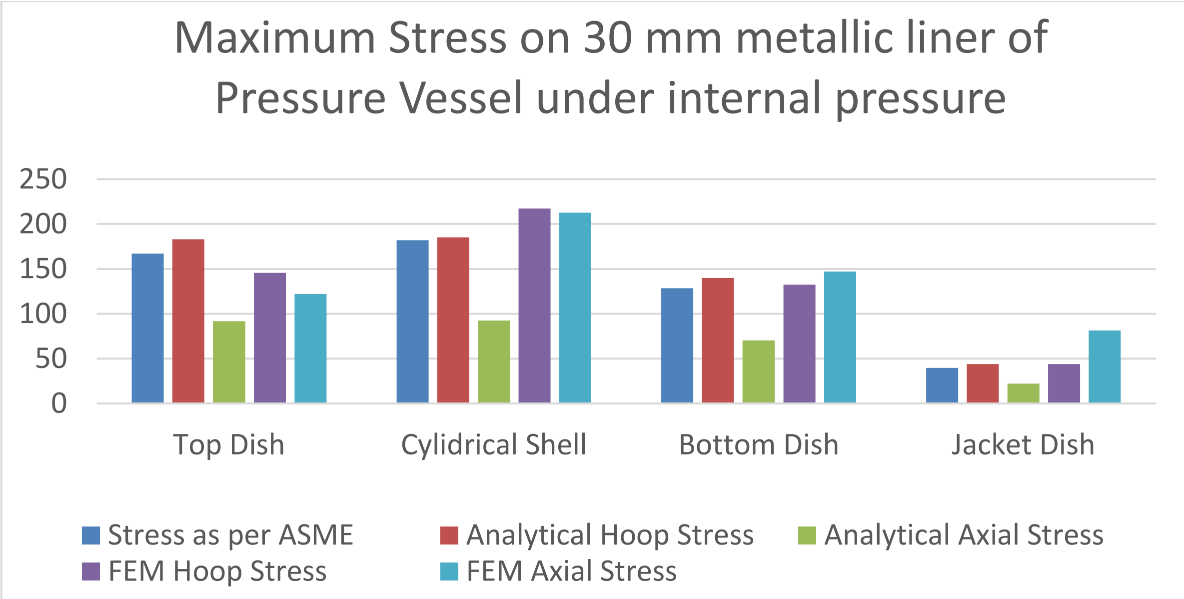


Figure 8.5: - Variation of stress on each component in Case 3

8.5 Comparison of stress in the composite material

In this section, stress in each ply of composite materials has been discussed. Variation of stress on Polyimide/E-glass composite shell and the top dish is shown in Figure 8.6. Variation of stress in the composite shell and a top dish of case 3 is also shown in Figure 8.6

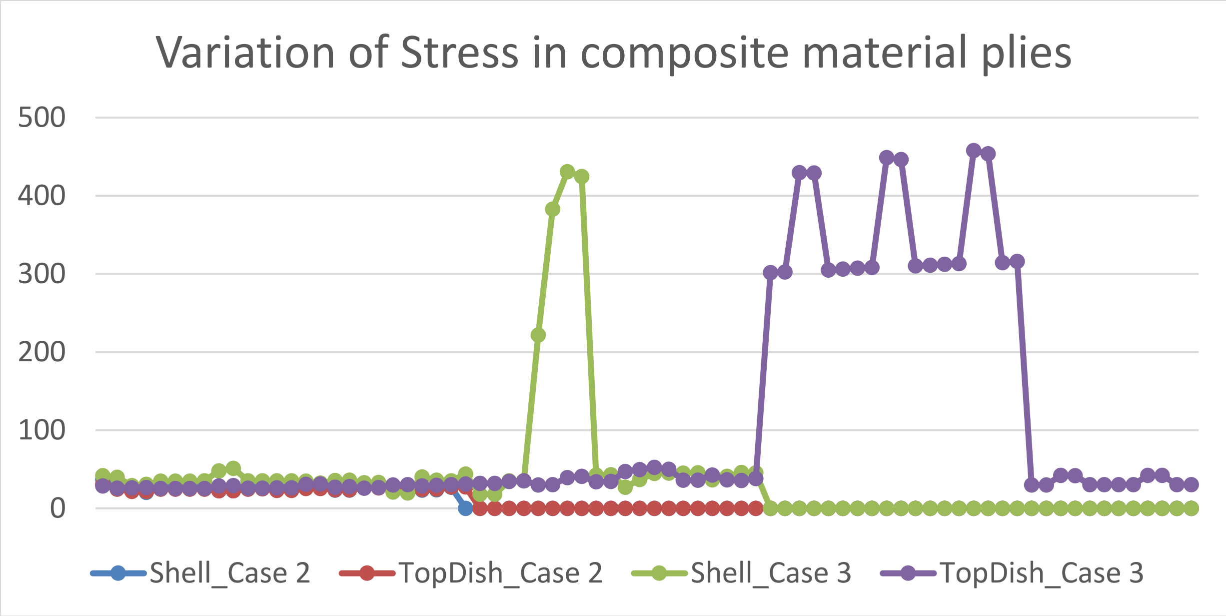


Figure 8.6: - Variation of stress in Composite plies in Case 2 and Case 3.

8.6 Weight Difference

For all the cases of analysis, geometry was created in ANSYS and material was assigned. Based on the density of each material and volume of material, weight is calculated. The overall weight of each case is shown in the following Table 8.4

Model	**Weight in Kg**	**Difference**
Case 1: - Carbon steel model	10986	N/A
Case 2: - 40 mm Metallic liner overwrapped with Polyimide/ E-glass	8100	26%
Case 3: - 30mm metallic liner overwrapped with Polyimide/ E-glass, Polyimide/ s-glass and polyimide/ carbon	7294	33%

Table 8.4: - Weight difference

Conclusion

- Based on the comparative study of weight of pressure vessel in all the cases, it is evident that weight of the pressure vessel metallic liner overwrapped with composite reduces by 25-33% due to low densities of composite materials.
- It is observed that safety Factor in carbon steel model and Metallic liner overwrapped with a composite material (Case 2 and Case 3) remains same, after reduction in overall thickness by 10mm.
- In Case 2 and Case3, stress in shell remains constant, while stress in Top dish gets reduced due to high strength and tensile stress of the composite material.
- The temperature on the outer wall is in the range of 30-40 °C, therefore it is not necessary to provide hot insulation to pressure vessel.
- Due to a reduction in weight of pressure vessel by 25-33% and no necessity of hot insulation, cost of operation, cost of maintenance and cost labor associated with the operation of the pressure vessel is reduced.

Future Work

- Transient thermal analysis can be performed considering thermal cycle or production cycle to determine the effect of thermal stress.
- Fatigue analysis can be performed to determine the life of reactor pressure vessel.
- Study of transient response analysis can be done to determine the mode of failure due to vibration.
- In this study, 60% fiber was used. Hence, fiber- Matrix concentration can be varied, and results can be observed.
- In this study, Kapton was used as polyimide matrix. Different industrially produced polyimide materials can be used as matrix material.

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Biological Statement

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