Design of Pressure Vessel : A Comparative Study Between Code Vs Analysis Approach

By

Salman Ahmed Qureshi

2011-NUST-MS PHD- 33

MS-68

Submitted to the Department of Mechanical Engineering in Fulfillment of the Requirements for the Degree of

MASTER OF SCIENCE

in

MECHANICAL ENGINEERING

Thesis Supervisor

Dr.Hasan Aftab Saeed

College of Electrical & Mechanical Engineering

National University of Sciences & Technology

2015

In the name of Allah, the most Beneficent and the most Merciful

Declaration

I hereby declare that I have developed this thesis entirely on the basis of my personal efforts under the sincere guidance of my supervisor Dr Hasan Aftab. All the sources used in this thesis have been cited and the contents of this thesis have not been plagiarized. No portion of the work presented in this thesis has been submitted in support of any application for any other degree of qualification to this or any other university or institute of learning.

Salman Ahmed Qureshi

Acknowledgments

All thanks and praises are to ALLAH with whose Grace and will I accomplished this task. I feel an obligation and an honor to express deep gratitude to **Dr.Hasan Aftab Saeed** for his encouraging behavior and continuous support. His kind and seasoned personality not only led me in our endeavors but also inspired greatly. He not only provided such a useful project but also helped and gave courage to achieve goal.

I am greatly indebted to **Dr.Wasim Akram** for providing me valuable guidance and advice. His encouragement was the main source of motivation that helped me to complete the project.

Finally, I express a special gratitude's to my parents whose prayers pull away all the hindrance faced by me during work.

Dedication

To,

My Loving Parents.

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Abstract

Pressure vessels are most widely used in industrial machinery contributing as main components for proper functioning of any mechanical system. Primitive purpose of these containers is storage. This includes preservation of fluids including liquids and gases with in specific safety conditions. When these gases are stored in adequate arrangements at required temperature and pressure conditions accidents are avoided and more safety is ensured. This reason has lead to common use of pressure vessels within different practices of any social network.

Use of pressure vessels in nuclear sector is very important. Application of these mechanical components in nuclear field has brought tremendous change in lives of nations over past few decades. Research and developments of countries is based majorly on adequate use of pressure vessels. Ranging from simple cruises to ballistics and even space launch vehicles ,the main driving applications are based on pressure vessels.

Pressure vessels serve as potential source for storing energy and therefore these have safety issues. During the start of twentieth century several accidents occurred and the world realised the safety risks involved with improper use of pressure vessels. These catastrophic accidents resulted in heavy loss of lives. Manufactures immediately felt the need to regularize the criteria for designing and fabrication of pressure vessels. Some of these interested parties jointly appealed to American Society of Mechanical Engineers for their assistance to formulate rules. The intention was to formulate standard specifications for steam boilers and pressure vessels. The very first set of codes for pressure vessels was established in 1925 with the title of “Rules for construction of

Pressure Vessels”. Nowadays all pressure vessels are designed in the United States in accordance with ASME Section VIII.

In the presented research design of pressure vessel according to ASME code section VIII is presented. Pressure Vessel has been designed for the specific application according to rules and procedures given in ASME Section VIII Division I. The designed pressure vessel has been modeled by using CREO. This model of complete 3D assembly of pressure vessel is subjected to structural analysis. The software tool used for analysis is ANSYS. Next several cases have been analyzed and results have been compared. The analysis procedure has been validated by experimental values of hydro static test results. Hence the efficiency of pressure vessel will be optimized by in terms of weight and cost effects.

Nomenclature

t	Wall thickness
p	Pressure
d	Diameter
σ_L	Longitudnal Stress
σ_H	Hoop Stress
σ_v	Von-Misses Stress
S	Maximum allowable stress value
E	Weld joint efficiency
K	Stress concentration factor
H	Height of elliptical secton
ν	Poisons Ratio
β	Constant depending on r/t ratio
ϕ	Included Weld Angle
K1	Stress concentration factor for longitudinal stresses
K2	Stress concentration factor for hoop stresses

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Chapter 1

INTRODUCTION TO PRESSURE VESSELS

Pressure Vessel is closed container to store fluids i.e. liquids and gases under required conditions of temperature and pressure. The pressure can be internal or external or both. Pressure vessels have a vast area of application in industrial sector and also in private sector. Ranging from domestic sector to industrial applications pressure vessels have vast scope of usage. Hot water storage tanks and compressed air receivers are common examples of domestic usage. Pressure vessels also find wide area of scope in mining operations, oil refineries and petrochemical plant and nuclear reactor vessels. External pressure vessels are used in submarine applications. Use of pressure vessels in process industries is also wide spread ranging from storage to continuous processes like distillation. Liquified gases like ammonia , chlorine, propane and butane are mostly stored in pressure vessels. Pressure vessels find wide area of application in oil exploration as well. One major area of use of pressure vessels is their application in aerodynamic applications like missile systems, space vehicles and satellite launch vehicles.

Fig 1- Simple Cylindrical Pressure Vessel

1.1 Pressure Vessels – Types

Based on shapes pressure vessels can be found to be of variety of shapes. However most commonly pressure vessels are spherical, cylindrical and conical vessels. Cylindrical body having hats at both ends known as heads. Heads of pressure vessels can also be of different shapes and it depends the design needs and application of the component. Heads can either be hemispherical as can be seen in submarines. A very common form of heads is elliptical heads and is usually employed in aerospace structures. But the shape of head should be less complex and less complicated from the design point of view for safe operation. Less complex shapes are also easy to manufacture. Following four criterias are adopted for classification of pressure vessels.

1) Shape of Pressure Vessel
2) Type of Pressure Applied (Internal/External)
3) Wall Thickness
4) Type of Head

1.2 Shape of Pressure Vessel

Based on shape pressure vessels are majorly of following two types.

1) Spherical Pressure Vessels
2) Cylindrical Pressure Vessels

1.2.1 Spherical Pressure Vessel

Fluids to be stored under high pressure are preferably stored in spherical pressure vessels. Sphere shaped vessels have good structural strength. From design view point it follows that stress distribution along spherical surfaces is very uniform and the structure has no weak zones. Hence there are very less chances of crack initiation and this contributes to the better strength. Manufacturing cost of spherical vessels is high as compared to cylindrical vessels. Spherical vessels used for storage purpose need ancillary equipment like accesing holes and hatch covers. Vents for pressure and vaccum must also be provided to prevent losses due to high or low temperatures and earthing points.

Spherical pressure vessels have thermal advantages also.This is mainly because of reason that heat transfer rate is very less and the liquids can be stored at much lower temperature and pressure.This is due to the fact that surface area to volume ratio for spherical vessels is lesser and makes major contribution to low

heat transfer rate.

Fig-2 Spherical Pressure Vessels

1.2.2 Cylindrical Pressure Vessel

Cylindrical pressure vessels usually bear less manufacturing cost as compared to spherical vessels. This reason makes them a good choice for storage uses. However cylindrical vessels are less strong as compared to spherical pressure vessels. Thus if cylindrical vessel is to be used as storage vessel then to compensate for better strength more material will be required than to spherical vessel of same capacity.

Fig-3 Cylindrical Pressure Vessel

An example of use of cylindrical pressure vessels is in missile applications. The combustion chamber of solid rocket is basically a cylindrical pressure vessel. This pressure vessel holds internal pressure generated by burnt solid fuel. Consequently thrust generated by burning of fuel propells the rocket. This process is shown below.

Fig-4 Use of Cylindrical Pressure Vessel in Solid Rockets

1.3 Type of Pressure Applied

Based on applied pressure there are two major types of pressure vessels.

1) Internal Pressure Vessels
2) External Pressure Vessels

1.3.1 Internal Pressure Vessel

Internal pressure vessels are those having pressure applied to the inner walls of the vessel. Generally all pressure vessels used for storage of gases are internal pressure vessels. Inner cylindrical surface is the bearing surface for pressure in these vessel types. Internal pressure vessels can be placed horizontally as well as on up right supports depending on its application. Diagrametic representation for internal pressure case is shown below:-

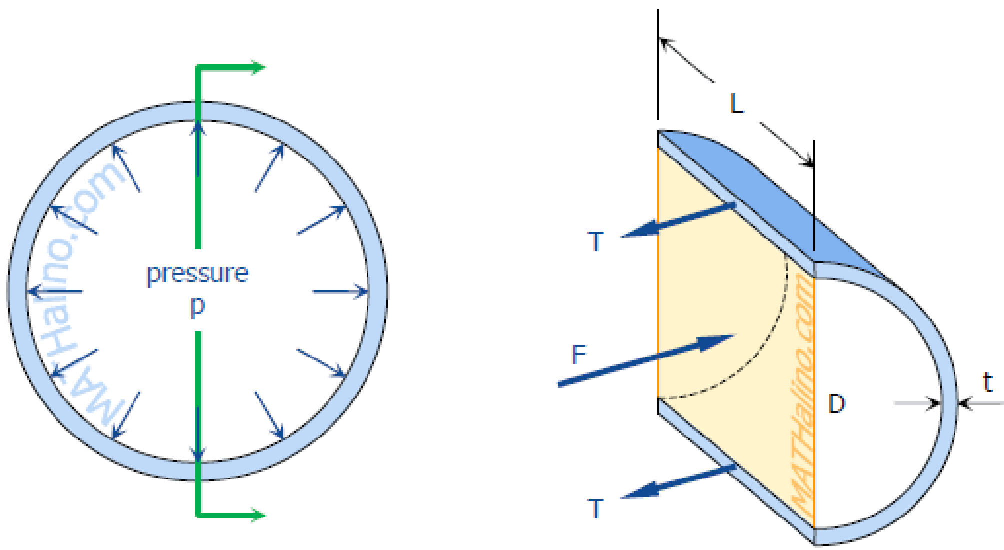


Fig 5 – Application of Internal Pressure

1.3.2 External Pressure Vessel

External pressure vessels are those having outer periphery as the pressure bearing surface. Generally there are following three case to apply external pressure on a vessel.

1- Existence of vaccum inside a vessel and atmospheric pressure is outside the vessel.

2- If the outside pressure is greater than the atmospheric pressure e.g. some types of jackets around the vessel or any other vessel in its surroundings.
3- When the first two are combined i.e. vaccum inside and greater pressure than atmospheric pressure outside.

External pressure application is depicted in following figure.

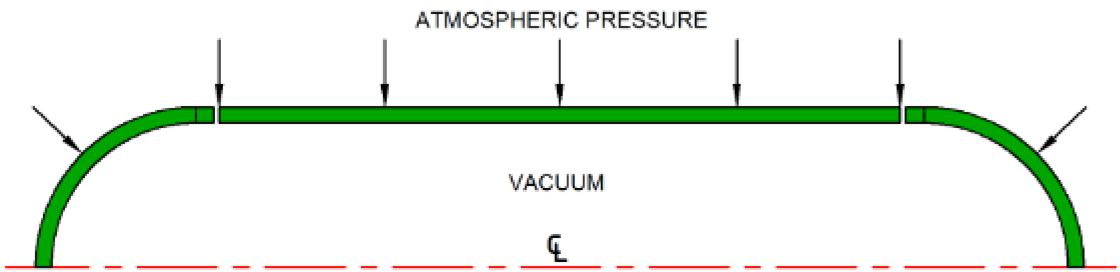


Fig 6 – Application of External Pressure

1.4 Wall Thickness

Pressure vessel wall thickness is a very important design aspect to ensure proper functioning of any vessel. Broadly pressure vessels can be classified in two major types based on wall thickness.

1) Thin Walled Pressure Vessels
2) Thick Walled Pressure Vessels

1.4.1 Thin Walled Pressure Vessel

Thin walled pressure vessel are those which have thickness to radius ratio less than or equal to 1/10. Let "t" represent the wall thickness and "r" be the radius of the vessel then mathematically ,

$$t / r \leq 1/10$$

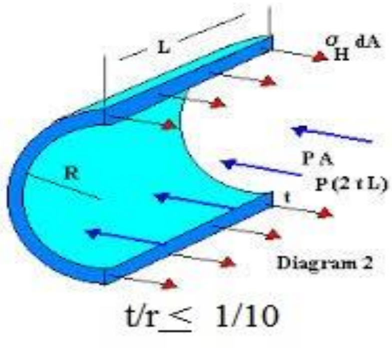


Fig 7 – Thin Walled Pressure Vessel

1.4.2 Thick Walled Pressure Vessel

Thick walled pressure vessel are those which have thickness to radius ratio greater than 1/10. Let "t" represent the wall thickness and "r" be the radius of the vessel then Mathematically ,

$$t / r > 1/10$$

1.5 Pressure Vessel – Type of Head

A major classification of pressure vessel is based on the head type. Following three types of pressure vessels can be categorized depending on head types.

1) Ellipsoidal Heads
2) Hemispherical Heads
3) Torispherical Heads

1.5.1 Ellipsoidal Heads

Mostly 2:1 elliptical heads are used in these type of vessels. In this type of vessel height of head is quarter of its diameter. This type of head of vessel is economical to manufacture. The radius of vessel is not constant and it varies between major and minor axis. Ellipsoidal head is shown in following figure.

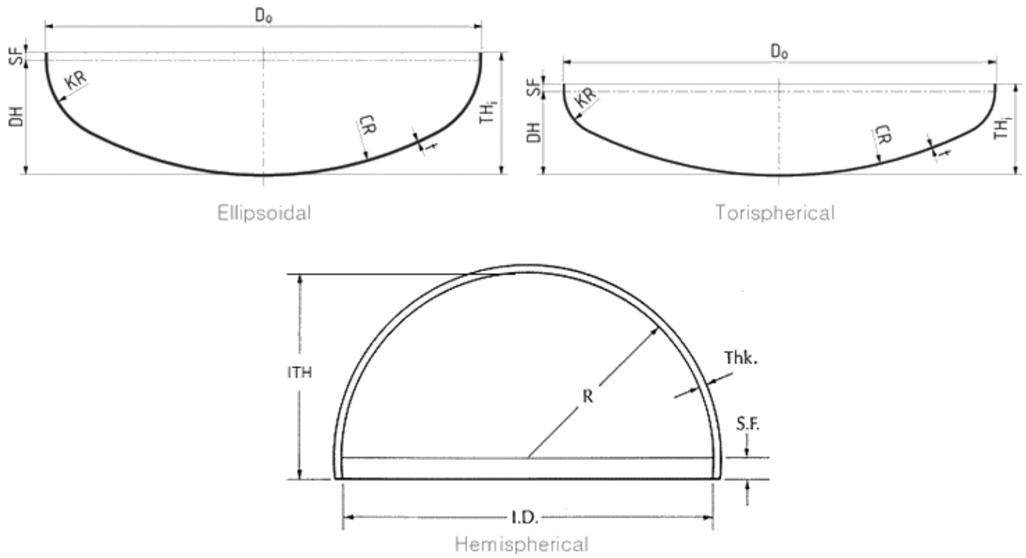


Fig 8 – Types of Heads of Pressure Vessels

1.5.2 Hemispherical Heads

The most preferred shape for head is sphere and this is because of the reason that pressure is equally distributed across the surface. The value of radius of head is equal to the cylindrical radius of the vessel.

1.5.3 Torispherical Heads

These type of head shapes are more complex than above discussed. Normally these heads have fixed dish of some constant radius R. Dish size depends on size of vessel. Cylindrical part of vessel is joint to the dish with torroidal shaped part.This is known as knuckle. The above figure 8 shows types of heads of vessel.

Chapter 2

ASME CODES FOR PRESSURE VESSELS

2.1 HISTORICAL BACKGROUND

Historically the first reference for the pressure vessel design dated back to 1495 when Leonardo da Vinci quoted in his Codex Madrid I. He gave his idea to push air beneath the surface of water with the effect to lift weights. He described a way to lift skins by filling them with air after securing them to weights under water surface. Similarly he described the way to lift weights by tying them to ships. This concept of filling air under pressure was used in early models of ateam generators as well.

The main source of power for the era of 19^{th} and 20^{th} century was steam and it brought a revolution in industry. Statistics of early 20^{th} century tell that boiler explosion accidents operating on steam were occurring very frequently. Approximate rate of boiler explosions was one per day and it claimed lives of two persons per day. Two catastrophic boiler explosions occurred in 1907 in Massachusetts and this led to the formulation of first legislation to deal with designing and manufacturing steam boilers. During following four years many other cities applied same rules for steam boilers. These enacted rules and regulations along with additional laws provoked customers and industrialists to apply these standard rules in the designing of boilers. The scope was further enhanced to fabrication of boilers and inspection requirements.

The first committee to formulate specifications for manufacturing of steam boilers along with pressure vessels was appointed by American Society of Mechanical Engineers (ASME) in 1911. The committee included seven regular members and there were eighteen members of the advisory committee. The members of committee were from all fields of the boilers and pressure vessels including designing , fabrication , installation , testing and commissioning of pressure vessels and steam boilers.

The first code was established on February 13, 1915 known as ASME boiler code. During following eleven years six sections were added to the initial code. The first publication for pressure vessels was issued during 1925. This publication had title as "Rules for the Construction of Unfired Pressure Vessels," Section VIII.

Several sub committees for services are part of the main ASME Boiler and Pressure Vessel Code Committee. The committee for books like subcommittee for boilers and pressure vessels is responsible to publish code books. The subcommittees for service include technical experts which are not included in books subcommittee to become consultants for the latter committee. , There are two more subcommittees for materials and welding to specify rules in their subjective fields. These subcommittees act as both book as well as service subcommittees. Several sub groups are formed under these subcommittees. The sub groups are further divided in to smaller working groups and task forces. These sub groups have the responsibility for a particular component and specific technical aspect of pressure vessel. For instance the layered vessels are been looked by a work group which reports to respective sub group and then sub group reports to subcommittee for inspection and manufacturing. The working group for layered vessels directly handles all technical matters relevant to layered vessels only.

2.2 ASME CODES – SECTIONS

In this section an effort is done to give a quick over view of ASME boiler and pressure vessel codes. This gives an ease to have a rapid look at different sub sections without the need to read all the long theoretical details given in code. This section will also be useful to pin point the various sub sections for the particular part and components. Also included is a handy reference chart (Fig. 9) that graphically illustrates the various parts of a pressure vessel and the Code paragraphs that apply to each.

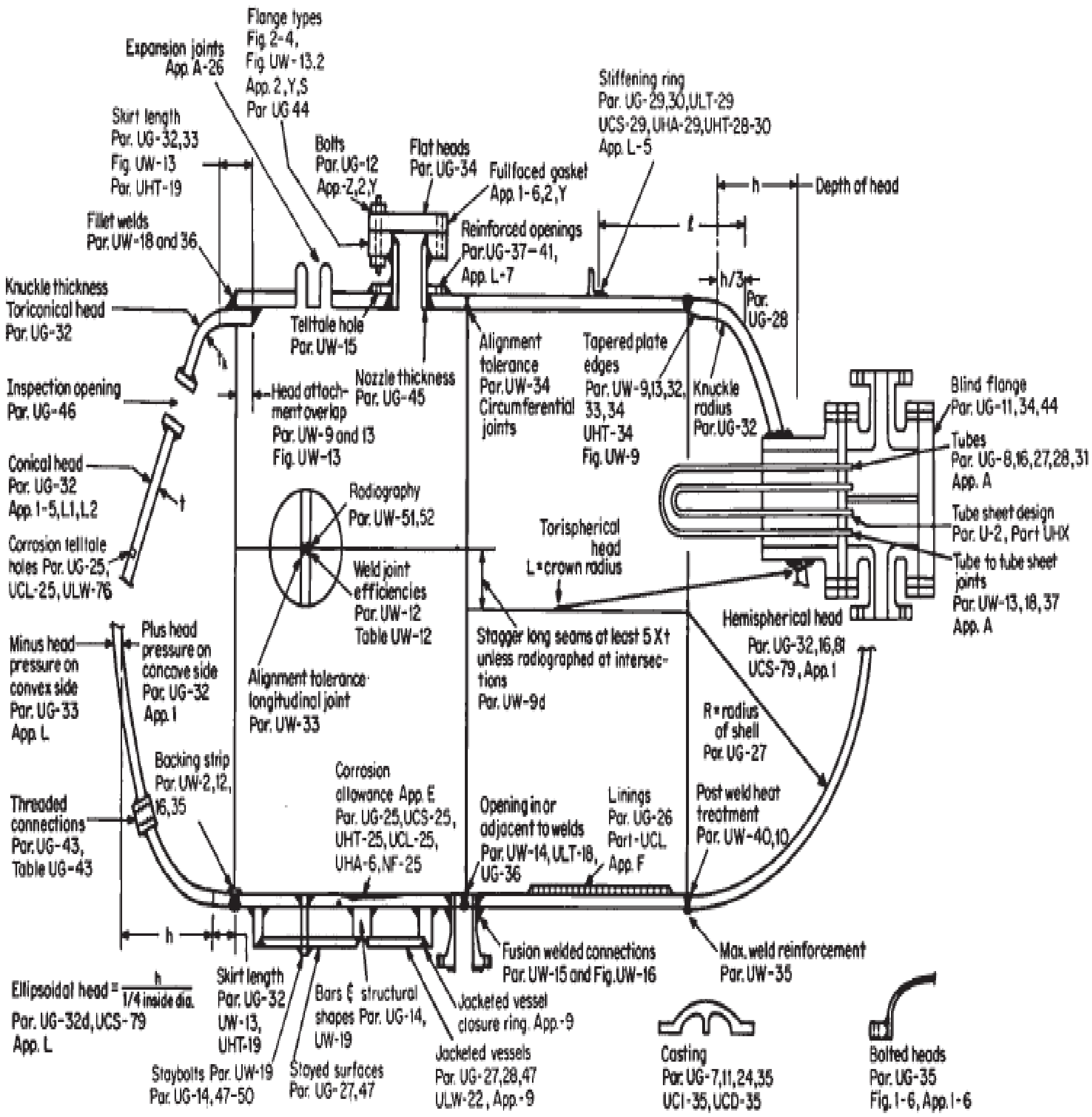


Figure 2.1 Reference chart for *ASME Pressure Vessel Code,* Section VIII, Division 1.

Fig -9 ASME Codes Sections Summary

2.3 ASME CODES – SCOPE

Every article is identified by an alphanumeric code in ASME Section VIII, Division 1. This system of labelling is used for all the Boiler and Pressure Vessel Code (BPVC) sections. Letter symbol U is used to start the labels of all symbols in Division I.

This first letter U is followed by a second letter which symbolizes the discussion information in the particular article. For example the items starting UG represent the general requirements in Division I. Similarly, UW symbolizes welding of pressure vessels and the general requirements associated to welding. Articles starting with the symbol UCS represent fabrication of pressure vessels using carbon steels and alloys of steel. The same trend continues for other articles as well. The two starting letters are followed by an alpha numeric number for sequence purpose. These numbers may or may not be consecutive

.

The review of the Division is done constantly. Those articles which become time barred with technology are continuously deleted depending on the advancements in that particular field. Addition of new articles is done also to keep pace with the latest technological needs. For instance the requirements related to riveting of pressure vessels are considered no longer in Division. Whereas some additions have been done to include refinements for fabrication of pressure vessels by using high strength carbon steels to minimize the failure risks and avoid catastrophies.

2.4 ASME CODES – MATERIALS (UG-4 To UG-9)

These sections give the requirements of materials that choice of material for designing should be according to technical specifications mentioned in ASME Section II. ASME section II part D enlists wide range of materials. The properties of materials are also given in ASME Section II part D along with the temperature ranges.

Hence the material used during fabrication must be according to the material specification given in ASME Section II. The material specifications mentioned in Division are very suitable for all conditions of usage and materials can be processed on all modern mills. Material properties which include chemical properties along with mechanical requirements are mentioned in the Division in terms of certain limits for the specific temperature ranges.

Division 1 describes welding requirements in pressure vessels when one component is welded to the other. The welding constraints further state that the material in use for welded pressure vessel must be weld able. Generally the materials which are not approved by code must not be used for designing of pressure vessels.

There can be different forms in which a particular material can be used. Some material are used in the form of plates while some are used in forged forms while others are used as castings. Articles from UG-5 to UG-7 provide these specific requirements. Section II part D gives stress values that are allowable for a particular product form.

UG−8 provides with requirements in seamless welded pipe that must meet the specification in Section II. The allowable stress values for these materials are also provided in Section II, Part D. Paragraph UG−8 provides guidelines to the internal tube structures that are often used with in piping.

UG-27 gives the design formulas to calculate thickness of shells in case of cylindrical pressure vessels. However the material selection requirements are same as those described keeping in view the maximum allowable stress values.UG-36 describes the design process for the head of pressure vessels.

Section UW-12 specifies the weld joint efficiencies for pressure vessels. These efficiency is given in the form of factor E and this value is used in thickness formulas with the maximum allowable stress values.

2.5 ASME CODES – WELDING JOINTS

Type of joint and radiography	Efficiency allowed, percent	Code reference
Double-welded butt joints (Type 1)		Par. UW-11
Fully radiographed	100	Pars. UW-51, UW-35
Spot-radiographed	85	Pars. UW-12, UW-52
No radiograph	70	Table UW-12
Single-welded butt joints		Par. UW-52
(backing strip left in place) (Type 2)		Par. UCS-25
Fully radiographed	90	Par. UW-51
Spot-radiographed	80	Par. UW-52
No radiograph	65	
Single-welded butt joints no backing strip (Type 3) limited to circumferential joints only, not over $\frac{5}{8}$ in thick and not over 24-in ouside diameter	60	Table UW-12
Fillet weld lap joints and single-welded butt circumferential joints		Table UW-12
Seamless vessel sections or heads (spot-radiographed)	100	Par. UW-12(d)
Seamless vessel sections or heads (no radiography)	85	Par. UW-12(d)

TABLE 1 <u>WELDING JOINT EFFICIENCIES</u>

2.6 ASME CODES – CALCULATING THICKNESS VALUES FOR SHELLS

Equations	Remarks	Code reference
Cylindrical Shells under Internal Pressure		
$t = \dfrac{PR}{SE - 0.6P}$	Equations used when t is less than ½R or P is less than 0.385SE	Par. UG-27
or		
$P = \dfrac{SEt}{R + 0.6t}$	Circumferential joint formula in Code Par. UG-27 applies only if the joint efficiency is less than one-half the longitudinal joint efficiency or if the loadings described in Code Par. UG-22 cause bending or tension [see examples in Code Appendix L-2(a) and (b)]	Pars. UG-27, UG-22 Appendix L-2(a) and (b)
in which: t = minimum thickness, in P = allowable pressure, psi S = allowable stress, psi E = joint efficiency, percent R = inside radius, in		
	Thick cylindrical shells	Appendix 1-2
	Formulas in terms of the outside radius may be used in place of those in Par. UG-27	Appendix 1-1 to 1-3
	Out-of-roundness measure	Par. UG-80
	Welded joint efficiencies	Par. UW-12 Table UW-12
	Stress values of materials (use appropriate table)	Code Section II, Part D, Tables UCI-23, UCD-23, ULT-23
	Corrosion allowance for carbon-steel vessels	Par. UCS-25

TABLE 2 THICKNESS CALCULATIONS FOR SHELLS

2.7 ASME CODES – CALCULATING THICKNESS VALUES FOR HEADS

Equations	Remarks	Code reference
Ellipsoidal Heads under Internal Pressure		
$t = \dfrac{PD}{2SE - 0.2P}$ or $P = \dfrac{2SEt}{D + 0.2t}$	Equations to be used when ratio of the major axis, D, to the minor axis, h, is 2:1 (see illustration above). For other ratios, see Appendix 1-4	Par. UG-32(d) Appendix 1 (also footnotes)
where t = minimum thickness, in P = internal pressure, psi S = allowable stress, psi E = minimum joint efficiency, percent D = inside diameter of head skirt, in	Minimum thickness (after forming)	Pars. UG-32(a) and (b)
	Allowable percentages of stress values and joint efficiencies for material depend upon type of radiography—full, spot, or no radiography	Pars. UW-11, UW-12 Pars. UW-51, UW-52
	Required skirt length	Pars. UG-32(l) and (m) Fig. UW-13.1
	Attachment of heads to shells	Fig. UW-13
	Tolerances for formed heads	Par. UG-81
	Cold-formed heads of P-1, Group 1 and 2 material may require heat treatment	Par. UCS-79

TABLE 3 <u>THICKNESS CALCULATIONS FOR HEADS</u>

Chapter 3

LITERATURE SURVEY

3.1 Approach for Literature Survey

Literature survey for the project gives a basic idea of carrying out the research. The previous work done by different researchers is available in the form of papers on different forums. These research papers can be accessed through on line resources. Literature survey for a particular work should be relevant to its field. The more are the research papers relevant to the area of work the easier it becomes for the researcher to make a pattern for working in specific field. Hence while carrying out literature survey, one should be clear about his field of work like finite element analysis and design , control systems , engineering optimization , computerized integrated manufacturing , bond graphs , composite structures and their design etc.

In this presented work my area of research was related to design and finite element analysis. It was really necessary to look for those ideas and papers which are relevant to my topic. Lot of and lots of researchers have done work in finite element analysis of pressure vessels. Therefore one needs to identify the design approach while carrying literature survey. Design approach for my work was based on ASME codes usage and applications.

Therefore I further constrained my literature survey to finite element analysis of pressure vessels designed initially by using ASME codes. This approach to literature survey was really pin pointed to my area of work. The most relevant research works are described in the following paragraphs along with author names and publication records. These papers are only studied for increasing knowledge level and original research is carried out keeping in view the project application and constraints. With the term constraints it should be clearified that these are the design constraints like temperature , pressure , geometric requirements , engineering material specifications and features like openings etc.

3.2 OVERVIEW OF KEY RESEARCH PAPERS STUDY

During the literature survey I studied number of papers but here only few are described briefly which were most relevant.

i) DESIGN AND STRUCTURAL ANALYSIS OF PRESSURE VESSEL DUE TO CHANGE OF NOZZLE LOCATION AND SHELL THICKNESS

Year of Publication:- 2012

Author:- Shaik Abdul Lathuef and K.Chandra sekhar

Brief Review :-

This paper describes ASME code design of pressure vessels and the results which are obtained for design by using finite element analysis approach. The author takes into account several design cases and analyses them and makes comparison. The nozzle locations are also varied and the effect of changing the nozzle locations has been depicted in terms of stress values. This paper also proves this fact and author has made good use of ANSYS which is general purpose finite element modeling package for numerically solving a wide variety of mechanical problems.

Relevance to my Research work:-

Structural failure has been examined by changing the nozzle locations of pressure vessel. These stress values are gradually reduced as the nozzles are shifted along the length of pressure vessel. The crux of this paper lies in the reduction of overdesign aspect of pressure vessel and this is the main frame of my research work as well. Therefore studying this research paper was really helpful in setting my objective.

ii) DESIGN OF PRESSURE VESSEL USING ASME CODE, SECTION VIII, DIVISION 1

Year of Publication:- 2012

Author:- B.S.Thakkr and S.A.Thakkar

Brief Review :-

Due to high pressure rise in vessel the structure has to bear severe high level of stresses. Thus proper design of pressure vessels is very essential. In this research work pressure vessel has been designed using ASME Section VIII. The author has used following sub-sections of ASME to reach his design:-

i) UG-4 General Materials
ii) UG-27 (C) Cylindrical Shells
iii) UG-32 (F) Ellipsoidal Heads
iv) UG-66 Shell Thickness.
v) UG-45 Nozzle Neck Thickness

Moreover the author has also done 3-D modeling of designed components and has included these 3-D designs in his research work. These modeled parts make a good connection between the ASME coded geometry and use of softwares like PRO-E which is widely used for CAD modeling all around the globe.

Relevance to my Research work:-

This research paper demonstrates good use of ASME codes to reach a final design for a particular application. The basic idea of using the sub-sections given in section Viii of pressure vessel codes get really clear after studying this research paper. Since in my research work I am going to design as per ASME section Viii therefore studying this research work was a privilege.

iii) DESIGN AND ANALYSIS OF PRESSURE VESSELS AMALGAMATING SELECTION OF MATERIALS USED IN MARINE APPLICATIONS .

Year of Publication:- 2013

Author:- Nitinchandra R. Patel

Brief Review:-

This paper describes the analysis of pressure vessels keeping in view the stresses induced in different materials. Pressure vessel with same design has been analyzed keeping in view different materials. Pressure vessel under consideration is for under water application from sea surface. Thickness of pressure vessels has been found out analytically for specific operating conditions. The author gives approximation of better material for pressure vessel for sub marine application. Comparison of seven different types of materials is done based on the stress values along with thickness values for seven different materials. These materials are as follows:-

MATERIAL	YEILD STRENGTH (N/mm2)
IS 2002-1962 1	200
IS 2002-1962 2A	205
IS 2002-1962 2B	255
IS 2041-1962 20M055	275
IS 1570-1961 ISCr90M 055	290
A242(Cr-Si-Cu-Ni-P type)	289.57
A588 (Mn-Cr- Cu-V type)	317.15

The beat material amongst the compared ones is **A588 (Mn-Cr- Cu-V type**) for marine applications and the most suitable head type is semi ellipsoidal head. Semi ellipsoidal heads are easily manufacture, thus using semi-ellipsoidal heads with thin walled cylindrical shells is an efficient design because it gives ease in fabrication also.

Relevance to my Research work:-

In this research work a good concept comes for designing head of pressure vessels in case of thin cylinders. The author has made very good use of different design parameters by manipulating values to get to the final in depth design and achieve the required conditions. Since my research work is also based on design of thin walled pressure vessels therefore study of this research paper was very helpful to get insight into design parameters. Also the design of semi ellipsoidal heads is relevant to my research work as well.

iv) DESIGN AND ANALYSIS OF MULTI LAYER HIGH PRESSURE VESSELS.

Year of Publication :- 2013

Author:- Siva Krishna Raparla

Brief Review:-

Designing of multi layer pressure vessels has been done. Analysis of the designed vessel is performed .Various features of multi layered vessels are discussed and their advantages over mono block vessels are described. Various components of solid pressure vessel are also designed by using ASME codes as per the principles specified. Comparison of stress values has been made between solid pressure vessel and multi layered vessels. ANSYS has been used as finite element package to compute the stress values. The author describes that 26.02 % saving of material is possible by the use of multi layered vessel. Hence the overall weight of pressure vessel is reduced .This in turn impacts better manufacturing and economic cost of fabrication. Layered structures with more strength thus benefit both manufacturing and cost effects. This is one of the main aspects of designer to keep the weight and cost as low as possible.

Relevance to my Research work:-

Here again use of ANSYS and its effectiveness comes into play for verifying the design. Also ASME codes and their importance in the design of pressure vessels cannot be ignored. Since my area of research work will include ASME codes and ANSYS therefore this paper is quite relevant to my area of work.

v) FATIGUE DESIGN AND VALIDATION OF AN ASME-CODED PRESSURE VESSEL

Year of Publication:- 2009

Author:- R. A. Pasha, N. A. Choudary and M.A. Nasir

Brief Review:-

Fatigue life of real sized pressure vessel is estimated. The pressure vessel under study is subjected to internal pressure. Pressure vessel has been designed according to ASME Section VIII Div II of boiler and pressure vessel code . ASME codes give guide lines for fatigue life assessment but the predicted fatigue life is not exact according to the author. The reason behind is that ASME codes provide assessments of fatigue life for individual parts and not for complete assembly of pressure vessel.

Relevance to my Research work:-

This research focuses to discuss design parameters for complete assembly of pressure vessels therefore it is very helpful for my research work because I also check the final design parameters at the assembly level. Also ASME code section Viii is to be used in my research work which is again a similarity with this research paper.

Chapter 4

DESIGN CALCULATIONS AS PER ASME CODES FOR PRESSURE VESSEL

4.1 Design Parameters

The most common design parameters used for pressure vessel are as follows:-

i) Type of Pressure

Before designing any Pressure vessel it should be known to the designer that whether internal or external pressure is applied on the vessel. Its magnitude is also to be confirmed depending on the application. In this research work the pressure vessel is designed for internal pressure having magnitude 9.6 MPa

ii) Operating Temperature

The temperature for the pressure vessel on which it operates is also an important design parameter. The temperature for designing the pressure vessel plays a vital role in design calculations because it effects the structure of vessel. Therefore the pressure vessel should be designed for maximum operating temperature. In this research work pressure vessel is designed for temperature of 300 degree celsius.

iii) Shell Thickness

The minimum thickness which can endure the required temperature and pressure is the shell thickness. Thickness of shell is a very important design parameter because this part has to bear stresses majorly hoop stresses and longitudinal stresses.

Hoop Stress:-

Hoop stresses are those which act along the circumference of the pressure vessel. Mathematically ,

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

Where

“p” stands for pressure

“d” stands for diameter of cylinder

“t” stands for thickness of cylinder

Longitudnal Stress :-

Longitudnal stresses are those which act along the axis of cylinder .

Mathematically ,

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

Following figure shows the distribution of hoop stresses and longitudinal stresses.

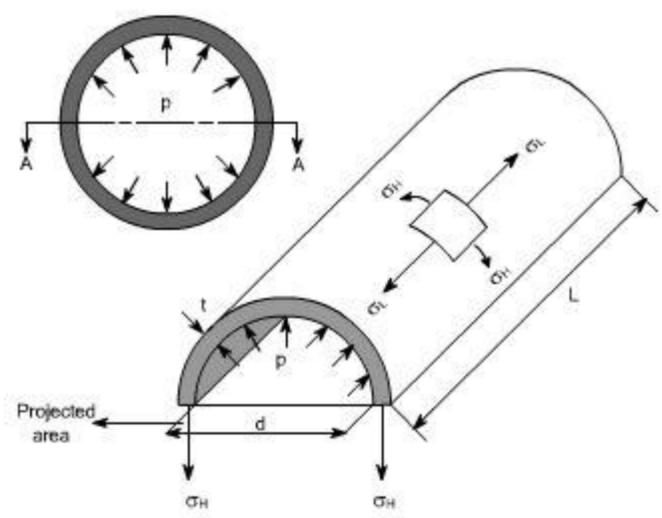


Fig 10 : Hoop Stresses and Longitudnal Stresses

iv) <u>Head Type :-</u>

In case of cylindrical shells the type of heads are either elliptical , doughnut shaped or dished heads. In this research work the head type chosen is 2:1 elliptical head.

Some other design parameters include length of pressure vessel, nozzle opening and thickness of heads. The design parameters used in this research work are given below in tabular form.

DESIGN PARAMETERS AS PER ASME SECTION VIII DIV-1		
Description	Values Initial Input	ASME Code Reference
Material	SA-723 Yield Point=1241 Mpa	ASME Sec II-Part D
Design Pressure	96 bar (constraint)	
Design Temperature	572 Degree Farenheit (constraint)	
Maximum Allowable Stress Value for SA-723 at 572 degree farenheit	54300 psi	ASME Sec II-Part D
Length of Vessel	3000 mm (constraint)	
Diameter of Vessel	540 mm (constraint)	
Weld Joint Efficiency	0.7	ASME Section VIII UW-12
Thickness of Shell	11.3 mm	ASME Section VIII UG-27©
Type of Head	2:1 Elliptical Head (constraint)	
Thickness of Head	11 mm	ASME Section VIII UG-32(d)
Nozzle Opening	80 mm (constraint)	ASME Section VIII UG-45

Table 4 Design Parameters for Pressure Vessel

It is important mentioning here an output parameter of strain intensity. ANSYS simulation and analysis package computes the values of strain intensities and gives results. On the back ground ANSYS computes the differences between three strain values along three planes X,Y and Z. The maximum of these differences is divided by the maximum strain value and then multiplied by 100 to give the percentage strain intensity.

4.2 Material Selection

The choice of material for any design is a complicated process. This is mainly because of large availablility of variety of materials . Being mechanical engineers we mostly deal with metals . Sometimes we benefit from the forgiving nature of metals and we can take margin in certain design parameters while selecting material.

However the proper usage and selection of material is a decisive process and requires early consideration. Different material selection charts are available from many sources which help as guide lines to select materials. But the key to proper material selection is that the designer should know the basic function of its component. Generally the material selection can be segregated into following four steps.

i) Translation
ii) Screening
iii) Ranking
iv) Support info

Each of these four steps are briefly described below .

i) Translation

Material selection is primarily based on type of application of pressure vessel. Designers ensure the proper usage of material. This in turn is dependent on the design parameters. Usually for high temperature and pressure applications those materials are selected having high yield strength. For low temperature applications there can be certain other characteristics other than yield which are important. For example for under water applications of pressure vessels designer must take in account that the material is corrosion resistant. So this is based on the final objective of the application.

ii) Screening

Once the basic criteria for selecting material is specified then comes screening of material. By screening it is usually referred that the material should be of proper standard available. There are many different standards of materials all round the globe like ISO , ASME , BRITISH ,DIN , CHINESE standards etc. Manufacturers who supply materials make sure that the supplied order should be according to standard. American society of mechanical engineers has given list of materials in ASME section II . These materials are for boiler and pressure vessels and these materials can be used for any design done in accordance with ASME Section VIII.

iii) Ranking

After establishing the criteria for material selection and choice of proper standards the next step is ranking of material. Material standards are very diverse and the list of available materials is very long usually. Some of these material can serve the purpose of any particular application. Designers have to decide at this stage that which material they should go for .Lots of factors like cost , availability , time constraints , transportation ,procurement , production rate etc. all come into play at this stage of material selection. The most common among these is cost factor and delivery on time. Hence it can be readily said that material selection has three considerations including material properties ,manufacturing , geometry of product and above all is product functioning.

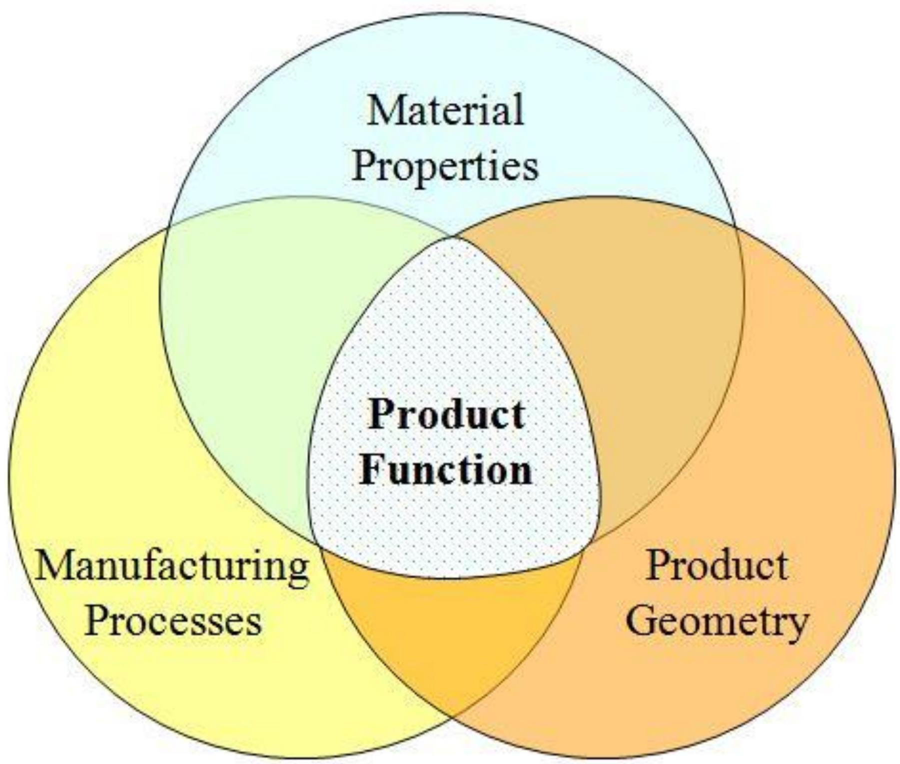


Fig 11 Material Selection

iv) Support Info

This is the last step in material selection process but it is of great importance .It again involves use of standards . Because of continuous research and development in the field of material sciences revisions of standards and manuals appear every year or after two years. So the technical staff in any organization keeps an eye on the support info of these used materials in their organizations. Sometimes materials are replaced by others having some better properties .Then the technical staff has to forecast the change in production schedule in their organization. So those materials of proper standards like ASME standard have proper technical support info. The following figure summarizes our discussion on material selection as below .

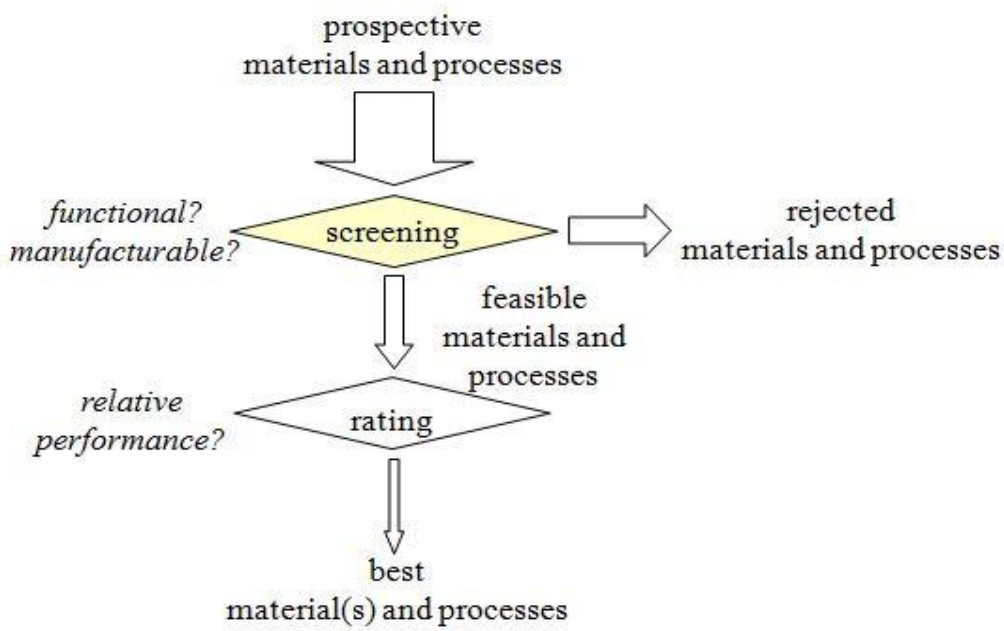


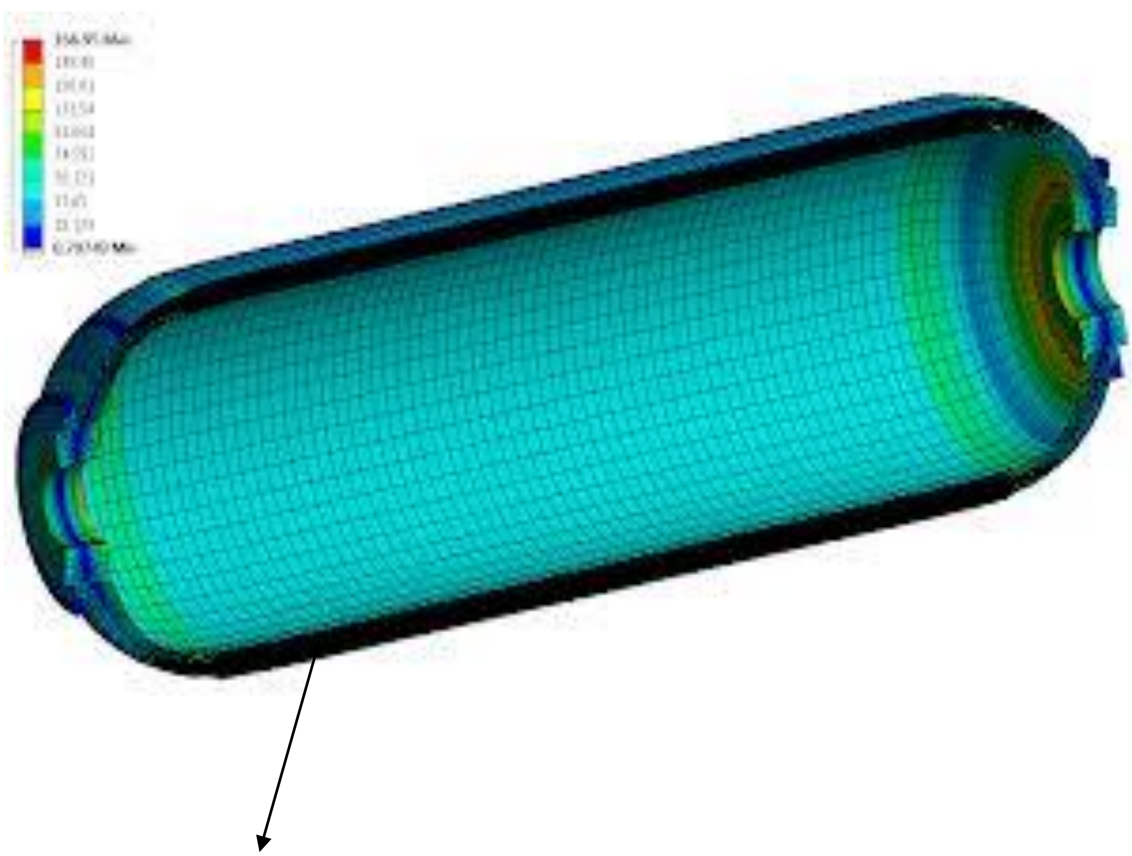
Fig 12 Material Selection Steps

In the presented research work the material chosen is SA-723.This material has been choosen based on high yield criteria and ease of availability. The material belongs to ASME standard .

The material selection chart can be reffered from ASME Section II part D, Table 1-A . This material has yield strength of 1241 MPa at room temperature. Maximum allowable stress value is 380 MPa (i.e.54300 psi) at operating temperature of 572 degree farenheit.

4.3 Shell Thickness Calculation

Thickness of shell is very important in pressure vessel designing. Following figure depicts thickness of shell precisely.



Shell Thickness

Fig 13   Shell Thickness

Thickness of shell is calculated using ASME Section VIII ,Division I .The sub section that is used for calculating shell thickness is UG-27. The formula for calculation is as follows :-

$$t = \frac{PR}{SE - 0.6P}$$

Where

"P" stands for pressure

"R" stands for radius

"S" stands for maximum allowable stress value

"E" stands for weld joint efficiency

One more parameter is corrosion allowance added to this calculated value denoted by C.A .

$$t = \frac{PR}{SE - 0.6P} = \frac{(1392)(10.24)}{(54300)(0.85) - (0.6)(1392)}$$

$$t = \frac{14254}{46155 - 835.2} = 0.314\ inch = 7.98\ mm = 8\ mm$$

$$Corrosion\ Allowance = \frac{1}{6}(\ t) = 1.33$$

$$Total\ Shell\ Thickness = 8 + 1.33 = 9.3\ mm$$

4.4 Head Thickness Calculation

Thickness of head is calculated by using ASME Section VIII , Division I. The basic formula used for the head thickness is given in sub section UG-32. A more precise formula for ellipsoidal heads thickness that also takes into account stress concentration factor is given in Mandatory Appendix I (1-4).

Thickness formula is as given below :-

$$t = \frac{PDK}{2SE - 0.2P}$$

Where

"t" stands for thickness

"P" stands for pressure

"D" stands for diameter

"S" stands for maximum allowable stress value

"E" stands for weld joint efficiency

"K" stands for stress concentration factor

After putting the values calculation is given below ,

$$t = \frac{PDK}{2SE - 0.2P}$$

$$K = \frac{1}{6}[\ 2 + (\frac{D}{2h})^2]$$

Where K is the stress concentration factor given in ASME codes

$$K = \frac{1}{6}[2 + (\frac{520}{2 * 130})^2] = 1$$

$$t = \frac{(1392)(20.4)(1)}{2(54300)(0.85) - (0.2)(1392)} = \frac{28396.8}{92031.6} = 0.308\ inch = 7.83\ mm\ \approx 7.9\ mm$$

$$Corrosion\ Allowance = \frac{1}{6} * t = 1.3$$

Total Head Thickness =7.9 + 1.3 = 9.2 mm

Chapter 5

3-D MODELING USING CREO

5.1 Part Modeling

In Part Modeling we can create a part from a conceptual sketch through solid feature-based modeling, as well as build and modify parts through direct and intuitive graphical manipulation. The Part Modeling Help introduces to the terminology, basic design concepts, and procedures that one must know before uilding a part. Part Modeling shows how to draft a 2D conceptual layout, create precise geometry using basic geometric entities, and dimension and constrain your geometry. One can learn how to build a 3D parametric part from a 2D sketch by combining basic and advanced features, such as extrusions, sweeps, cuts, holes, slots, and rounds. Finally, part modeling provides procedures for modifying part features and resolving failures.

To plan any design, one needs to have a basic understanding of model from a broad perspective. In other words, understand the overall function, form, and fit of the product. This understanding includes the following points:

- Over all Size
- Basic Model Characteristics
- Way to model part
- Approximate amount of components assembly would contain
- Manufacturing technique for part

Fig 14: Part Modeling

5.2 Design Approaches

Even the best plans are imperfect. However, one can eliminate many future modeling issues by thinking out the model before starting design. The following two design approaches can help in determining the planning strategy:

- **Top Down Design**—Analyze your product from the finished product and work down. So, begin with the master assembly and break it down into assemblies and subassemblies. Then, identify the main assembly components and their key features. Finally, understand the relationships within and between assemblies, and assess how the product will be assembled. With this information, one can plan a design and leverage overall design intent into the models. Top down design is the industry paradigm for companies that design products that undergo frequent design modifications or for those companies that design diverse products.

- **Bottom Up Design**—In this approach analyze your product from the component level and work up to the master assembly. Note that successful bottom up design demands a basic understanding of the master assembly. Designs based on the bottom up approach do not fully leverage design intent. Even though the end result can be the same as using top down design, but one increase the risk for design conflicts and errors that result in a less flexible design. Bottom up design remains the most used paradigm in the design industry today. Companies that design similar products or products that do not demand frequent modifications during their life cycle use bottom up design approach.

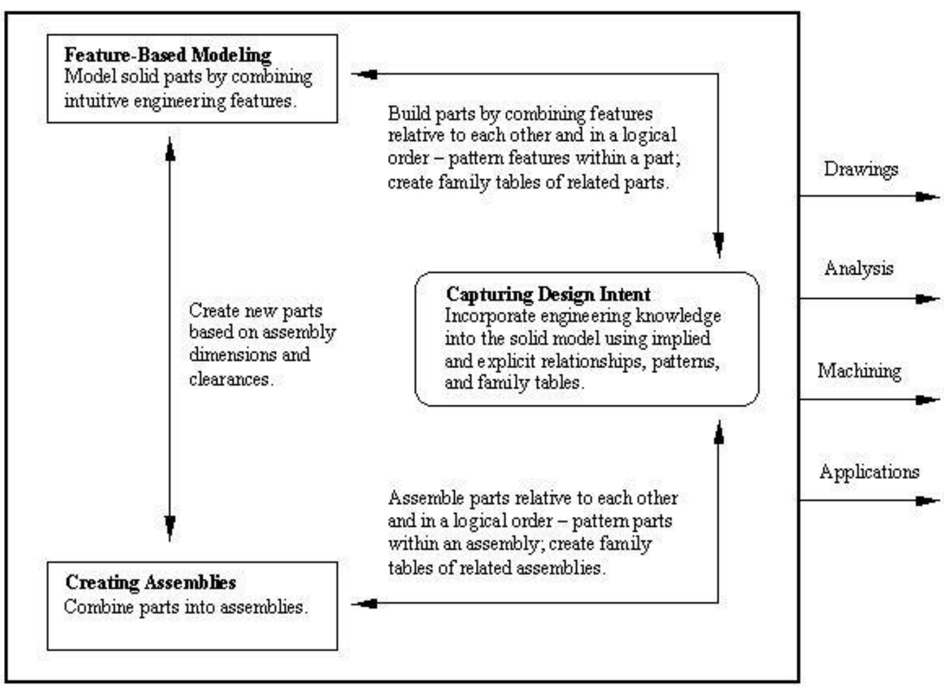


Fig 15: Part Modeling Process

5.3 Sketcher OverView

The sketcher describes how to sketch section geometry, configure a sketch, create references to dimension and constrain the sketch geometry, and other operations that can be performed with a sketch. Main engineering features majorly require sketcher tool to form parts. To dimension and constrain geometry, Creo requires to create references. References can be created through the references dialog box. To open the references dialog box, click Sketch ▶ References.

Creo Elements/Pro prompts to create references in the following situations:

- When we create a new feature, the References dialog box opens. Creo Elements/Pro prompts user to select a perpendicular surface, edge, intent edge vertex, datum reference or composite curve relative to which the section will be dimensioned and constrained.
- When we redefine a feature that has missing references.
- When we do not have enough references to place a section.

The Sketcher toolbar and the Sketch menu contain tools to create the following entities:

- Lines
- Quadrilaterals
- Circles
- Ellipses
- Arcs
- Fillets
- Chamfers
- Splines
- Points

A constraint is a condition defining the geometry of the entity or a relationship among entities. Constraints can refer to geometry entities or construction entities. Create constraints or accept the constraints offered as we draw sketch. One can select an existing constraint, delete it, or get more information about it. The constraint tools work in continue mode. Some constraints can be applied to a single entity, and other constraints can be applied to pairs or groups of entities.

Number of Entities	Applicable Constraints
A single entity	Vertical Horizontal
Pairs of entities	Perpendicular Tangent Midpoint Coincident Mirror Equal Parallel
Three or more entities	Equal Parallel

Fig 16: Relation between constraints and entities

The following figure illustrates types of available constraints in CREO. Symbols used for each constraint type are also shown.

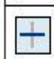
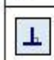
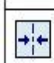
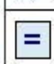
Button	Constraint
	Makes a line or two vertices vertical.
	Makes a line or two vertices horizontal.
	Makes two entities perpendicular.
	Makes two entities tangent.
	Places a point on the middle of a line or an arc.
	Makes points coincident.
	Makes two points or vertices symmetric about a centerline.
	Creates equal linear or angular dimensions, equal curvature, or equal radii.
	Makes two or more lines parallel.

Fig 17: Types of sketcher constraints

5.4 Assembly Modeling

The Assembly Help describes the procedure of assembling parts created in Creo Elements/Pro Part mode. It offers information about working with assembled models and managing assembly projects. When you place a component relative to its neighbors (components or assembly features), its position is updated as its neighbors move or change, provided the assembly constraints are not violated. This is called parametric assembly.

To add a component to an assembly, click or click Insert ▶ Component ▶ Assemble. Component placement is based on placement definition sets. These sets determine how and where the component relates to the assembly. The sets are either user-defined or predefined.

Placement of a component in an assembly is determined by the constraints in all sets defined. A single set of constraints can define placement of a component. If constraints from one set conflict with constraints from another set, the placement status becomes invalid. The constraints must be redefined or removed until placement status becomes valid. Constraint sets are displayed in the Placement folder of the Model Tree. Display hierarchy follows the order in which they were defined. Constraint icons are the same as those in the Component Placement dashboard. If only one set is defined, only constraints appear. The Model Tree Placement filter must be activated to view the Placement folder. Component interfaces contain stored constraints or connections that are used to quickly place a component. After an interface is defined, it can be used whenever the component is placed in an assembly. Additional interface information, such as assembly conditions, can also be specified. Assembly constraints used in CREO are shown below :-

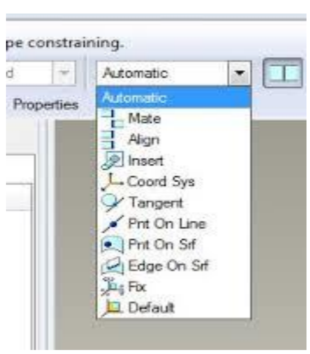


Fig 18: Types of Assembly Constraints

5.5 Surface Modeling

Surfacing lets you create and manipulate surfaces and curves on a model, including the ability to manipulate curve tangents directly on the screen. Surfacing area helps about creating, changing, and manipulating surfaces. Surfacing shows you how to use surfacing applications to work with the curves and surfaces of parametric, nonparametric, and imported models. In addition, how to alter the form and shape of curves, quilts, facets, and solids by transforming, scaling, rotating, stretching, tapering, bending, or twisting the geometry. Surfacing is an advanced modeling technique used to model objects of free form shapes.

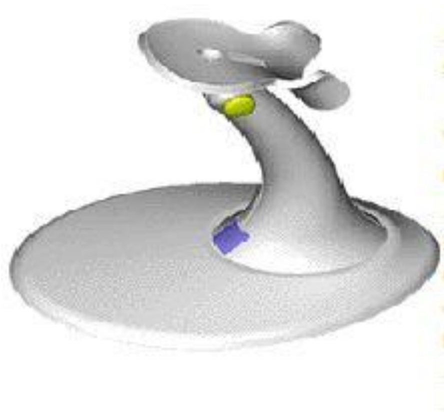


Fig 19 : Surface Modeling

5.6 Data Exchange

Through Data Exchange functionality one can share data from many other CAD/CAM systems. Certain CAD formats, such as CATIA, even allow for update of 3D imported data to changes in the source. Mostly data is exchanged in IGES format for analysis purpose . Parasolid file format is often used also for exchanging assembly data files with analysis software packages. Moreover CPLC files can alo be generated to exchange data with CAM source.

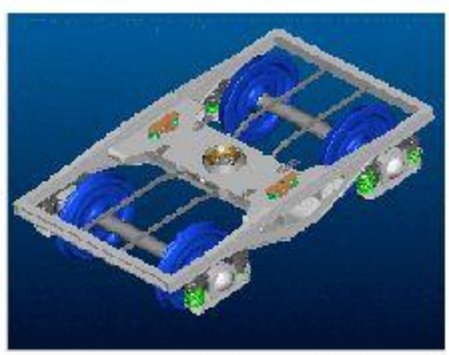


Fig 20: Data exchange Assembly file

5.7 Design Cases For Research Work

Four cases for pressure vessel design have been modelled for analysis purposes in this research work .These cases are as follows.

Case 1: - Thickness at assembly level 9.3 mm (ASME Code Case)

Case 2: - Thickness at assembly level 8.5 mm

Case 3: - Thickness at assembly level 8 mm

Case 4: - Thickness at assembly level 6 mm

3-D images for all above four cases are shown below along with their cross section images.

5.8 Case 1: - Thickness at assembly level 9.3 mm

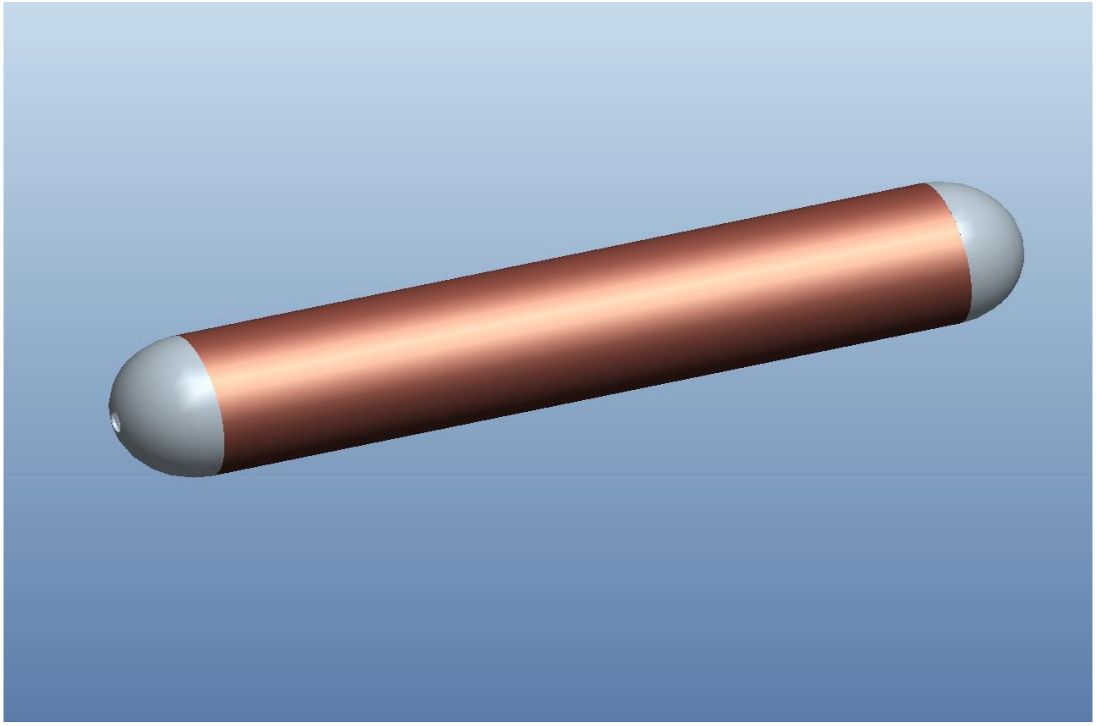


FIG 21- ASME BASE CASE MODEL 9.3 mm THICKNESS

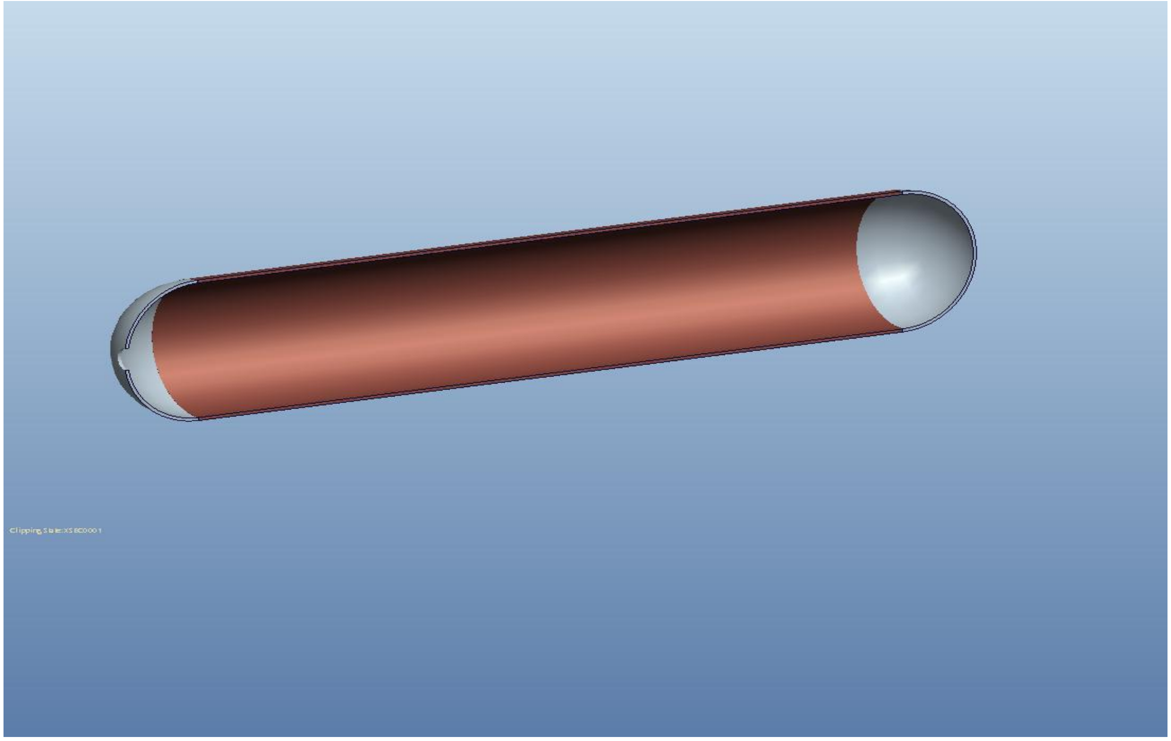


FIG 22- ASME BASE CASE MODEL 9.3 mm THICKNESS CROSS-SECTION

5.9 Case 2: - Thickness at assembly level 8.5 mm

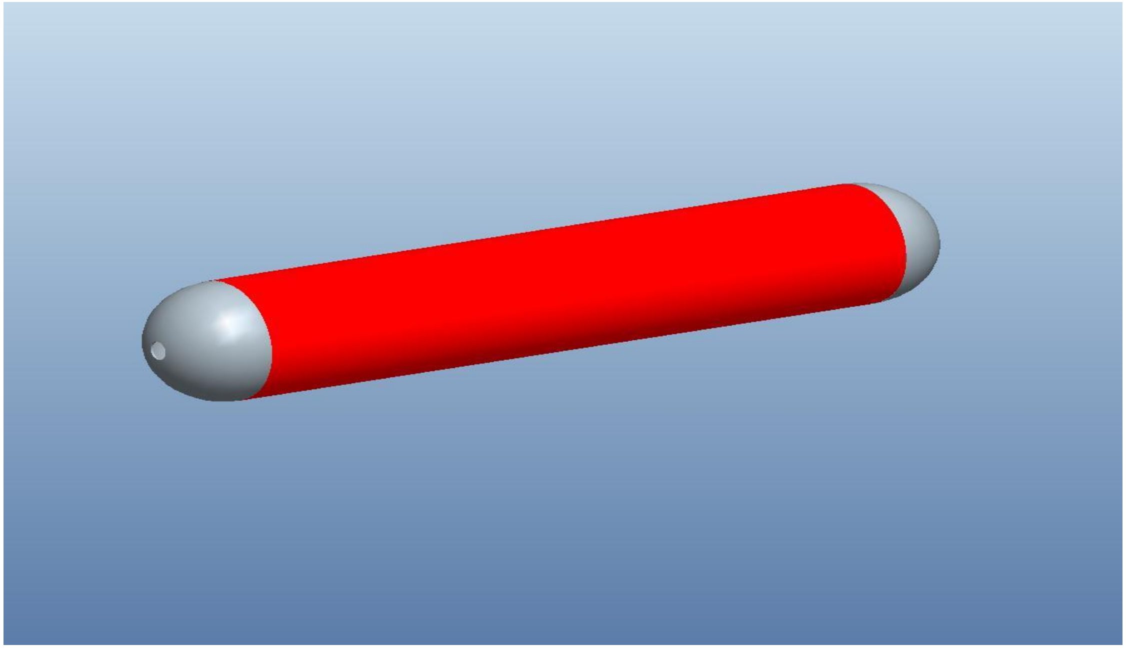


FIG-23 CASE 2 MODEL THICKNESS 8.5 mm

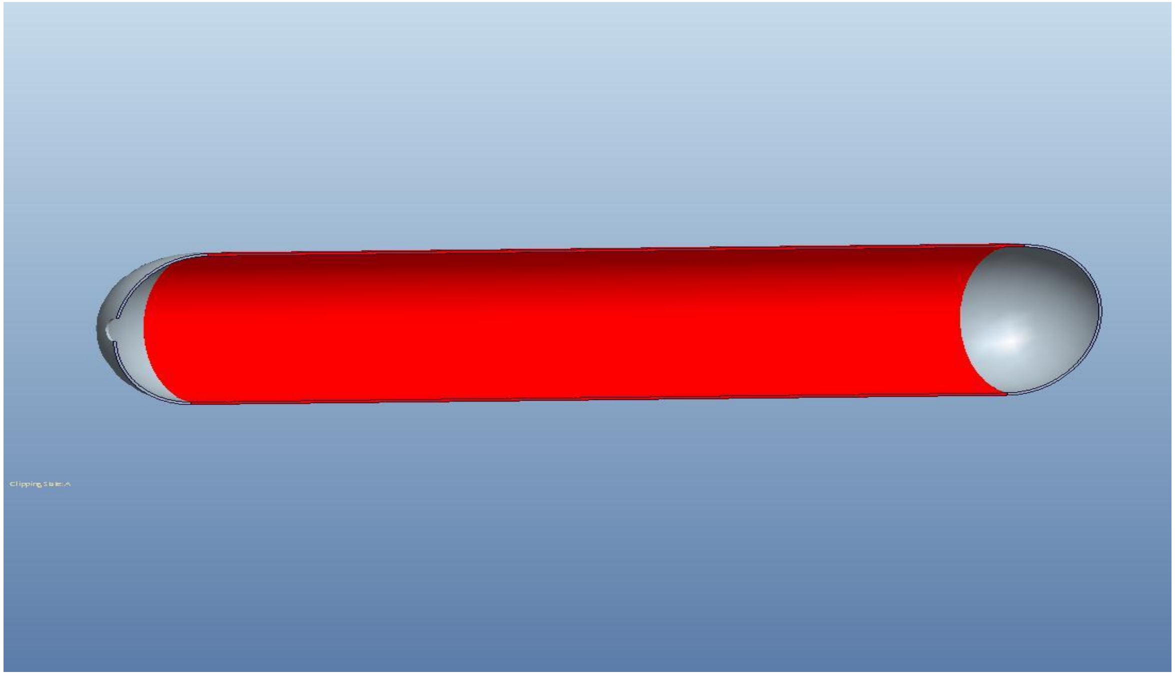


FIG-24 CASE 2 MODEL THICKNESS 8.5 mm CROSS SECTION

5.10 Case 3: - Thickness at assembly level 8 mm

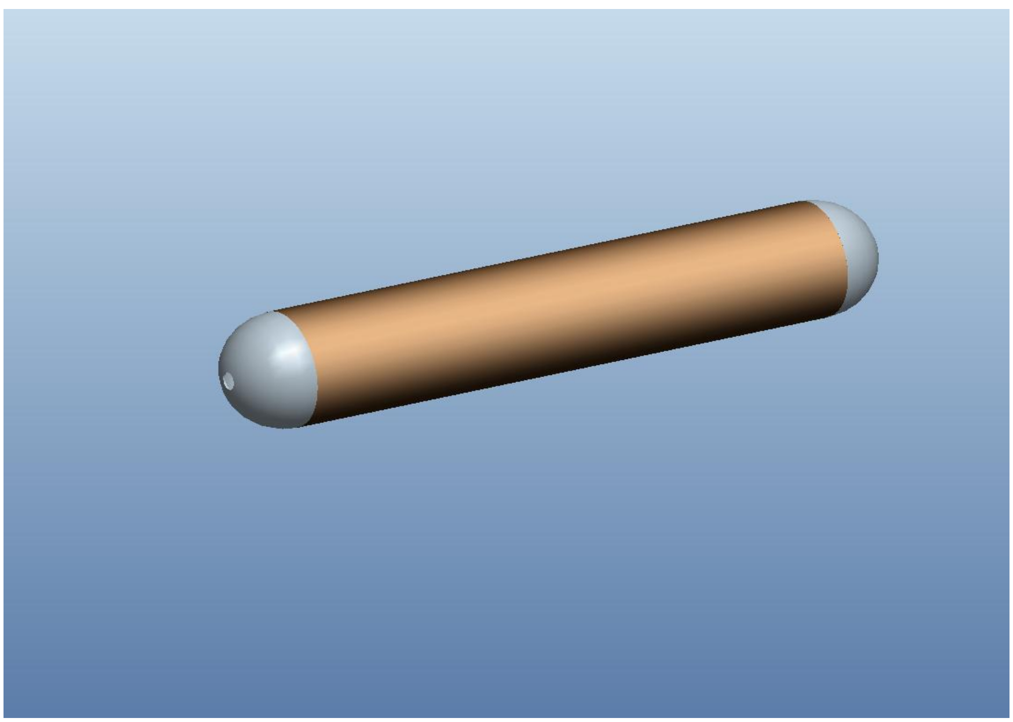


FIG-25 CASE 3 MODEL THICKNESS 8 mm

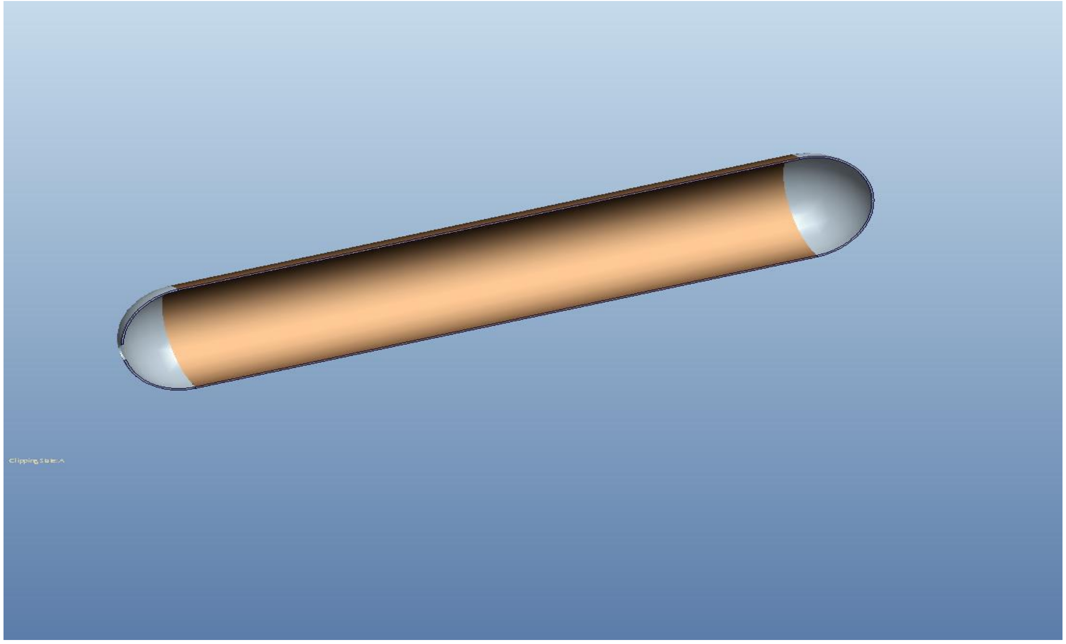


FIG-26 CASE 3 MODEL THICKNESS 8 mm CROSS SECTION

5.11 Case 4: - Thickness at assembly level 6 mm

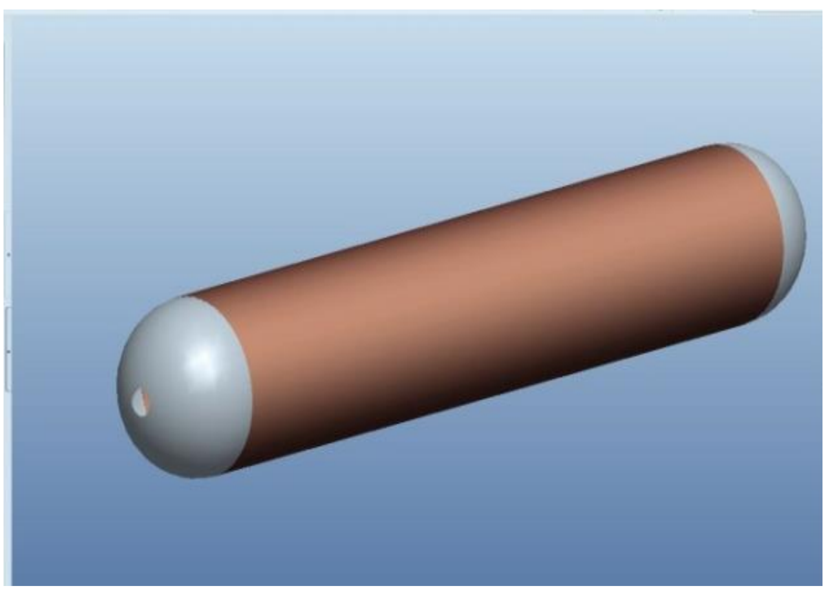


FIG-27 CASE 4 MODEL THICKNESS 6 mm

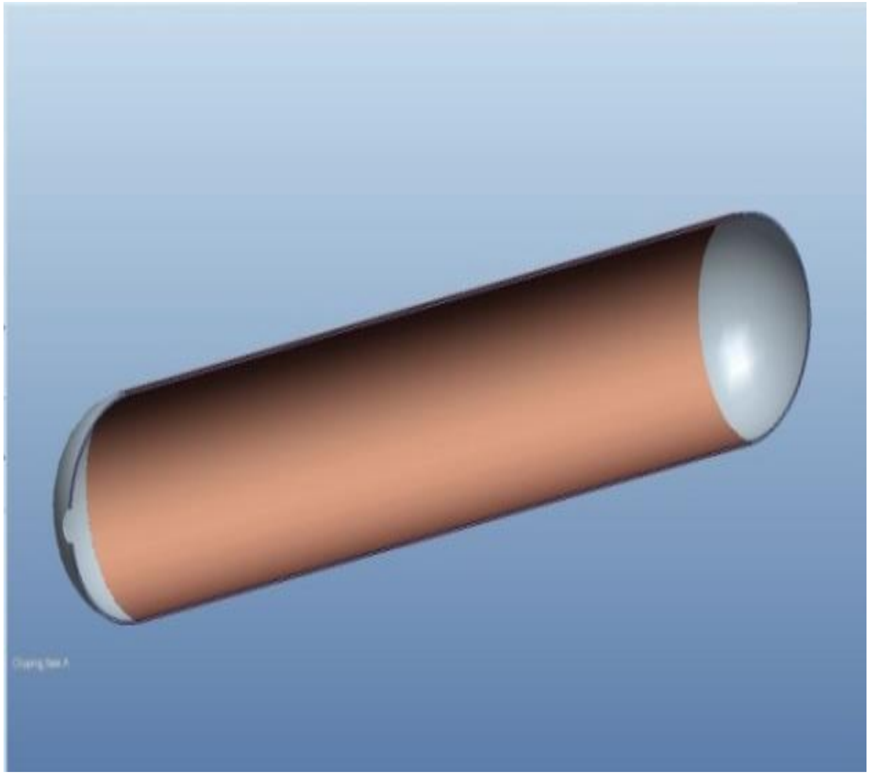


FIG-28   CASE 4 MODEL THICKNESS 6 mm CROSS SECTION

Chapter 6

ANALYSIS AND RESULTS USING ANSYS

6.1 Importing Geometry

ANSYS is a general purpose finite element analysis software package having wide variety of modules. Any sort of real scenarios can be created in this modern day tool for analysis purpose. Presently i have used ANSYS workbench 15.0 to do the analysis of pressure vessel. While doing analysis the first step is to import geometry of model. Model can be imported in various formats depending upon the software connection, compatibility and system requirements. Here the pressure vessel model has been imported in para solid (.xt) file format.

The 3-D assembly of the model needs to be first saved as para solid file in the system's directory. When static structural analysis is setup on ANSYS workbench , geometry of model is imported in para solid file format by browsing through system. When the file is picked up from system repository it is imported as solid file .

Fig 29 ANSYS Workbench Environment

6.2 ANSYS Design Modeler

After importing the geometry file , the next step is to enter ANSYS design modeller mode. The file which is imported looks as follows.

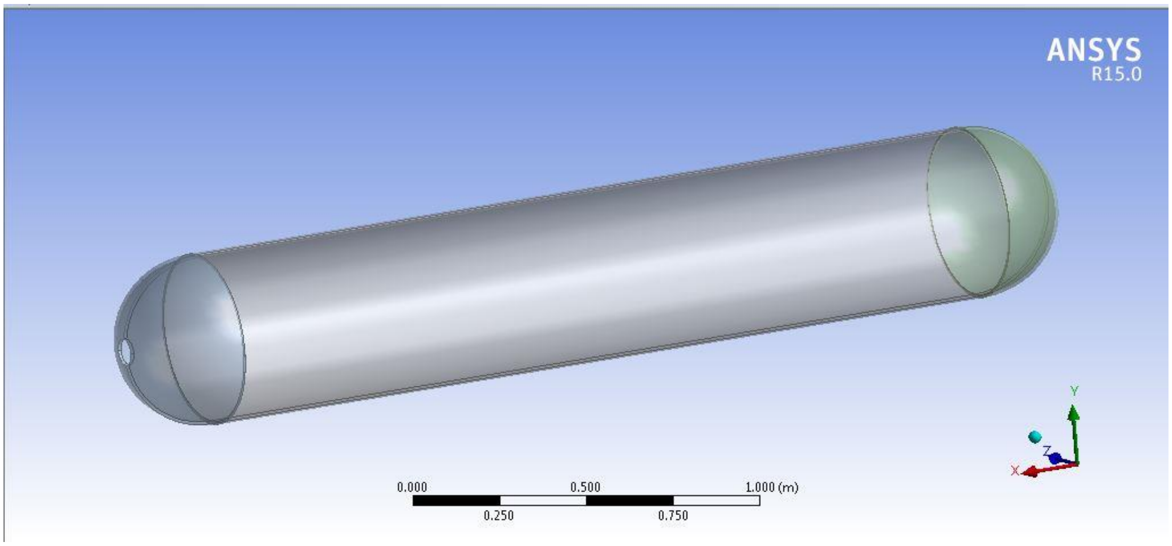


Fig 30: ANSYS Workbench Importing Geometry

The pressure vessel model is symmetric about planes and axis .Therefore it is preferable to apply symmetry conditions to the model. The symmetry is applied about three planes X,Y and Z . The symmetric slice of the pressure vessel model looks as follows:-

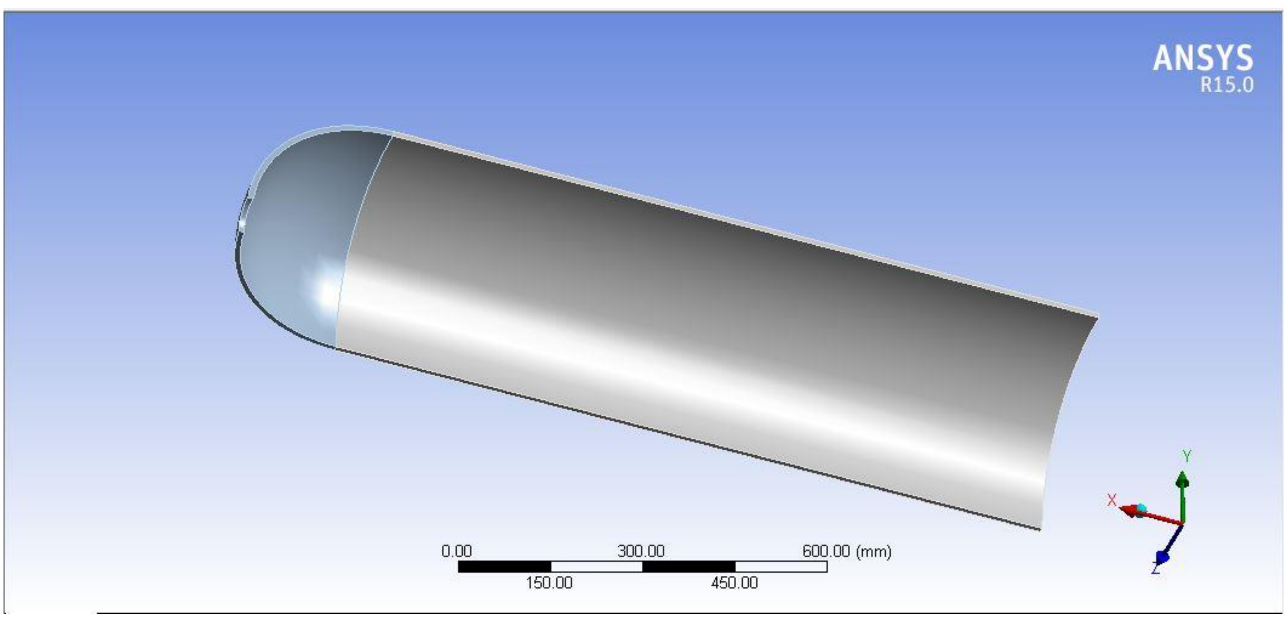


Fig 31: Symmetric Model

After applying symmetry next step is to run ANSYS workbench mechanical wizard. Simulation environment and conditions are generated in mechanical wizard.

6.3 Meshing

As we proceed for meshing the model, one thing needs to be assured that the solution converges on mesh size. This is known as mesh convergence .Appropriate way to achieve mesh convergence is to carry out solution for one case at different mesh sizes. As the final output which in our case are stress values become constant at one mesh size , the solution becomes constant and mesh convergence is achieved. Presently mesh convergence is achieved at mesh size 10 mm. This is shown in following graph.

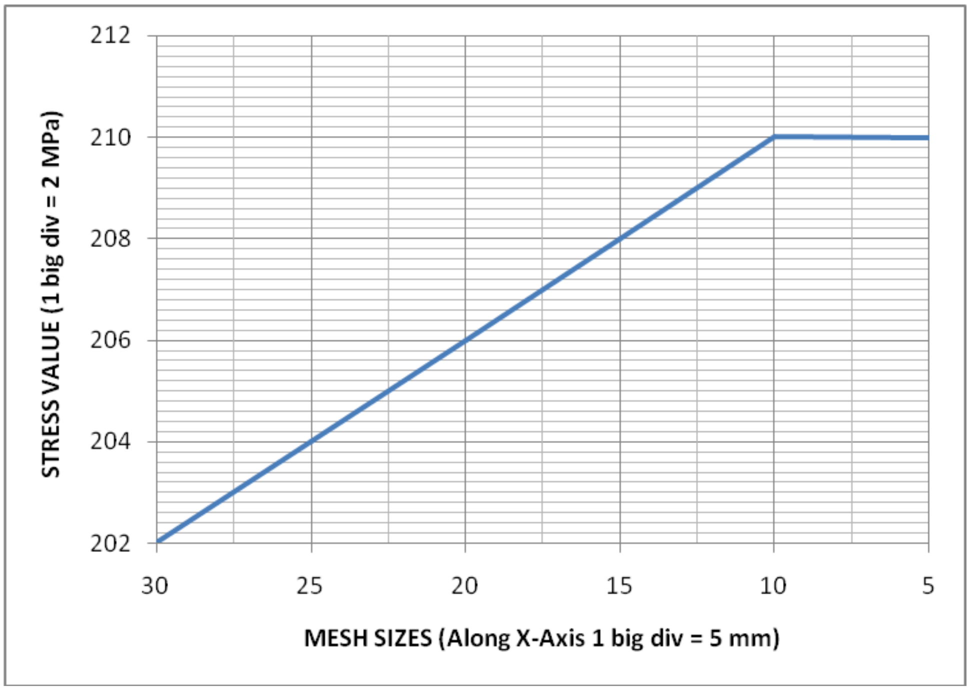


Fig 32: Mesh Convergence Graph

Meshed view of the model is shown in following figure.

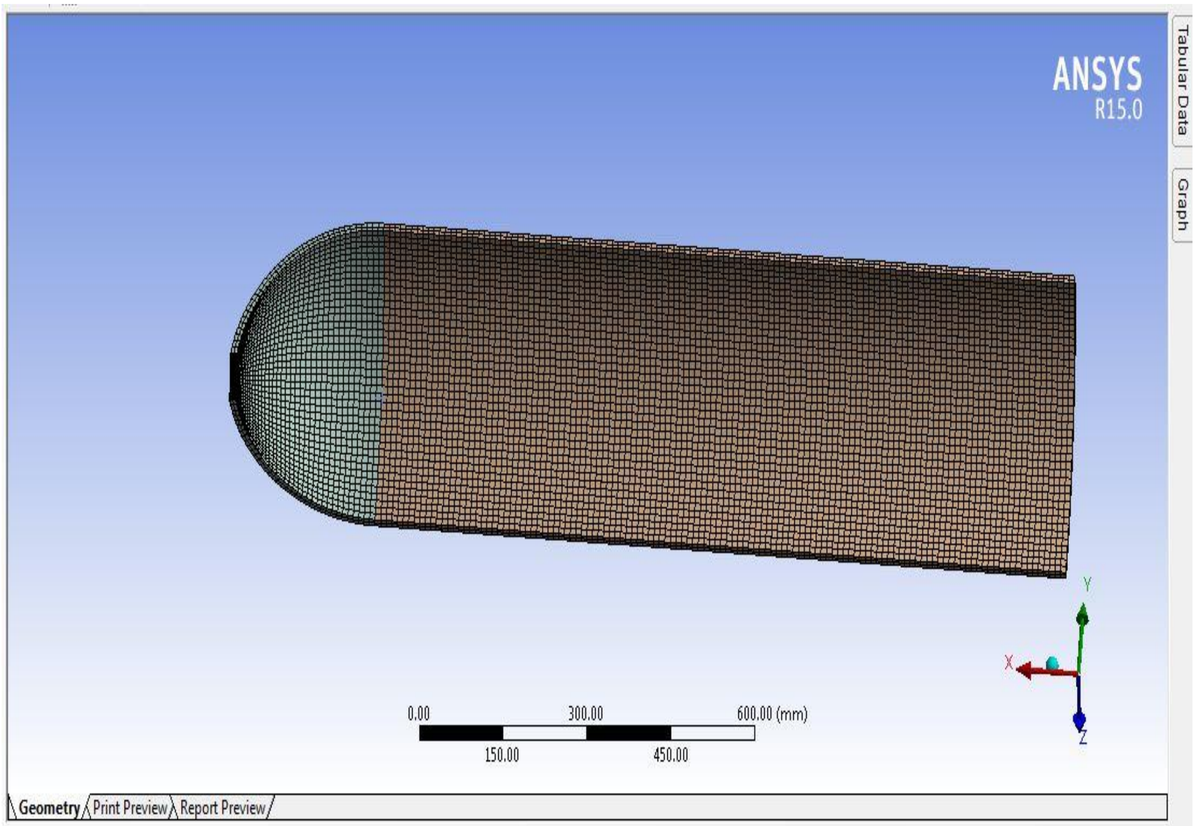


FIG 33 – MESHED VIEW OF MODEL

6.4 Applying Loads and Temperature

After doing meshing of model the next stage is to apply loads. Here two type of loads are applied to the pressure vessel.

- Pressure load having magnitude equal to 96 bars or 9.6 M Pa
- Temperature applied having magnitude 300 degrees centigrade.

Magnitude of pressure and temperature is our inital design constraint and the pressure vessel is designed against these initial conditions .Application of temperature and pressure is shown in the following figures.

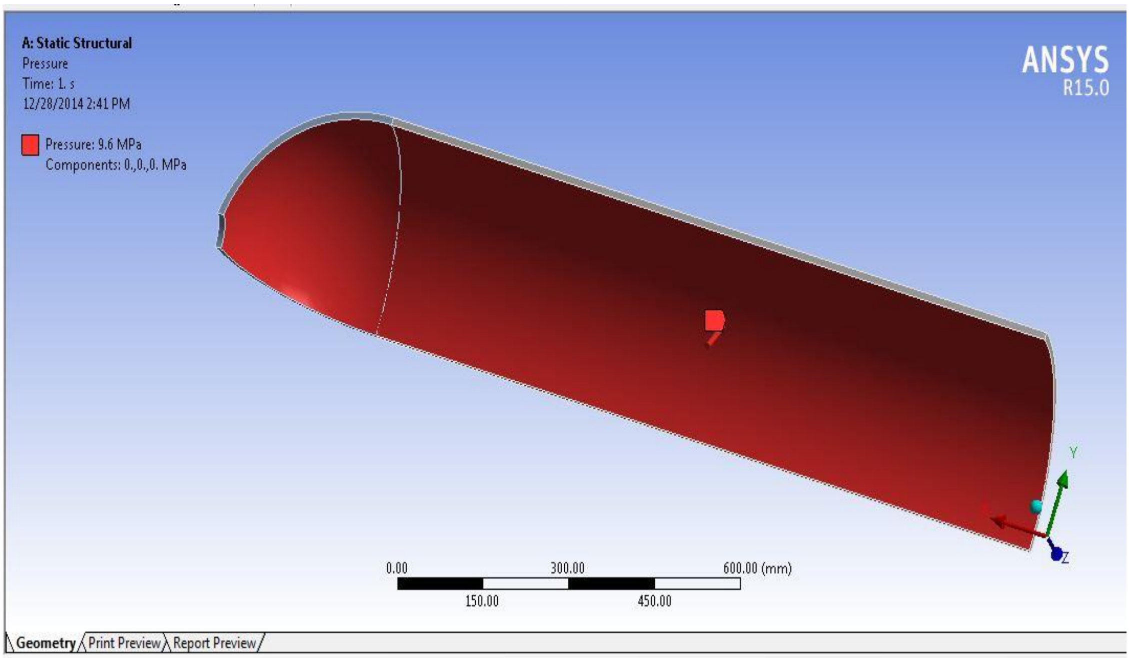


FIG 34: PRESSURE APPLIED

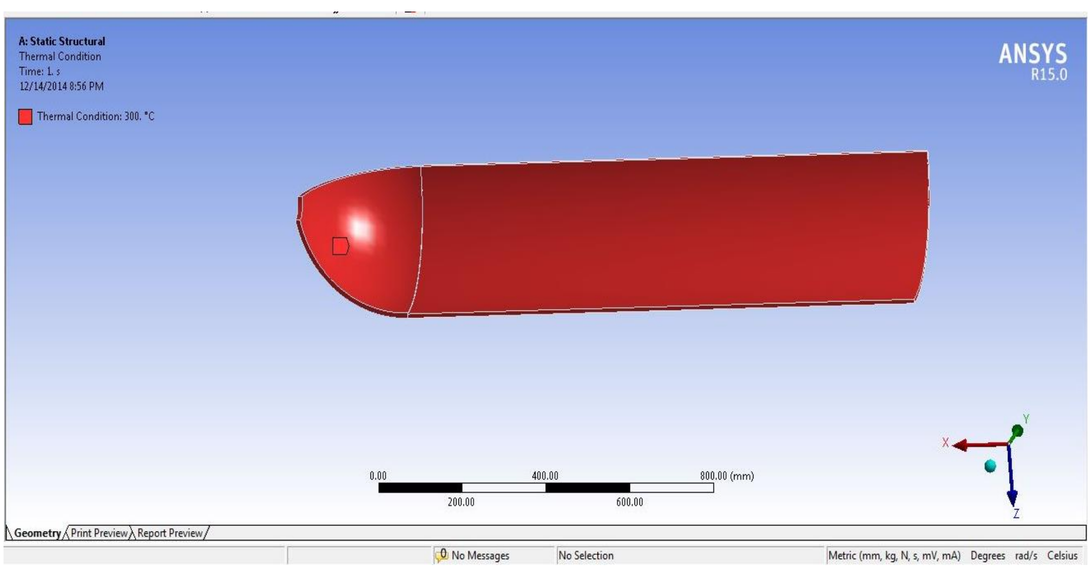


FIG 35: TEMPERATURE APPLIED

All four design cases have been analyzed and corresponding stress plots and strain plots are obtained. The stress values obtained are Von-Misses stresses because we are considering yield criteria. Strain plots are also shown for each design case. Strain values are shown because, if any strain value crosses 0.2 % threshold it will be considered as failure for this particular application. Stress and strain plots for each case are shown as under.

6.5 Case 1: - Thickness at assembly level 9..3 mm stress plot

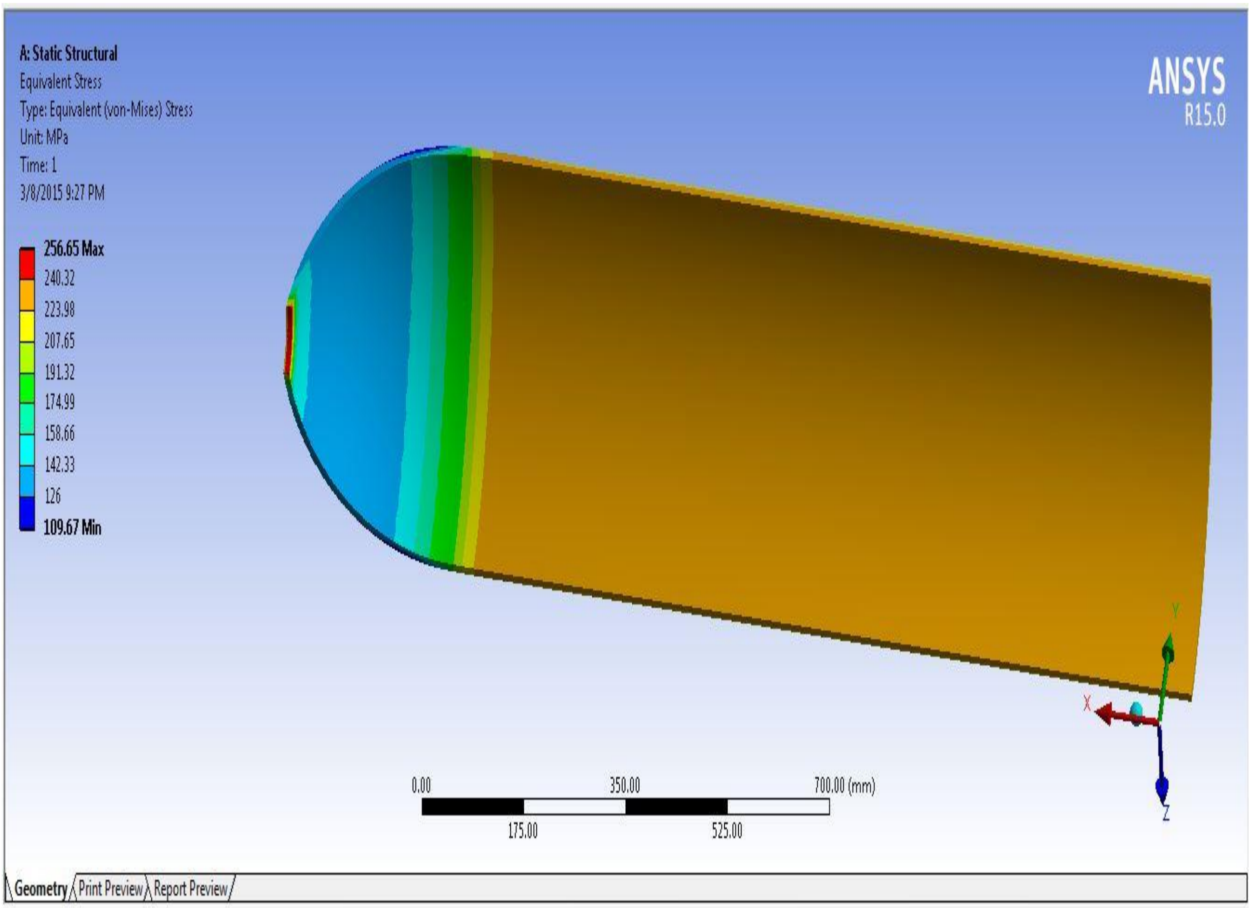


FIG 36: Stress plot at assembly for 9.3 mm

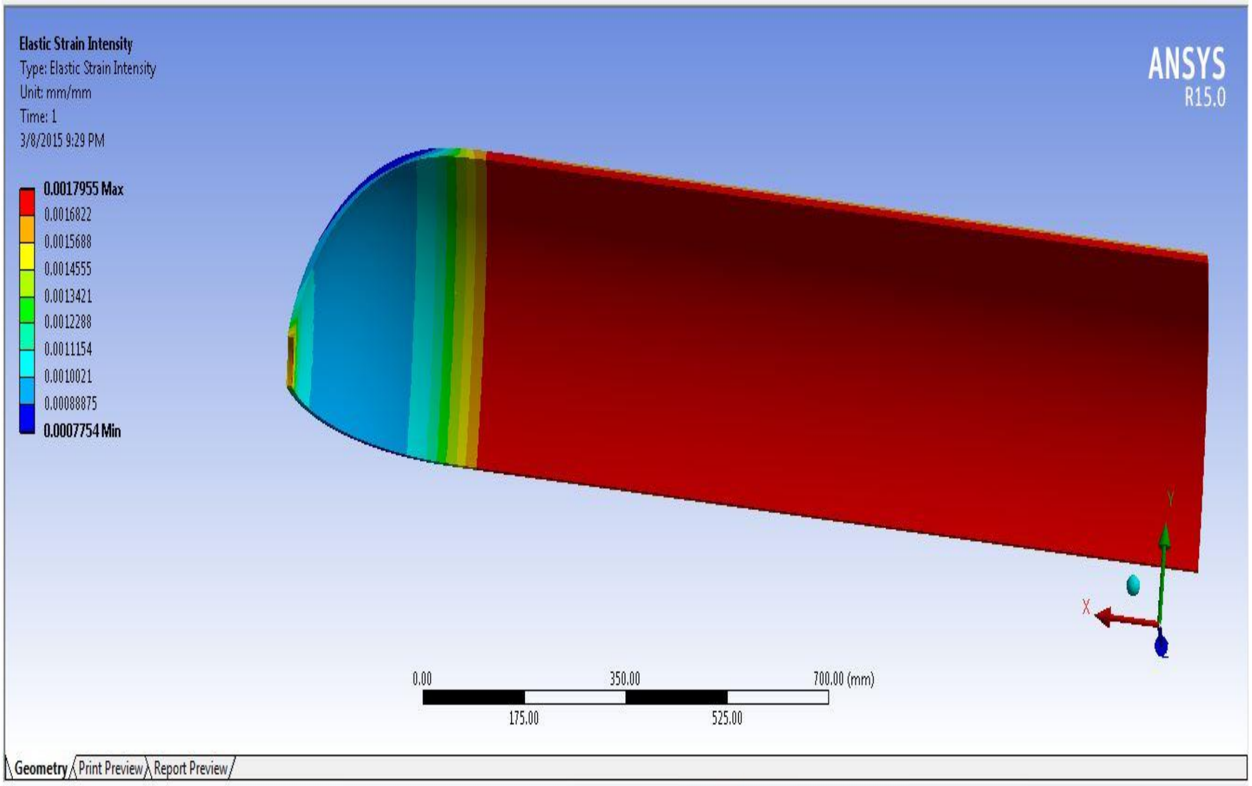


FIG 37: Strain plot for 9.3 mm

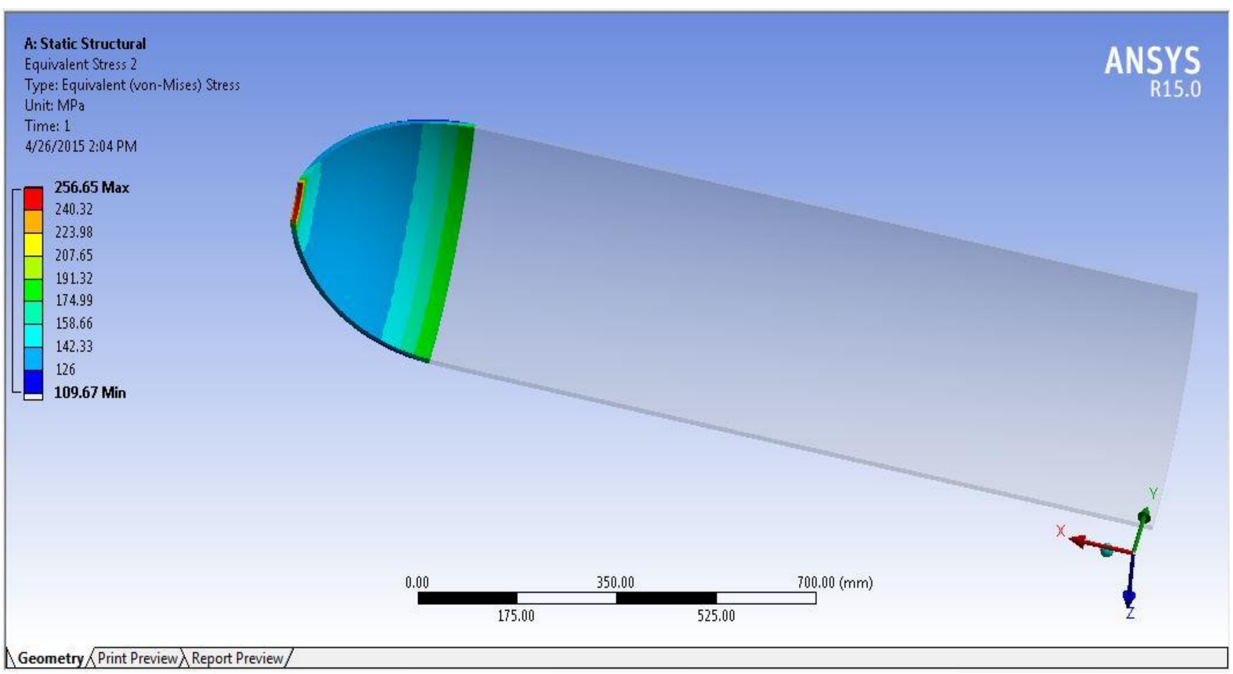


FIG 38: Stress plot at Head 9.3 mm

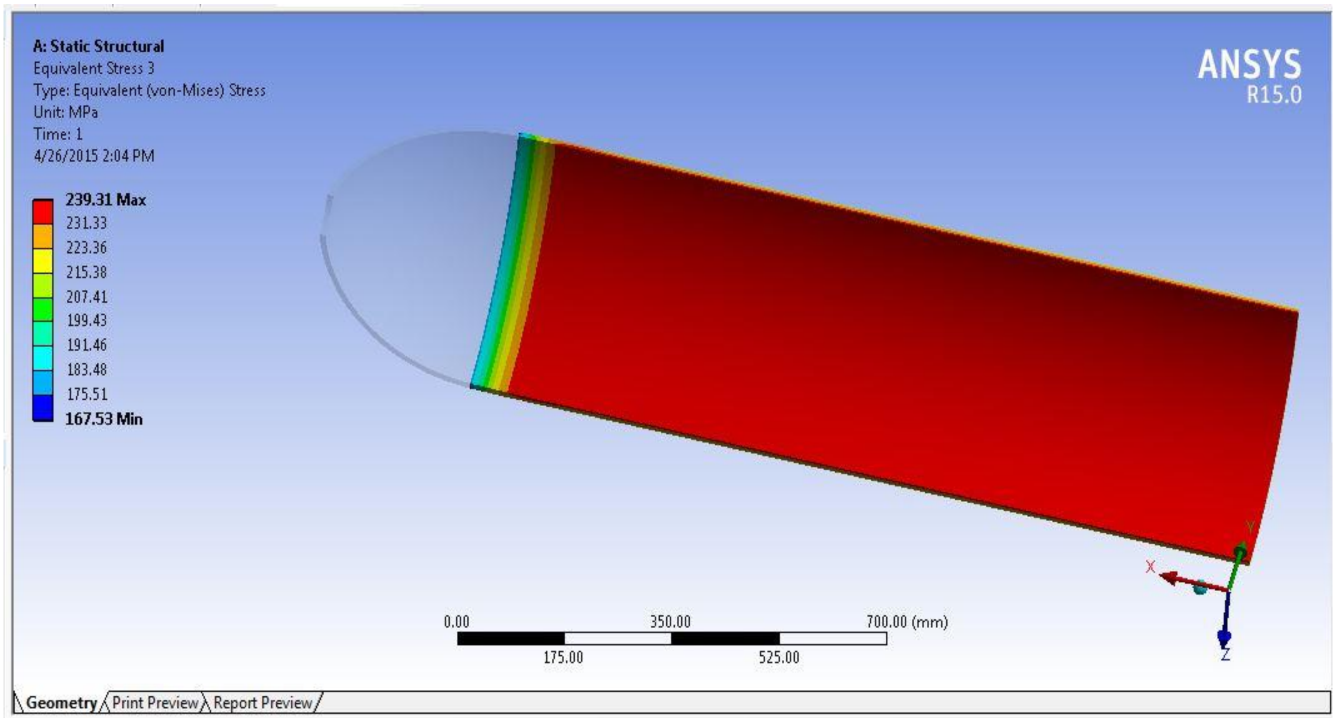


FIG 39: Stress plot at Shell 9.3 mm

6.6 Case 2 : - Thickness at assembly level 8.5 mm stress plot

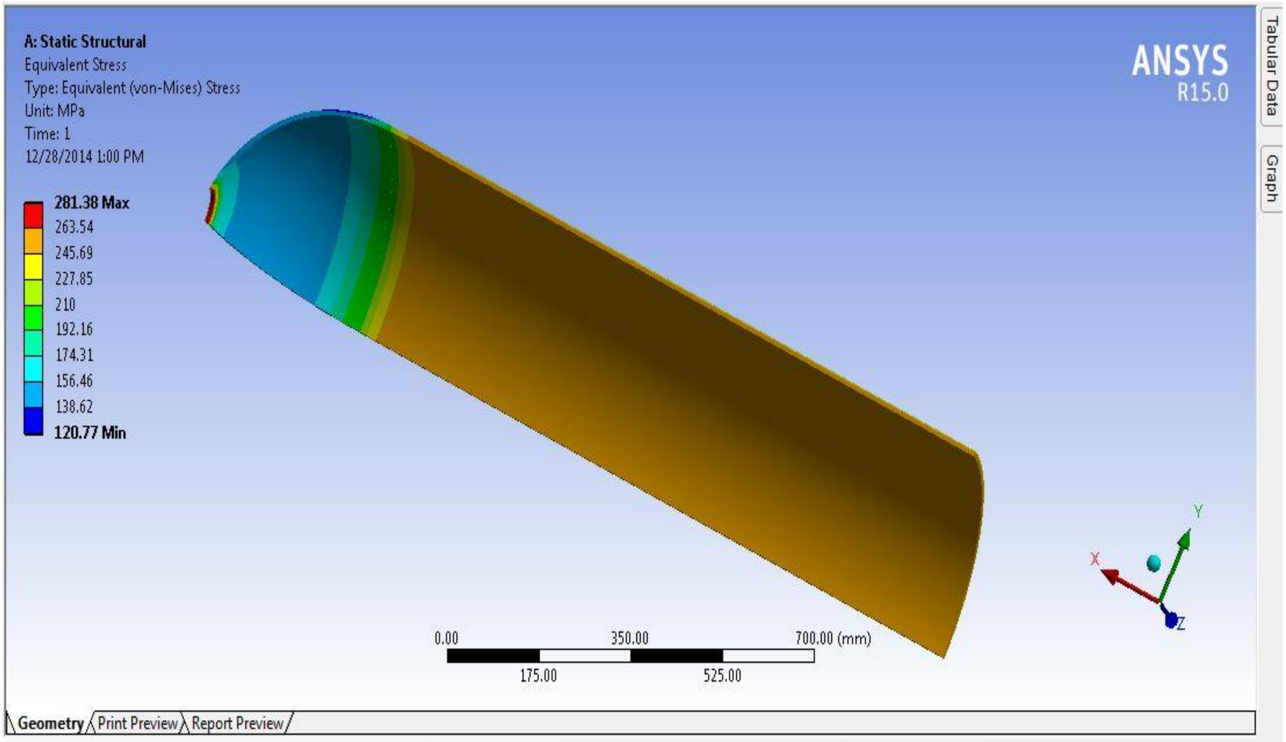


FIG 40: Stress plot at Assembly 8.5 mm

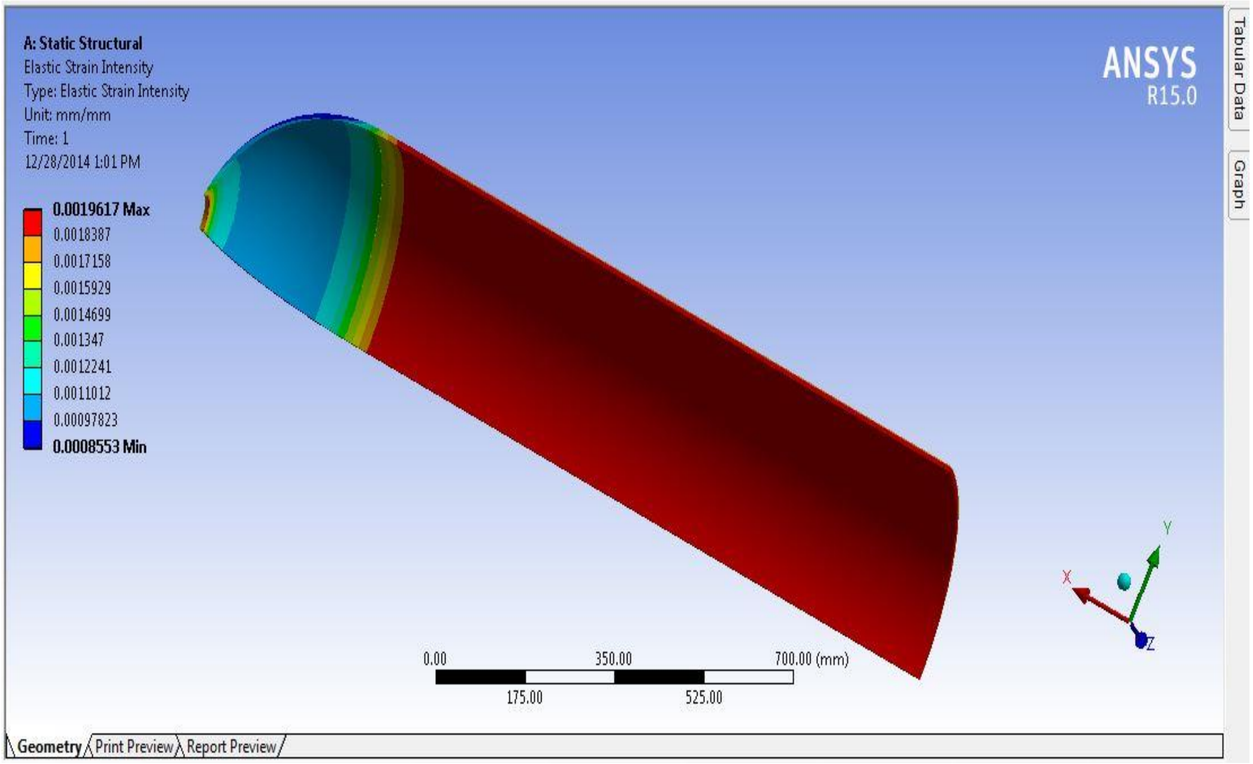


FIG 41: Strain plot at Assembly 8.5 mm

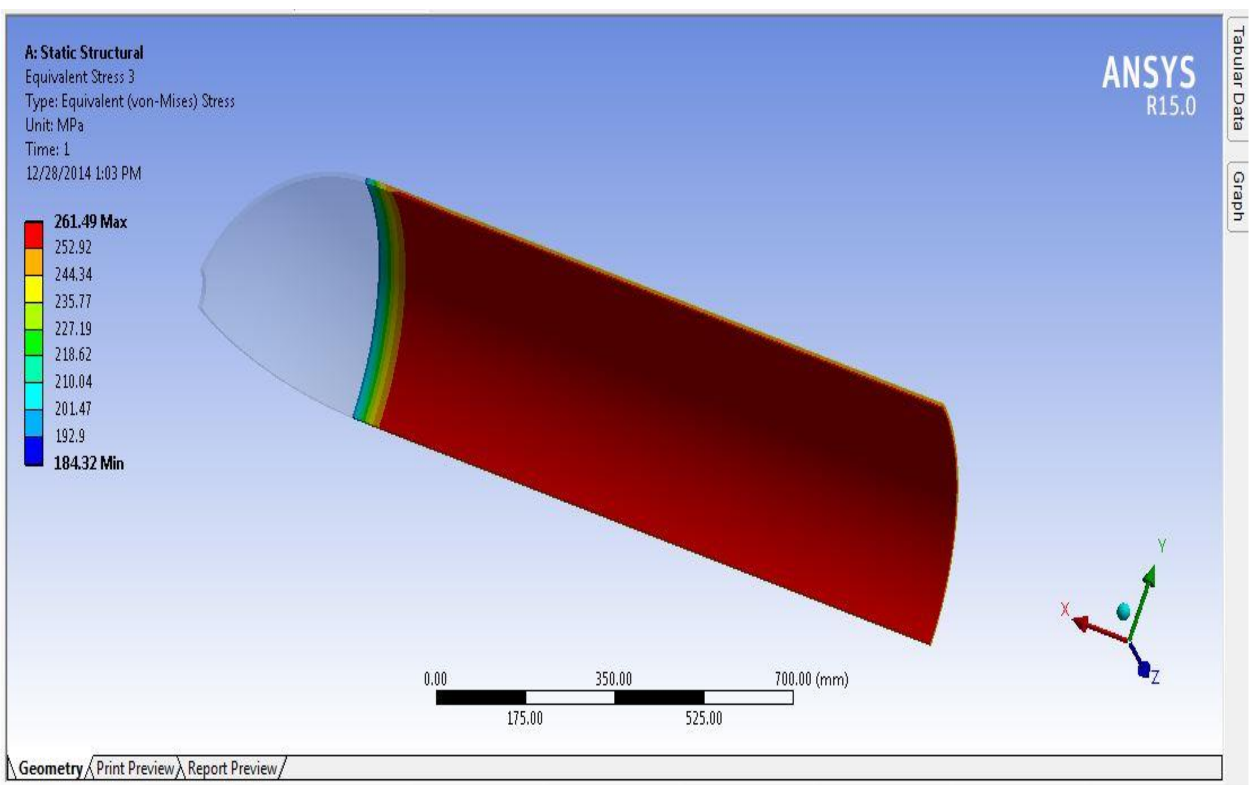


FIG 42: Stress plot at Shell 8.5 mm

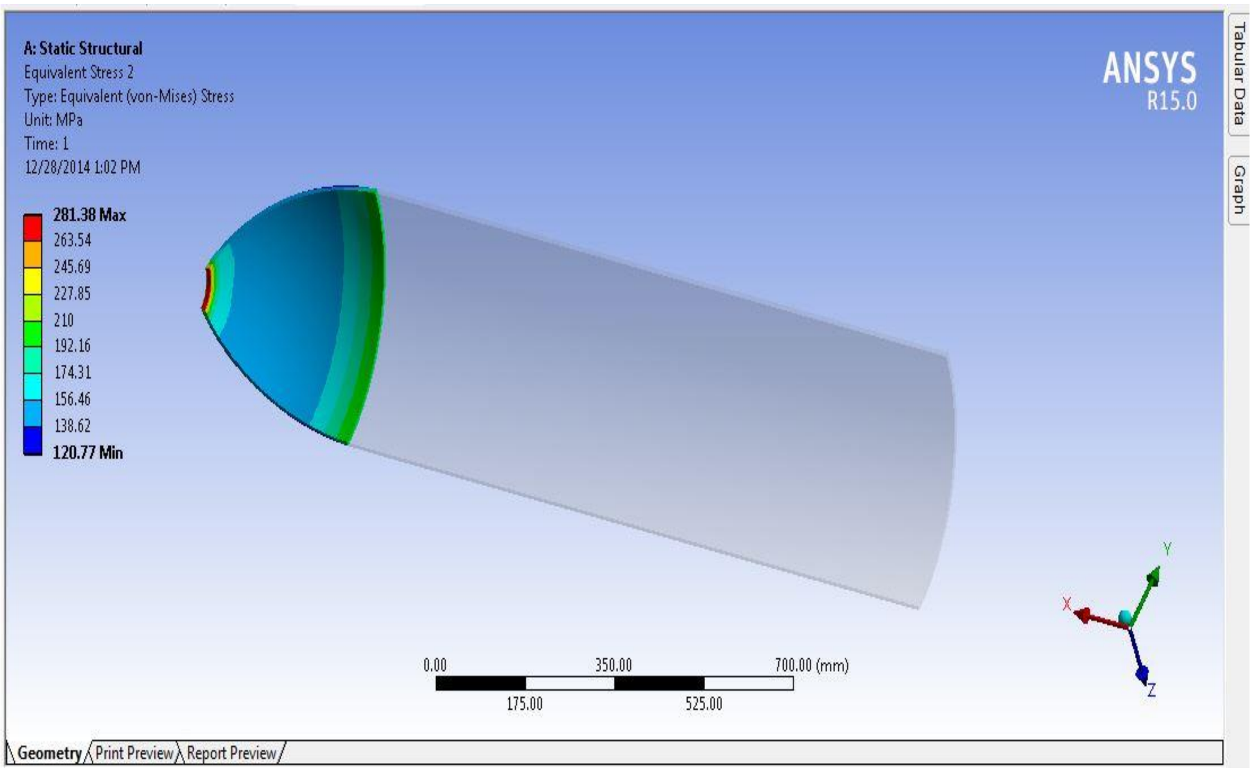


FIG 43: Stress plot at Head 8.5 mm

6.7 Case 3 : - Thickness at assembly level 8 mm stress plot

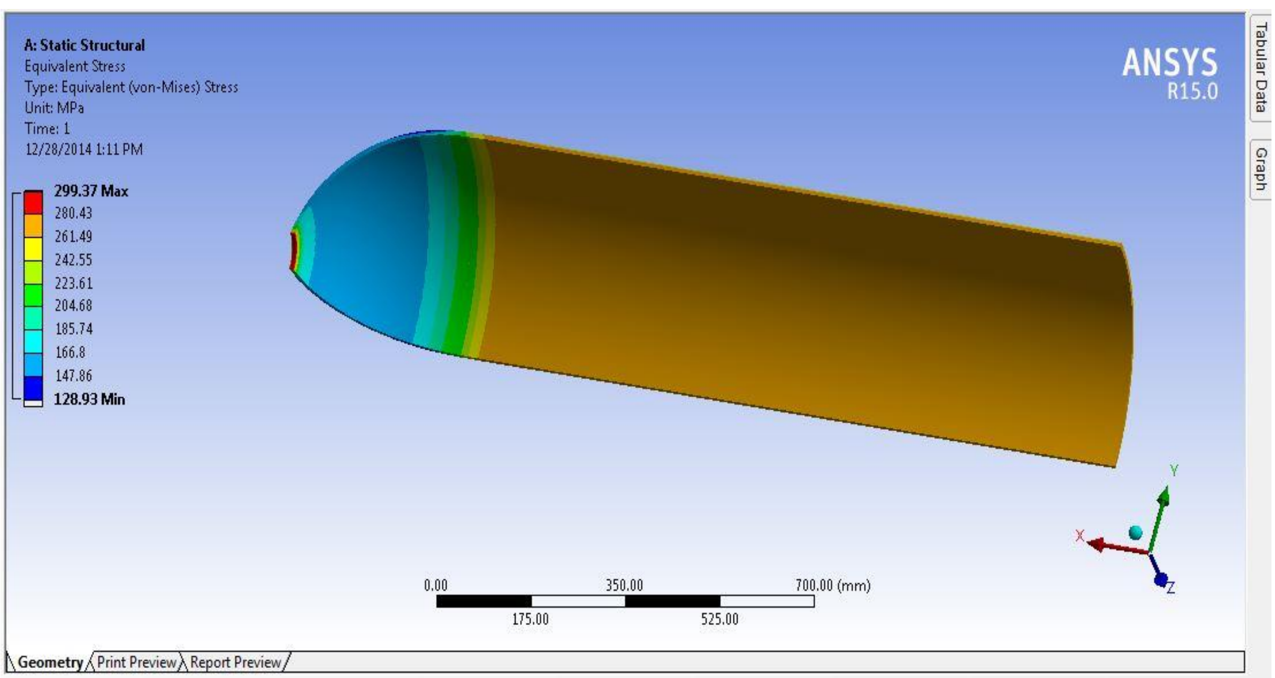


FIG 44: Stress plot at Assembly 8 mm

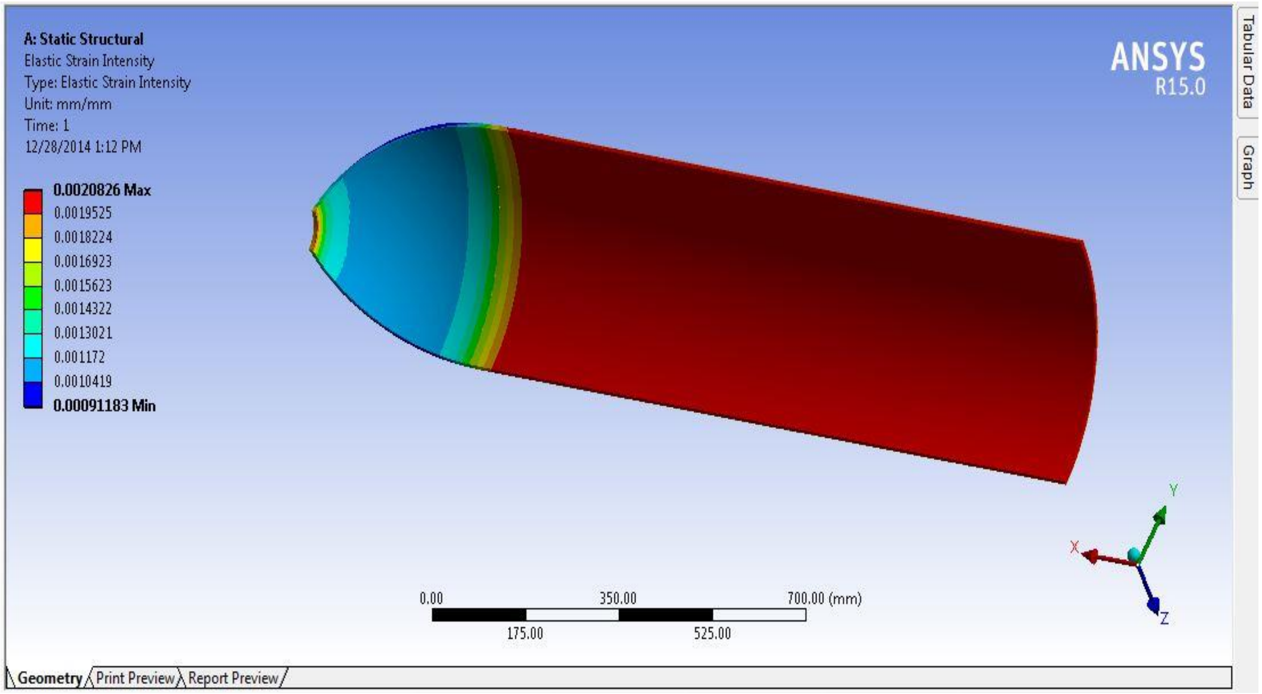


FIG 45: Strain plot at Assembly 8 mm

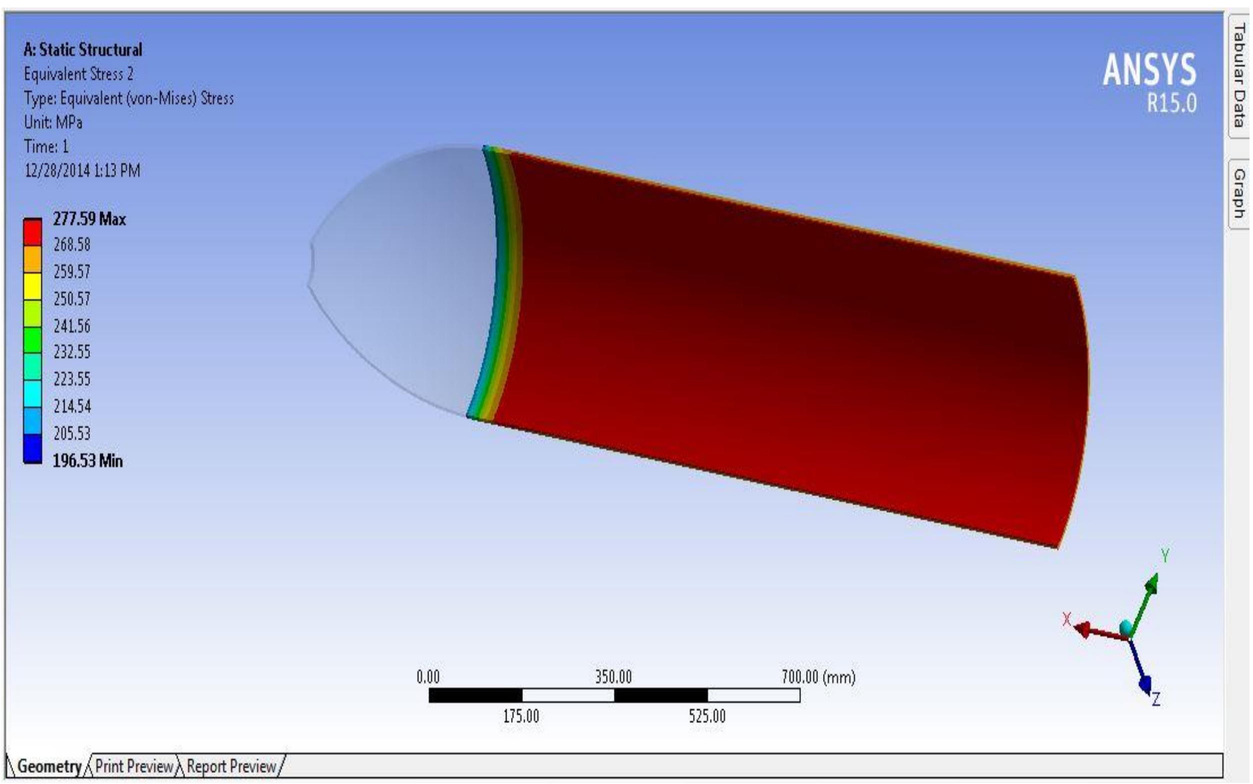


FIG 46: Stress plot at Shell 8 mm

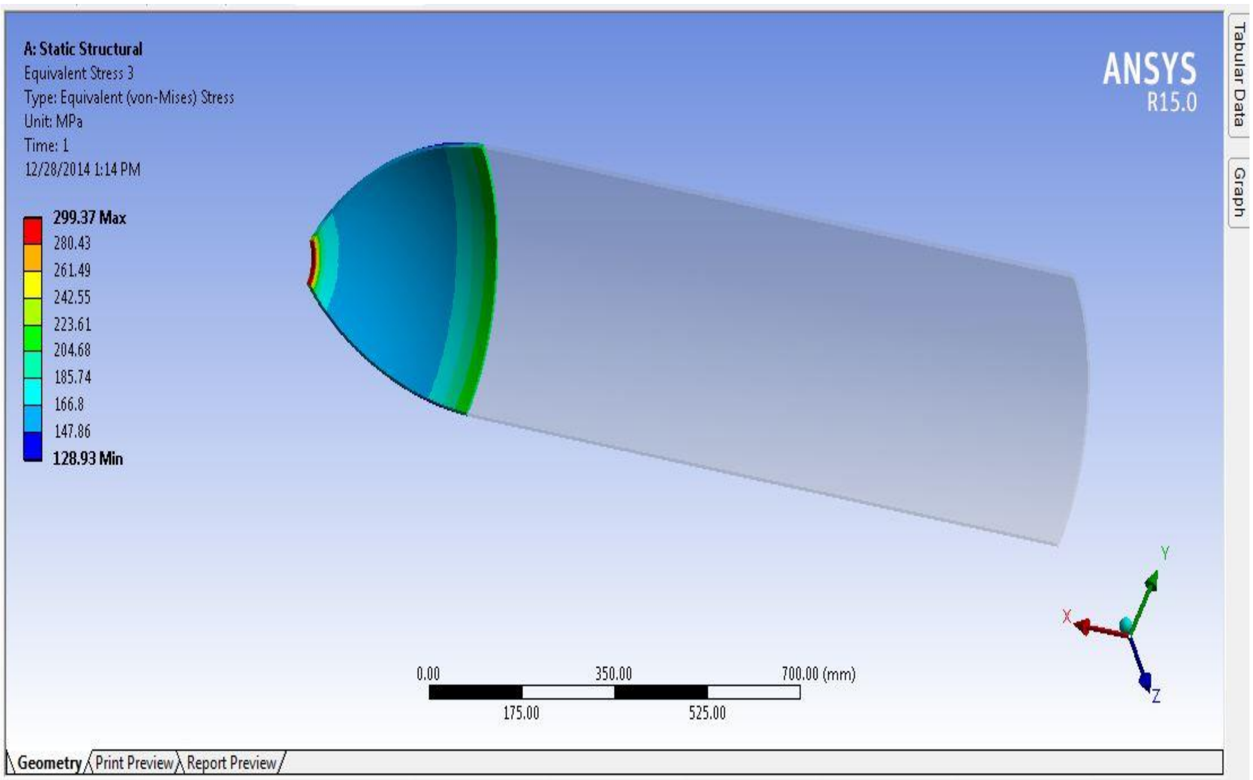


FIG 47: Stress plot at Head 8 mm

6.8 Case 4 : - Thickness at assembly level 6 mm stress plot

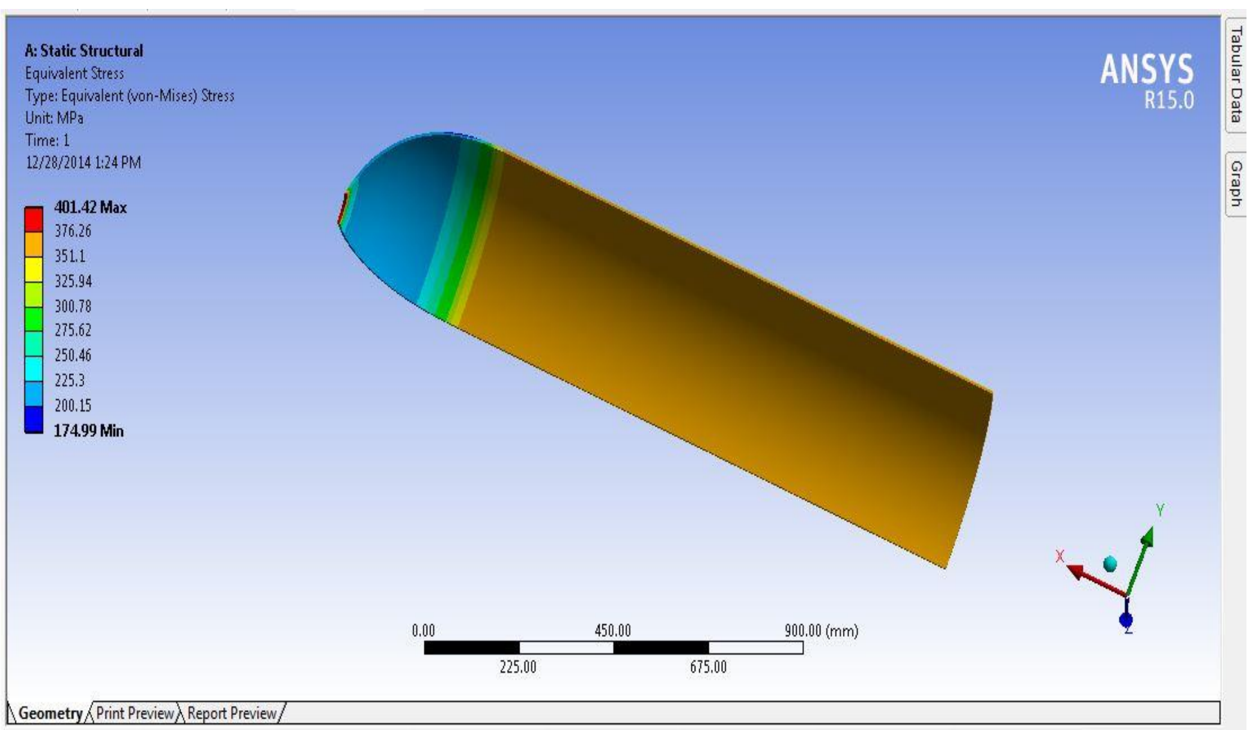


FIG 48: Stress plot at Assembly 6 mm

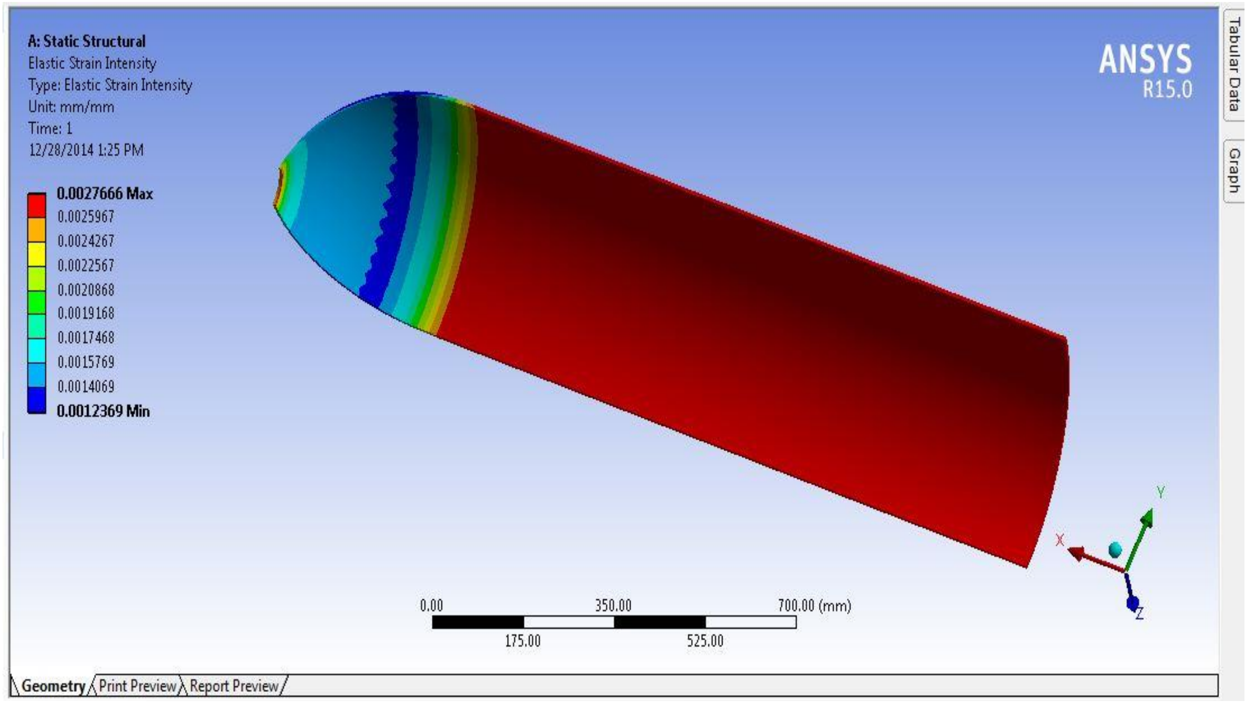


FIG 49: Strain plot at Assembly 6 mm

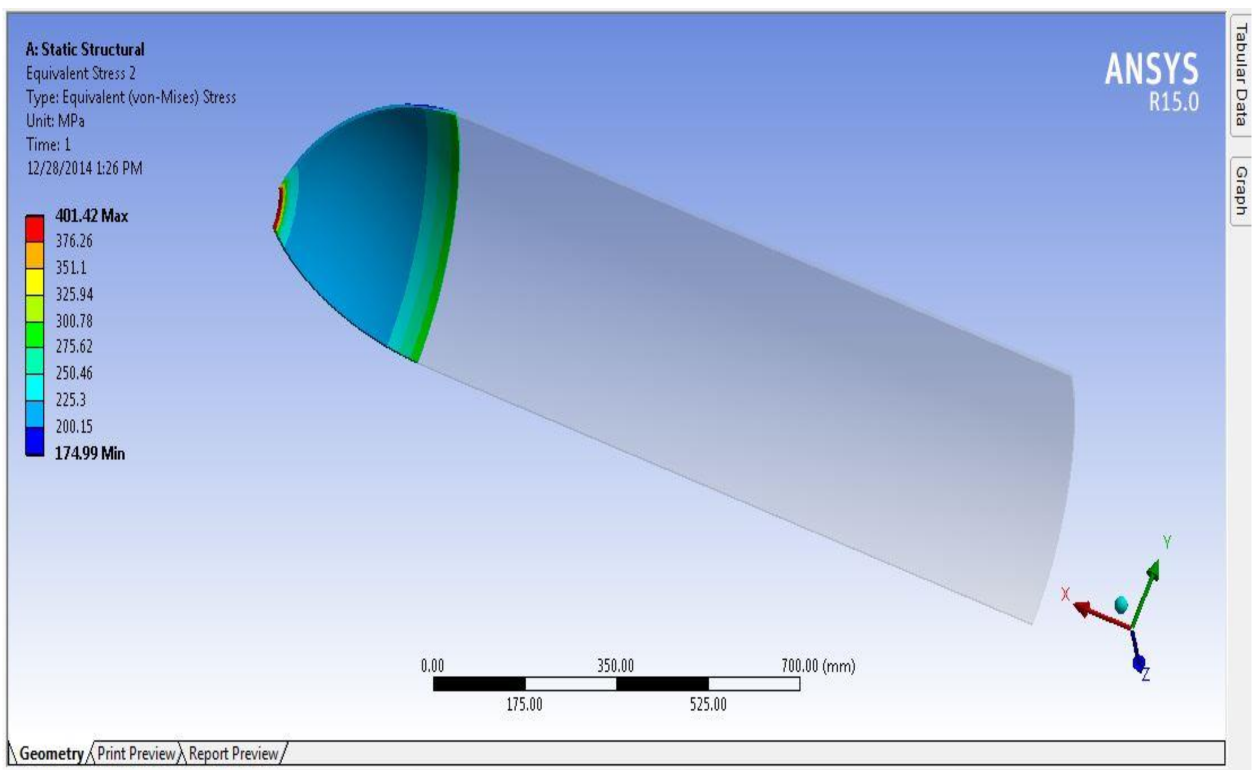


FIG 50: Stress plot at Head 6 mm

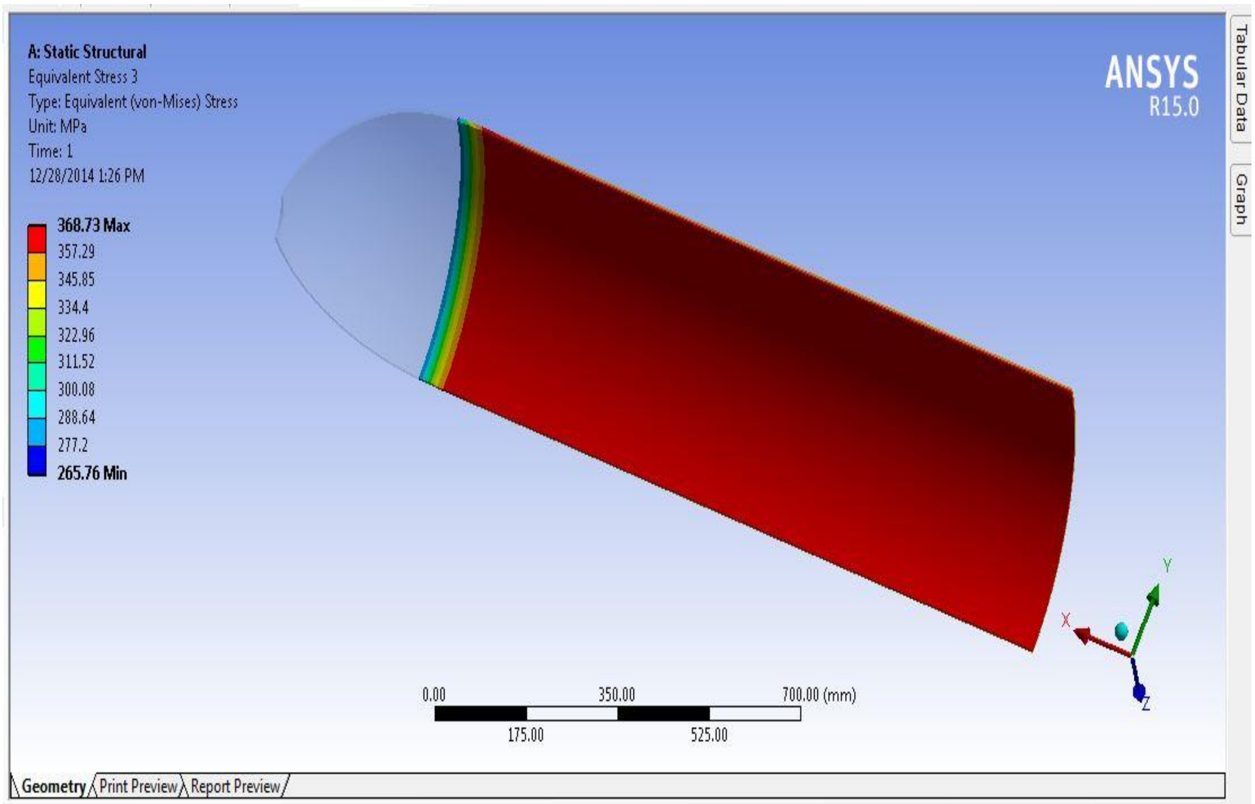


FIG 51: Stress plot at Shell 6 mm

6.9 Factor of Safety Calculations based on Stress Values

Factor of safety is defined as the ratio of the maximum stress that a structural part can withstand to the maximum stress estimated for it in the use for which it is designed.

Mathematically ,

F.O.S=Material Strength @ operating temperature/Max Stress Value

Yeild strength of material SA-723 @ 300 degrees taken is 450 MPa for analytical calculations.

Temp. [°C]	Modulus of Elasticity E [N/mm^2]	Yield strength $R_{p0.2}$ [N/mm^2]	Yield strength $R_{t0.5}$ [N/mm^2]
20	210000	520	526
100	210000	520	526
200	189000	485	496
300	168000	439	455
400	147000	381	399
500	126000	255	280
600	65100	118	132
700	27300	66	72
750	23100	46	51
800	18900	29	33
850	16537.5	20	23
900	14175	13	17
950	11812.5	12	14
1000	9450	10	11

Table 5 Yeild Strength of Steel at Elevated Temperatures

It is clear from above data that the yield strength of steel at 300 degree centigrade is 450 MPa approximately.

- ASME Code case : Thickness =9.3 mm

  F.O.S = 450 / 256.6 = 1.75

- CASE 2:Thickness = 8.5 mm

  F.O.S = 450 / 281.3 = 1.59

- CASE 3:Thickness = 8 mm

  F.O.S = 450 / 299.3 = 1.5

- CASE 4:Thickness = 6 mm

F.O.S = 410 / 401.4 = 1.1

Having calculated factor of safety values next we calculate the stress values analytically . Stress concentration factors have been taken into account here. For the four design cases following stresses will be calculated.

- Hoop Stress
- Longitudnal Stress
- Von-Misses Stress

6.10 Hoop Stress Plot for ASME Code Case

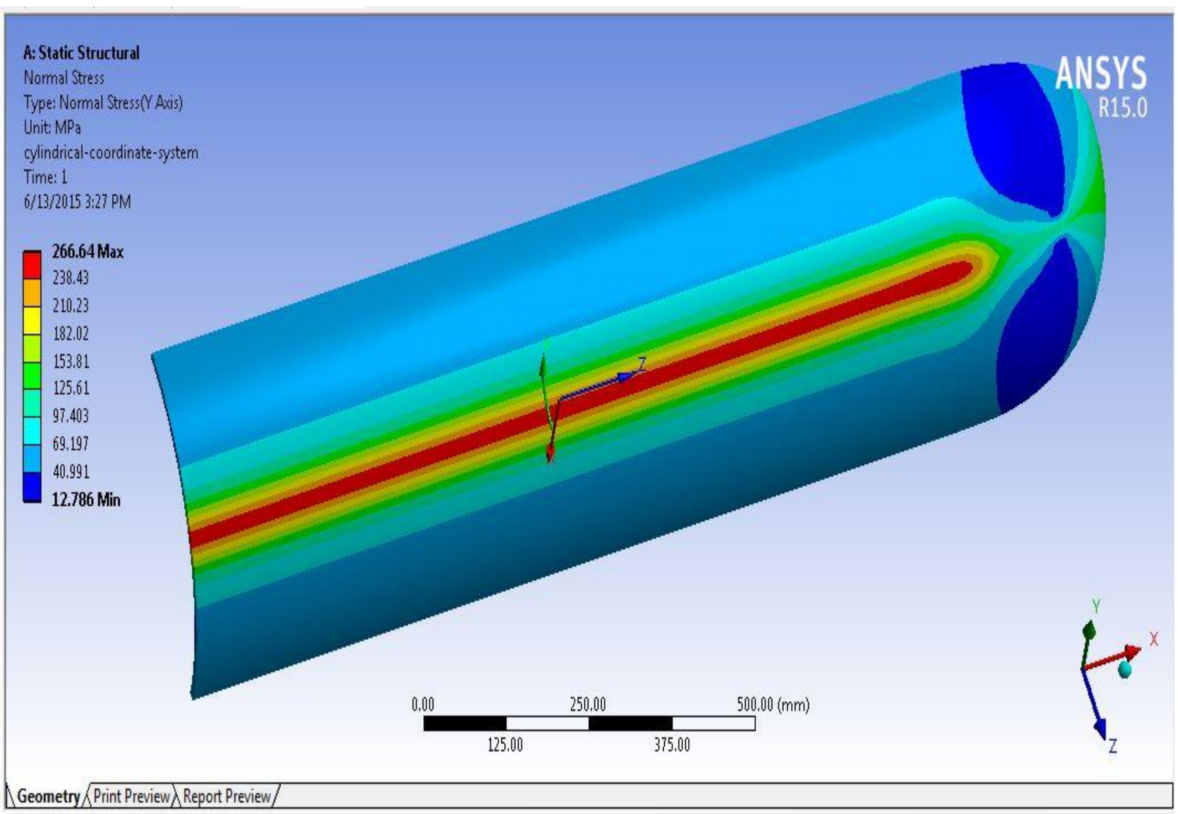
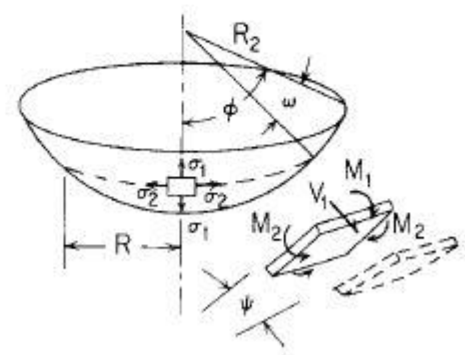


FIG 52: Hoop Stress Plot for ASME Code case

6.11 ANALYTICAL CALCULATIONS FOR 9.3 mm ASME CASE



$$\beta = [\, 3(1 - v^2)\left(\frac{R}{t}\right)^2 \,]^{\frac{1}{4}}$$

FIG 53: Constant Beta β Used in Stress concentration factor calculation

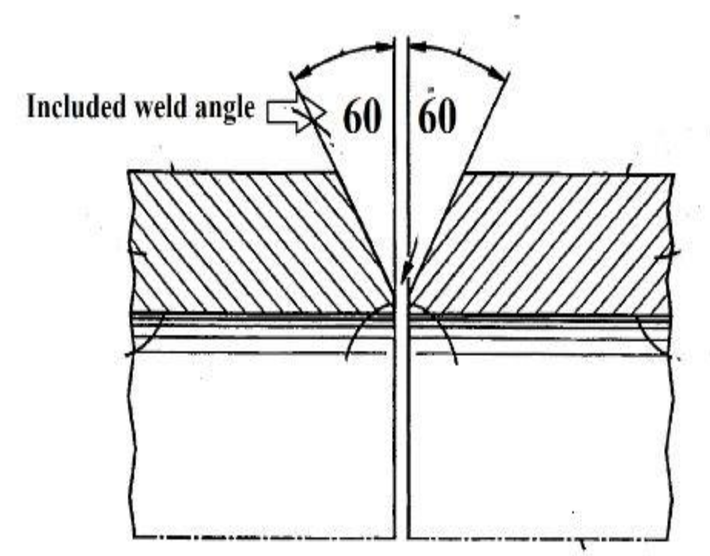


FIG 54: Weld included angle ϕ used in Stress concentration factor calculation

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

$$\sigma_2 = \sigma_H = \frac{9.6*10^6*520*10^{-3}}{2*9.3*10^{-3}} = 268.3 \text{ MPa}$$

Now including the stress concentration factor

$$K2 = 1 - \frac{1+2v}{2\beta} \cot \emptyset$$

After putting values

$$K2 = 1 - \frac{1+2*0.3}{2*6.79} \cot 120 = 1.068$$

Therefore

$$\sigma_{2\,max} = K2*\sigma_2$$

$$= 1.068*268.3 = 286.4 \text{ MPa}$$

Now We Proceed For Longitudnal Stress Value

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

$$\sigma_1 = \sigma_L = \frac{9.6*10^6*520*10^{-3}}{4*9.3*10^{-3}} = 134.1\ MPa$$

Now including the stress concentration factor

$$K1 = 1 - \frac{1-2v}{2\beta} \cot \emptyset$$

After putting values

$$K1 = 1 - \frac{1-2*0.3}{2*6.79} \cot 120 \;=1.017$$

Therefore

$$\sigma_{1\,\max} = K1*\sigma_1$$

$$= 1.017*134.1 = 136.3 \text{ MPa}$$

In these calculations β is constant

$$\beta = [\,3(1-v^2)\left(\frac{R}{t}\right)^2\,]^{\frac{1}{4}}$$

And φ is the bevel angle

$$\sigma_r = \sigma_1 = 0$$

Hence the Von-MISES Stress can now be calculated as follows:-

$$\sigma_v = \sqrt{\frac{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2}{2}}$$

After putting values

$$\sigma_v = \sqrt{\frac{(0-286)^2+(286-136.2)^2+(0-136.2)^2}{2}}$$

$$\sigma_v = 247.7\ MPa$$

6.12 ANALYTICAL CALCULATIONS FOR 8.5 mm CASE

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

$$\sigma_2 = \sigma_H = \frac{9.6*10^6*520*10^{-3}}{2*8.5*10^{-3}} = 293.6 \text{ MPa}$$

Now including the stress concentration factor

$$K2 = 1 - \frac{1+2v}{2\beta} \cot \emptyset$$

After putting values

$$K2 = 1 - \frac{1+2*0.3}{2*7.1} \cot 120 = 1.064$$

Therefore

$$\sigma_{2\max} = K2*\sigma_2$$

$$= 1.064*293.6 = 312.3 \text{ MPa}$$

Now We Proceed For Longitudnal Stress Value

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

$$\sigma_1 = \sigma_L = \frac{9.6*10^6*520*10^{-3}}{4*8.5*10^{-3}} = 146.82\ MPa$$

Now including the stress concentration factor

$$K1 = 1 - \frac{1-2v}{2\beta} \cot \emptyset$$

After putting values

$$K1 = 1 - \frac{1-2*0.3}{2*7.1} \cot 120 = 1.016$$

Therefore

$$\sigma_1 \text{ max} = K1 * \sigma_1$$

$$= 1.016 * 146.82 = 149.1 \text{ MPa}$$

In these calculations β is constant

$$\beta = [\, 3(1 - v^2)\left(\frac{R}{t}\right)^2 \,]^{\frac{1}{4}}$$

And φ is the bevel angle

$$\sigma_r = \sigma_1 = 0$$

Hence the Von-MISES Stress can now be calculated as follows:-

$$\sigma_v = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}}$$

After putting values

$$\sigma_v = \sqrt{\frac{(0 - 312)^2 + (312 - 149)^2 + (0 - 149)^2}{2}}$$

$$\sigma_v = 270.5\ MPa$$

6.13 ANALYTICAL CALCULATIONS FOR 8 mm ASME CASE

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

$$\sigma_2 = \sigma_H = \frac{9.6 * 10^6 * 520 * 10^{-3}}{2 * 8 * 10^{-3}} = 312 \text{ MPa}$$

Now including the stress concentration factor

$$K2 = 1 - \frac{1+2v}{2\beta} \cot \emptyset$$

After putting values

$$K2 = 1 - \frac{1+2*0.3}{2*7.32} \cot 120 \; = 1.063$$

Therefore

$$\sigma_2 \max = K2 * \sigma_2$$

$$= 1.063*312 = 331.6 \text{ MPa}$$

Now We Proceed For Longitudnal Stress Value

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

$$\sigma_1 = \sigma_L = \frac{9.6*10^6*520*10^{-3}}{4*8*10^{-3}} = 156\ MPa$$

Now including the stress concentration factor

$$K1 = 1 - \frac{1-2v}{2\beta} \cot \emptyset$$

After putting values

$$K1 = 1 - \frac{1-2*0.3}{2*7.32} \cot 120 \; =1.015$$

Therefore

$$\sigma_1 \max = K1 * \sigma_1$$

$$= 1.015*156 = 158.3 \text{ MPa}$$

In these calculations β is constant

$$\beta = [\,3(1-v^2)\left(\frac{R}{t}\right)^2\,]^{\frac{1}{4}}$$

And φ is the bevel angle

$$\sigma_r = \sigma_1 = 0$$

Hence the Von-MISES Stress can now be calculated as follows:-

$$\sigma_v = \sqrt{\frac{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2}{2}}$$

After putting values

$$\sigma_v = \sqrt{\frac{(0-331.6)^2+(331.6-158.3)^2+(0-158.3)^2}{2}}$$

$$\sigma_v = 287.2\ MPa$$

6.14 ANALYTICAL CALCULATIONS FOR 6 mm ASME CASE

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

$$\sigma_2 = \sigma_H = \frac{9.6*10^6*520*10^{-3}}{2*6*10^{-3}} = 416\ \text{MPa}$$

Now including the stress concentration factor

$$K2 = 1 - \frac{1+2v}{2\beta}\cot\emptyset$$

After putting values

$$K2 = 1 - \frac{1+2*0.3}{2*8.46} \cot 120 = 1.054$$

Therefore

$$\sigma_2 \max = K2*\sigma_2$$

= 1.054*416 = 438.4 MPa

Now We Proceed For Longitudnal Stress Value

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

$$\sigma_1 = \sigma_L = \frac{9.6*10^6*520*10^{-3}}{4*6*10^{-3}} = 208\ MPa$$

Now including the stress concentration factor

$$K1 = 1 - \frac{1-2v}{2\beta} \cot \emptyset$$

After putting values

$$K1 = 1 - \frac{1-2*0.3}{2*8.46} \cot 120 = 1.013$$

Therefore

$$\sigma_1 \max = K1*\sigma_1$$

= 1.013*208 = 210.7 MP

In these calculations β is constant

$$\beta = [\ 3(1 - v^2)\left(\frac{R}{t}\right)^2]^{\frac{1}{4}}$$

And φ is the bevel angle

$$\sigma_r = \ \sigma_1 = 0$$

Hence the Von-MISES Stress can now be calculated as follows:-

$$\sigma_v = \ \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}}$$

After putting values

$$\sigma_v = \ \sqrt{\frac{(0 - 443.4)^2 + (443.4 - 211.3)^2 + (0 - 211.3)^2}{2}}$$

$$\sigma_v = 379.7\ MPa$$

6.15 Comparison of Von-Misses Stress Values (Analysis Results Vs Analytical Calculations) & Error Quantification

SR.No	STRESS VALUES (PRESSURE+ TEMPERATURE APPLIED) MPa	STRESS CALCULATED ANALYTICAL (MPa)	ERROR (DIFFERENCE B/W FEA AND ANALYTICAL)
CASE 1: 9.3 mm	256.6	247.7	8.9
CASE 2: 8.5 mm	281.3	270.5	10.8
CASE 3: 8 mm	299.3	287.2	12.1
CASE 4: 6 mm	401.4	379.7	21

Table 6 - Comparison of Analysis Results with Calculations

The above table shows comparison of analysis results with analytical calculations. The criteria choosen for stress calculations is von misses stresses. There is minor difference of in stress values because of assumptions followed in formulas while derivation.

6.16 Validation of Results by Experimental Values.

- The experimental results obtained after hydro testing for the pressure vessel are as follows:-

✓ Test 1: 98.6 Bar (Failure) Hydrostatic Pressure

✓ Test 2: 97.2 Bar (Failure) Hydrostatic Pressure

✓ Test 3: 101.1 Bar (Failure) Hydrostatic Pressure

By experiments we found that the tests failed because the material crosses the elastic limit strain value of 0.2 %.

Now by using our same analysis technique we will check whether the designed pressure vessel crosses the limit strain value. If yes then our design by analysis approach is verified and validated against the experimental results.

Result 1 (97.2 bars)

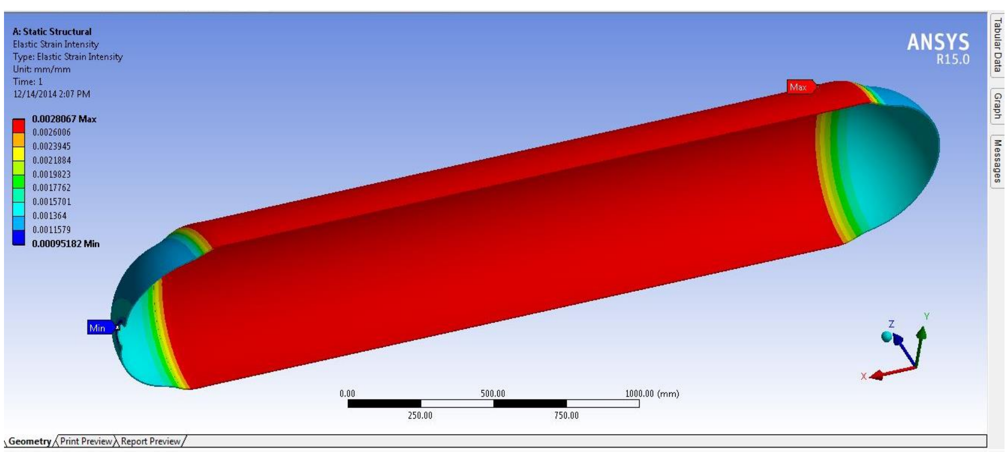


FIG 55: Analysis Results for strain value at 97.2 bars

Result 2 (98.6 bars)

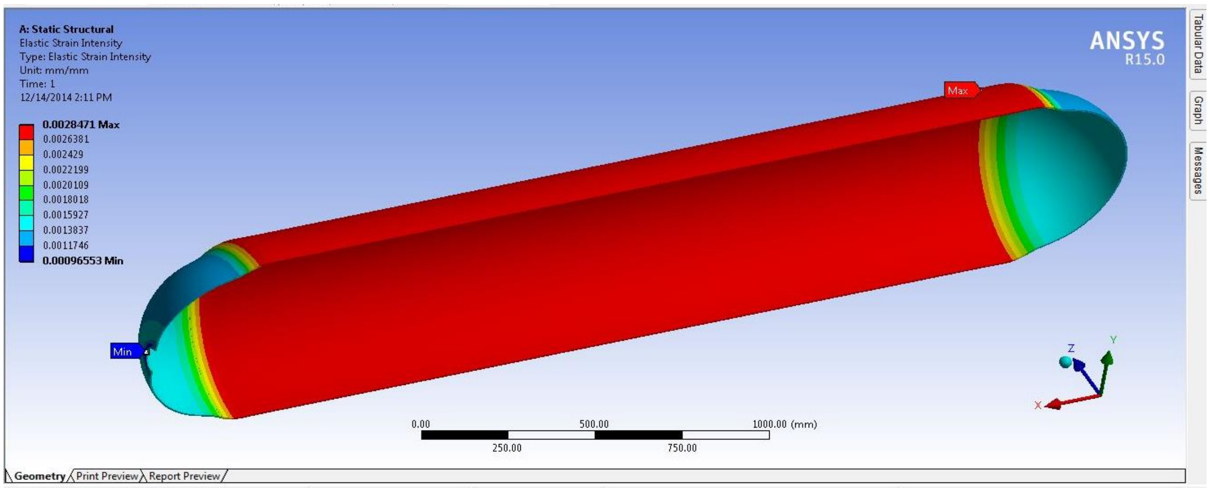


FIG 56: Analysis Results for strain value at 98.6 bars

Result 3 (101.1 bars)

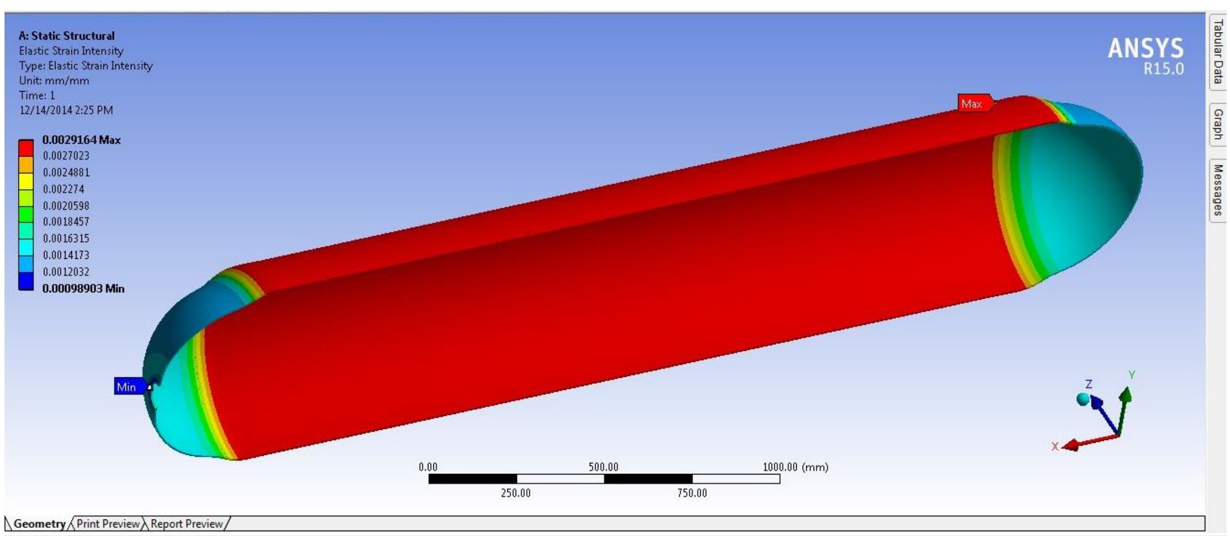


FIG 57: Analysis Results for strain value at 101.1 bars

- Using the design and analysis approach in present research we find that the linear strain values are as follows:-

TEST PRESSURE	LINEAR STRAIN VALUE
97.2 bar	0.281 %
98.6 bar	0.284 %
96 bar	0.27 %
101.1 bar	0.291 %

Table 7 - Linear Strain Values

The analysis results also depict that with 6 mm thickness, pressure vessel fails because it crosses the limit strain value of 0.2% at 96 bar also. So it matches with experimental criteria.

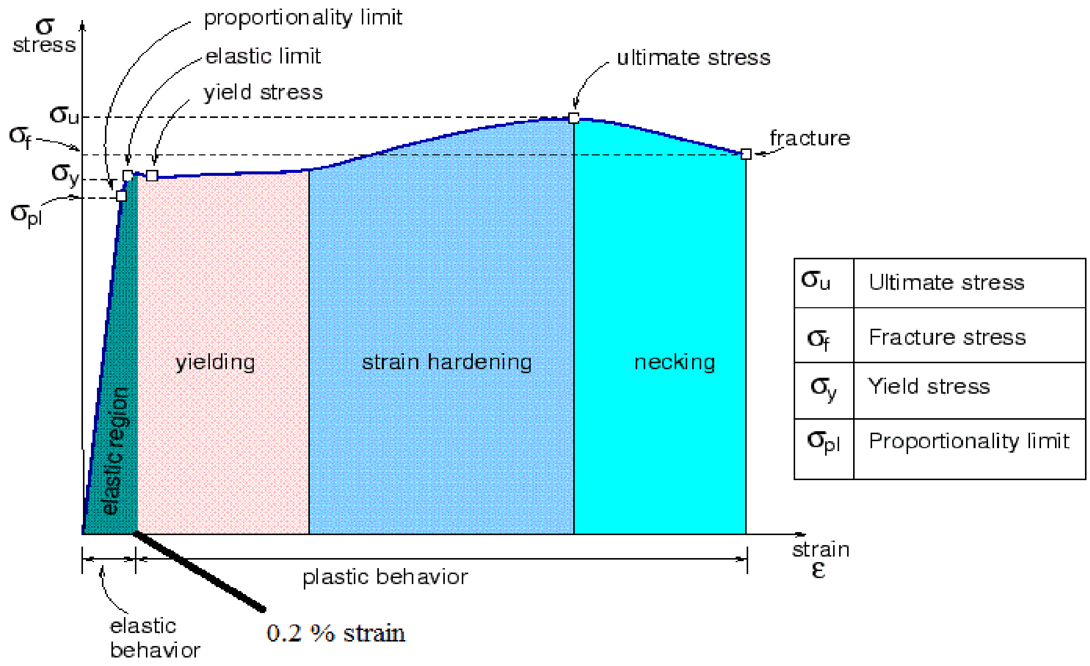


Fig 58 : Stress Strain behavior for steel at room temperature

6.17 Conclusions

- As we solve the base case of ASME codes the FOS is 1.75.Then I reduced this FOS by keeping thickness 8.5 mm F.O.S reduced to 1.59. In next case I further changed thickness to 8 mm and then F.O.S. Reduced to 1.5
- Below 8.5 mm as we further reduce thickness to 8 mm the F.O.S reduces to 1.5 but the pressure vessel is not safe for this particular application because the strain value crosses limit of 0.2 % . And same trend continues for 6mm thickness value.
- Therefore the best possible results we get with 8.5 mm case and we are able to get 9 % efficient design in terms of factor of safety as compared to that of our code results. Hence we get better design by keeping uniform factor of safety of 1.5 at the assembly level.

Description	Stress Values	Strain Values	Safe /Not Safe
9.3mm(F.O.S=1.75)	256.6 MPa	0.17 %	Safe
8.5 mm(F.O.S=1.59)	281.3 MPa	0.19 %	Safe
8 mm (F.O.S =1.5)	299.3 MPa	0.209 %	Not Safe
6 mm (F.O.S=1.1)	401.4 MPa	0.27 %	Not Safe

Table 8 - Safe and Unsafe based on Strain Values

Another important conclusion that can be drawn from this research is that ASME codes are based on plane stress assumption whereas the simulations are by 3-D analysis. So this fact explains the difference between ASME and simulations.

6.18 Future Scope

Cylindrical Pressure Vessels have vast applications especially in aerospace applications.
If the present technique for design approach is used it can have great impact in reducing weight of aerospace structures by decreasing the over design aspects.

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