DESIGN AND ANALYSIS OF PRESSURE VESSEL USING DIFFERENT MATERIALS

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ABSTRACT

The pressure vessel is one of a large number of plant components for which stress analyses must be performed. A pressure vessel is a container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. This Project deals with the Finite element analysis of Pressure vessels with different type of heads keeping the same cylindrical volume and thickness. The desired pressure vessel is designed as per ASME standard section VIII, division I for 8 bar pressure and 24 lit volume. Thus some end connections are tested under FEA for the cause of finding stress concentration zone in each type of pressure vessel head under the same volume and sane pressure. The aim of the project is different designs and static and thermal analysis using ansys software of describes, flat head and elliptical head pressure vessel has low stresses distributed as compare to other heads, so for most applications elliptical heads selected. It shows basic structure and the finite element modeling for analyzing the pressure vessels with different type of heads and different materials like Nimonic 80A,SA516 Gr70 also under high stress zones. In this project we are working on approximate stresses that exist in cylindrical pressure vessels supported on two saddles support are calculated under the different type of end connections by using Finite Element tool. Static structural analysis and thermal analysis is done in order to calculate stresses in vessel finally concluded the suitable design and material.

KEYWORDS: Pressure Vessel, End connections, Stress analysis, steady state thermal .

1. INTRODUCTION

1.1 INTRODUCTION TO PRESSURE VESSELS

A pressure vessel is defined as container with internal pressure, higher than atmospheric pressure. The fluid inside the pressure vessel may undergo state of change like in case of boilers. Pressure vessel has combination of high pressure together with high temperature and may be with flammable radioactive material. Because of these hazards it is important to design the pressure vessel such that no leakage can take place as well as the pressure vessel is to be designed carefully to cope with high pressure and temperature. Plant safety and integrity are one of the fundamental concerns in pressure vessel design and these depend on adequacy of design codes. In general the cylindrical shell is made of a uniform thickness which is determined by the maximum circumferential stress due to the internal pressure. Since the longitudinal stress is only one-half of this circumferential stress. The structure is to be designed fabricated and checked as per American Society of Mechanical Engineers standards .Pressure vessels are used in number of industries like power generation industry for fossil and nuclear power generation, In petrochemical industry for storage of petroleum oil in tank as well as for storage of gasoline in service stations and in the chemical industry.

The size and geometric form of pressure vessel is varying from large cylindrical vessel for high pressure application to small size used as hydraulic unit of aircraft. In pressure vessel whenever expansion or contraction occurs normally as result of heating or cooling, thermal stresses are developed. There are many types of stresses developed in the vessel. Stresses are categorized into primary stresses and secondary stresses. Primary stresses are generally due to internal or external pressure or produced by moments and these are not self limiting. Thermal stresses are secondary stresses because they are self limiting. That is yielding or deformation of the part relaxes the stress (except thermal stress ratcheting).Thermal stresses will not cause failure by rupture in ductile materials except by fatigue over repeated loading applications.

FIGURE 1 TYPES OF BOILERS

1.2 Different types of heads are discussed in brief below:

1.2.1 FLAT HEADS: Flat heads or plates are the simplest type of end closures used only for small vessels. They can be used as manhole covers in low pressure vessels and as covers for small openings.

1.2.2 HEMISPHERICAL HEADS: A hemispherical head is the strongest shape and is capable of resisting nearly twice the pressure of a torispherical head of the same thickness. The cost of forming a hemispherical head will be higher than that for a shallow torispherical head. The amount of forming required to produce hemispherical shape is more, resulting in increased forming cost. As they are the expensive to form they are reserved for high pressure applications.

1.2.3 ELLIPSOIDAL HEADS: Ellipsoidal heads are often used for pressures over 10 bar. In cross-section, the head is like an ellipse with its radius varying continuously. This results in a smooth transition between the dome and the cylindrical part of the vessel. The shape of the ellipsoidal head is defined by the ratio of the major and minor axis. A standard arrangement on vessels is the 2:1 elliptical head. Due to shallow dished shape the forming cost is reduced.

1.2.4 TORISPHERICAL HEADS: A torispherical shape, which is extensively used as the end closure for a large variety of cylindrical pressure vessels. The shape is close to that of an ellipse but is easier and cheaper to fabricate. Torispherical heads are made of a dish, with a constant radius. Joining the dish directly to the cylindrical section of the vessel leads to a rapid change in geometry, resulting in excessive local stresses. To avoid this, a transition section (knuckle) is used between the dish and the cylinder. They are generally used for very high pressure applications.

1.2.5 CONICAL HEADS: The conical heads are widely used as bottom heads to facilitate the removal or draining of fluid. The semi-cone angle is usually taken as 30°

1.3 SHAPE OF PRESSURE VESSELS:

Pressure vessels are leak-proof containers. Pressure vessels can theoretically be almost any shape and range from beverage bottle to the sophisticated ones encountered in engineering Construction, but shapes made of sections of spheres, cylinders, and cones are usually employed. A common design is a cylinder with end caps called heads. Head shapes are frequently either hemispherical or dished (tori spherical). Theoretically, a spherical pressure vessel has approximately twice the strength of a cylindrical pressure vessel with the same wall thickness However, a spherical shape is difficult to manufacture, and therefore more expensive, so most pressure vessels are cylindrical with 2:1 semi-elliptical heads or end caps on each end. Smaller pressure vessels are assembled from a pipe and two covers.3

1.4 TYPES OF PRESSURE VESSELS

(a)According to dimension:

i. Thin Pressure vessel d/t > 20
ii. Thick Pressure vessel d/t< 20 1.4.2

(b)According To support:

i. Skirt supported
ii. Saddle supported
iii. Leg supported

(c)According to shape:

i. Cylindrical ii. Spherical

(d) According to construction;

i. Vertical ii. Horizontal

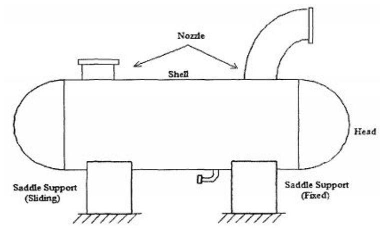


FIGURE 2 BOILER WITH SADDLE

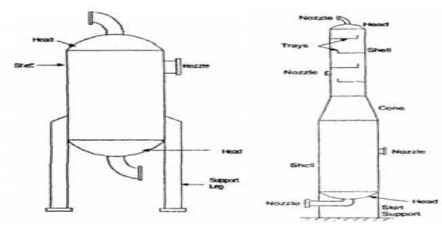


FIGURE 3 DIFFERENT TYPES OF PRESSURE VESSELS

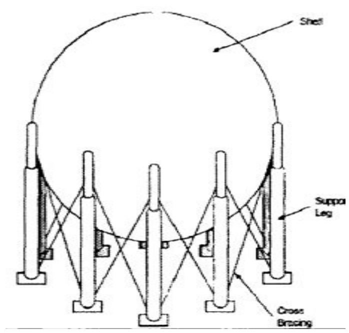


FIGURE 4 SPHERICAL PRESSURE VESSEL

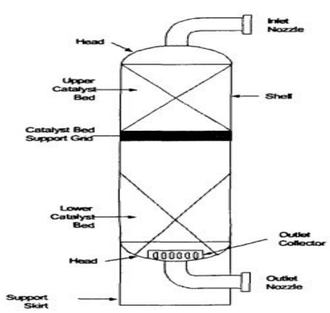


FIGURE 5 VERTICAL RECTATOR

1.5 COMPONENT OF PRESSURE VESSEL:

1. Shell
2. Head
3. Nozzle
4. Support
5. Lifting attachments

1.6 APPLICATIONS OF PRESSURE VESSELS:

There are numerous applications that require the use of containers for storage or transmission of gasses and fluids under high pressure. Pressure vessels have been used for a long time in various applications in both industry and the private sector. Pressure vessels are probably one of the most widespread equipment within the different industrial sectors. In fact, there is no industrial plant without pressure vessels, steam boilers, tanks, autoclaves, collectors, heat exchangers, pipes, etc. More specifically, pressure vessels represent fundamental components in sectors of paramount industrial importance, such as the nuclear, oil, petrochemical, and chemical sectors and also in the sectors as industrial compressed air receivers and domestic hot water storage tanks Other examples of pressure vessels are diving cylinders, recompression chambers, distillation towers, pressure reactors, autoclaves, and many other vessels in mining operations, oil refineries and petrochemical plants, nuclear reactor vessels, submarine and space ship habitats, pneumatic reservoirs, hydraulic reservoirs under pressure, rail vehicle airbrake reservoirs, road vehicle airbrake reservoirs, and storage vessels for liquefied gases such as ammonia, chlorine, propane, butane and LPG.

2. LITERATURE REVIEW

1. V. V. Wadkar, S.S. Malgave, D.D. Patil, H.S. Bhore, P. Gavade Assistant Professor, Mechanical Department, Aitrc, Vita, India. This study is about some of the current developments in the determination of stress concentration factor in pressure vessels. The literature has indicated a growing interest in the field of stress concentration analysis in the pressure vessels. Pressure vessels find wide applications in thermal and nuclear power plants, process and chemical industries, in space, ocean depths and fluid supply systems in industries. The main objective of this study is to design and analyse the features of pressure vessels. Various parameters of Solid Pressure Vessel are designed and checked according to the principles specified in American Society of Mechanical Engineers (A.S.M.E) Sec VIII Division 1. The stresses developed in Solid wall pressure vessel and Head of

pressure vessel is analysed by using ANSYS, a versatile Finite Element Package. The theoretical values and ANSYS values are compared for both solid wall and Head of pressure vessels.

2. Aziz onder, onursayman, tolgadogan, necmettintarakciogluSelcuk University, department of mechanical engineering, Konya, turkey. In this study, optimal angle-ply orientations of symmetric and antisymmetric [h/h] s shells designed for maximum burst pressure were examined. Burst pressure of filament wound composite pressure vessels under alternating pure internal pressure was investigated. The study deals with the influences of temperature and winding angle on filament wound composite pressure vessels. Finite element method and experimental approaches were employed to verify the optimum winding angles. An elastic solution procedure based on Lekhnitskii's theory was developed in order to predict the burst failure pressure of the pressure vessels.

3. A.th. Diamantoudis, th.Kermanidis laboratory of technology and strength of materials, department of mechanical engineering and aeronautics, university of Patras. A comparative study for design by analysis and design by formula of a cylinder to nozzle intersection has been made using different finite element techniques. The cylinder to nozzle intersection investigated is part of a typical vertical pressure vessel with a skirt support. For the study the commonly used ductile P355 steel alloy and the high strength steel alloy P500 QT were considered. The comparative results clearly show disadvantages in terms of limit load capability when the design-by-formula procedures are used in the design of high strength steel pressure vessels. The FE results also clearly show advantages of the shell to solid sub-modelling technique, as it combines the accuracy of 3D-solid modelling with the affordable computing time of the 3D-shell modelling technique.

4. Aniruddha A. Sathe, Vikas R. Maurya, Shriyash V. Tamhane, Akshaya P. Save, Parag V. NikamBachelor of Engineering Students, and Assistant Professor Department of Mechanical Engineering, St. John College of Engineering and Management, Palghar(E), Palghar, India The aim of this project isto perform the detailed design & analysis of pressure vessel for optimum thickness using SOLIDWORKS software. The selected components of pressure vessel like Shell, Heads, Nozzles, Supports and Lifting Lugs etc. are compared with Standard available thickness and optimization being done for the allowable stresses for MOC. The thickness of the pressure vessel is checked for different load cases. This results in the optimization of pressure vessel component thickness and hence reduces the overall weight and the cost the pressure vessel due optimum wall thickness for same service conditions. The optimized pressure vessel will be able to withstand all conditions applied on the pressure vessel during the service period of time with same safety factor but lower weight compared to the existing model.

5. Davidson, Thomas E. Kendall, David P. WATERVLIET ARSENAL NY BENET WEAPONS LAB The report is a review of the theory and practice of pressure vessel design for vessels operating in the range of internal pressures from 1 to 55 kilobars approximately 15,000 to 800,000 psi and utilizing fluid pressure media. The fundamentals of thick-walled cylinder theory are reviewed, including elastic and elastic-plastic theory, multi-layer cylinders and autofrettage. The various methods of using segmented cylinders in pressure vessel design are reviewed in detail. The factors to be considered in the selection of suitable materials for pressure vessel fabrication are discussed.

6. Mackenzie, A. Dalrymple, E. W. Schwartz, F. PICATINNY ARSENAL DOVER NJ FELTMAN RESEARCH LABS. The report contains special sections on the design of end closures, shock attenuation, providing for electrical lead-throughs needed for instrumentation, and the use of a thin window in the vessel needed for irradiation experiments. From this information a pressure vessel for a particular application can be designed.

7. W. S. PELLINI, P. P. PUZAK Metallurgy Division, U. S. Naval Research Laboratory, Washington, D.C. Practical Considerations in Applying Laboratory Fracture Test Criteria to the Fracture-Safe Design of Pressure Vessels. This report presents a "broad look" analysis of the opportunities to apply new scientific approaches to fracture safe design in pressure vessels and of the new problems that have arisen in connection with the utilization of higher-strength steels. These opportunities follow from the development of the fracture analysis

diagram which depicts the relationships of flaw size and stress level for fracture in the transition range of steels which live well-defined transition temperature features.

8. T.R. Tauchert department of engineering mechanics university of Kentucky Lexington. The distribution of fibres in a cylindrically reinforced pressure vessel of given size and constituent properties is optimized using the criterion of minimum strain energy. A stress function approach, in conjunction with the modified Rayleigh-Ritz technique, is employed to obtain an approximate solution to the non-linear optimization problem. Constraint conditions include specification of the global volume fraction of fibres and satisfaction of stress boundary conditions. Numerical results are presented for reinforced cylinders having various radii, modulus ratios, and global volume fractions. Included is the case of a reinforced concrete cylinder, in which the concrete is assumed to be ineffective in tension. In most cases examined, use of the optimum fibre distribution, rather than a uniform distribution, results in a substantial reduction in the maximum radial displacement and an increase in the failure pressure load.

9. LevendParnas, NuranKatirci Department of Mechanical Engineering, Middle East Technical University, 06531 Ankara, Turkey. An analytical procedure is developed to design and predict the behaviour of fibre reinforced composite pressure vessels. The classical lamination theory and generalized plane strain model is used in the formulation of the elasticity problem. Internal pressure, axial force and body force due to rotation in addition to temperature and moisture variation throughout the body are considered. Some 3D failure theories are applied to obtain the optimum values for the winding angle, burst pressure, maximum axial force and the maximum angular speed of the pressure vessel. These parameters are also investigated considering hygrothermal effects.

10. Piotr Dzierwa Faculty of Mechanical Engineering, Cracow University of Technology.Optimum Heating of Pressure Vessels with Holes. A method for determining time-optimum medium temperature changes is presented. The heating of the pressure elements will be conducted so that the circumferential stress caused by pressure and fluid temperature variations at the edge of the opening at the point of stress concentration does not exceed the allowable value. In contrast to present standards, two points at the edge of the opening are taken into consideration. Optimum fluid temperature changes are assumed in the form of simple time functions. It is possible to increase the fluid temperature stepwise at the beginning of the heating process and then the fluid temperature can be increased with a constant rate.

3. PROJECT OVER VIEW

3.1 OBJECTIVE OF THE PROJECT:

In the below point the background of the project is stated

1) Brief study of pressure vessel types and working is discussed in this project.

2) Stress evaluation for pressure vessel by optimizing different ends hemi spherical, toriconical head and flat conditions.

3) Modeling of pressure vessel is done in Catia v5 design software with wall thickness of 20 mm & diameter of 880mm.

4) Generally using materials are SA-516 GR.70 (CARBON STEEL) MATERIAL, HSLA but in this project selected for Pressure vessel is assigned two different materials such as one general materials NIMONIC 80A another one is Nimonic75 Material.

5) Analysis purpose using Ansys software we are choosing two type of analysis static and steady state thermal analysis.

6) Working Pressure 0.824 MPa is applied on the inner section wall of pressure vessel.

Working temperature is 200^{o}c is applied on the inner section wall of the pressure vessel

7) Finally identification Stress, deformation, Heat flux values as a result due to pressure is noted and concluded which material can sustain max pressure based on these values stress, deformation and heat flux.

3.2 METHODOLOGY:

1. To achieve the above objective the following methodology has been adopted in the present work.
2. A pressure vessel is select the two heads in this project hemi spherical, toriconical head and flat conditions
3. Modeling of the pressure vessel is done using CATIA software.
4. The model is imported to Ansys and analysis is preformed as follows.
5. Material properties are added.
6. Meshing is done, finally static and thermal boundary conditions are applied & it is solved.
7. After solution the results are viewed in general postprocessor and check stress, deformation and Heat flux.
8. Then the results from the analytical method Shown in graphical method concluded the suitable material

FIGURE 6METHODOLOGY

3.3 PROBLEM DEFINITION:

Improper design and material leads to the failure because Humidity, temperature, rain, wind, impurities and metal wet times have an effect on the pressure corrosion rate Corrosion reaction Basically the metallic pressure vessels are having good strength but due to their high weight to strength ratio and corrosive properties they are least preferred in aerospace as well as oil and gas industries. These industries are in need of pressure vessels which will have low weight to strength ratio without affecting the strength in this project pressure vessel with wall thickness of 20mm and diameter of 880mm is used with different different designs and materials is possible generally when the temperature is above 0°C and the relative humidity is over 80% (the surface is wet). Air impurities that dissolve in condensed water or rain water may accelerate corrosion.

FIGURE 7 PROBLEMS OF PRESSURE VESSELS

3.4 CAUSES OF PRESSURE VESSEL:

- Stress
- Faulty Design Operator error or poor maintenance
- Operation above max allowable
- working pressures
- Change of service condition
- Over temperature
- Safety valve
- Improper installation
- Corrosion
- Cracking
- Welding problems
- Erosion
- Fatigue
- Improper selection of materials or defects
- Low –water condition
- Improper repair of leakage
- Burner failure
- Improper installation Fabrication error
- Over pressurization
- Failure to inspect frequently enough Erosion
- Creep
- Embrittlement
- Unsafe modifications or alteration
- Unknown or under investigation

3.5 MAJOR MODELLED DIMENSIONS OF THE DEMO VESSEL:

Shell outside diameter D	**880 mm**
Shell length L	**1520mm**
Spherical head outside	**880mm**

diameter	
Corrosion allowance	**1.28mm**
Thickness of all ribs , tr	**16mm**
Distance b/w saddles, ds	**937.6mm**
Height of ribs, Hr	**470mm**
Width of rib Wr	**176mm**
Length of base plate	**815mm**
Saddle angle, 0	**120°**
Shell angle, (j)	**117.4°**
Thickness of all plates (shell), ts	**20 mm**

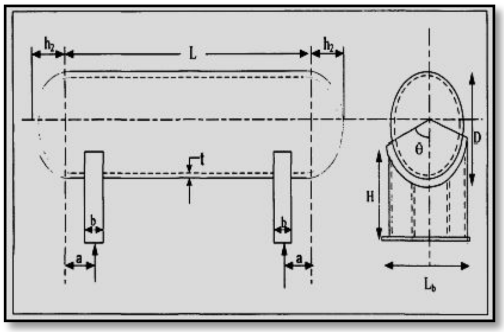


FIGURE 8 SPECIFICATION OF PRESSURE VESSEL

3.6 MATERIAL PROPERTIES:

Material	Density (g/cm³)	Young's Modulus (GPa)	Poisson's Ratio	Thermal Conductivity (W/(m·K))	Specific Heat Capacity (J/(kg·K))
High-Strength Low-Alloy Steel	7.85	200	0.3	45	460
Nimonic 75	8.37	220	0.33	11	410
SA-516 Gr.70 (Carbon Steel)	7.85	200	0.3	52	470
Nimonic 80A	8.19	220	0.33	11	410

FIGURE 9 MATERIAL PROPERTIES

3.7TAKING WORKING CONDITIONS:

Design Pressure= 0.824E+6 N/m^2

Design Temperature= 200^{0}c

3.8 DESIGN CALCULATION:

Moment Equation $M_1 = -Q \times a \times (1 - \dfrac{1 - \frac{a}{L} + \frac{R_m^2 - h_2^2}{2 \times A \times L}}{1 + \frac{4}{3}\frac{h_2}{L}})$

$$M_1 = -1549 \times 20.32 \times (1 - \frac{1 - \frac{20.32}{152} + \frac{428.64^2 - 421.28^2}{2 \times 20.32 \times 152}}{1 + \frac{4}{3} \times \frac{421.28}{152}})$$

$= -113.3 kgf - m$ ...... (1)

Moment Equation $M_2 = \dfrac{Q \times L}{4} \times (\dfrac{1 + \frac{2 \times (R_m^2 - h_2^2)}{L^2}}{1 + \frac{4}{3}\frac{h_2}{L}} - 4 \times \dfrac{a}{L})$

$$M_2 = \frac{1549 \times 152}{4} \times (\frac{1 + 2 \times (\frac{428.64^2 - 421.28^2}{152^2})}{1 + \frac{4}{3} \times \frac{421.28}{152}} - 4 \times \frac{20.32}{152})$$

$= 117.4 kgf - m$ .... (2)

The longitudinal membrane plus bending stresses in the cylindrical shell between the supports are given by the following equations.

Longitudinal stress at top of the shell $\sigma_1 = \dfrac{P \times R_m}{2 \times t} - \dfrac{M_2}{\pi \times R_m^2 \times t}$

$$\sigma_1 = \frac{8.45 \times 428.64}{2 \times 14.72} - \frac{117.4}{\pi \times 428.64^2 \times 14.72}$$

$= 121.61 kgf / cm^2$ .... (3)

Longitudinal stress at the bottom of the shell $\sigma_2 = \dfrac{P \times R_m}{2 \times t} + \dfrac{M_2}{\pi \times R_m^2 \times t}$

$$\sigma_2 = \frac{8.45 \times 428.64}{2 \times 14.72} + \frac{117.4}{\pi \times 428.64^2 \times 14.72}$$

$= 124.32 kgf / cm^2$ ..... (4)

are given by the following equations.

$$\sigma_3 = \frac{P \times R_m}{2 \times t} - \frac{M_1}{\pi \times R_m^2 \times t}$$

$$\sigma_3 = \frac{8.45 \times 428.64}{2 \times 14.72} - \frac{-113.3}{\pi \times 428.64^2 \times 14\ 72}$$

$= 124.32 kgf / cm^2$ ...... (5)

Longitudinal stress at the bottom of the shell at support $\sigma_4 = \dfrac{P \times R_m}{2 \times t} + \dfrac{M_1}{\pi \times R_m^2 \times t}$

$$\sigma_4 = \frac{8.45 \times 428.64}{2 \times 14.72} + \frac{-113.3}{\pi \times 428.64^2 \times 14.72}$$

$= 121.66 kgf / cm^2$ ...... (6)

T is the total shear force induced on the saddle support and it is determined by the following equation 7.

Maximum shear force in the saddle $T = \dfrac{Q \times (L - 2 \times a)}{L + \frac{4}{3} h_2}$

$$= \frac{1549 \times (152 - 2 \times 20.32)}{152 + \frac{4}{3} \times 421.28}$$

$= 828.7 kgf$ .... (7)

4. INTRODUCTION TO CATIA V5R20

4.1 INTRODUCTION TO CATIA V5R20:

Welcome to **CATIA (Computer Aided Three Dimensional Interactive Application)**. As a new user of this software package, you will join hands with thousands of users of this high-end CAD/CAM/CAE tool worldwide. If you are already familiar with the previous releases, you can upgrade your designing skills with the tremendous improvement in this latest release.

CATIA V5, developed by Dassault Systems, France, is a completely re-engineered, Next-generation family of CAD/CAM/CAE software solutions for Product Lifecycle Management.

Through its exceptionally easy-to-use and state-of-the-art user interface, CATIA V5 delivers innovative technologies for maximum productivity and creativity, from the inception concept to the final product. CATIA V5 reduces the learning curve, as it allows the flexibility of using feature-based and parametric designs.

CATIA V5 provides three basic platforms: P1, P2, and P3. P1 is for small and medium-sized process-oriented companies that wish to grow toward the large scale digitized product definition.

4.2 DESIGN PROCEDURE IN CATIA:

4.2.1 ELLIPTICAL HEAD : Go to the sketcher workbench select xy plane create the sectional view as per the dimensions length is 1520 height is 440 and create the offset distance is 440 apply fillet radius is 440 after go to the part design workbench apply shaft 360^{o} .

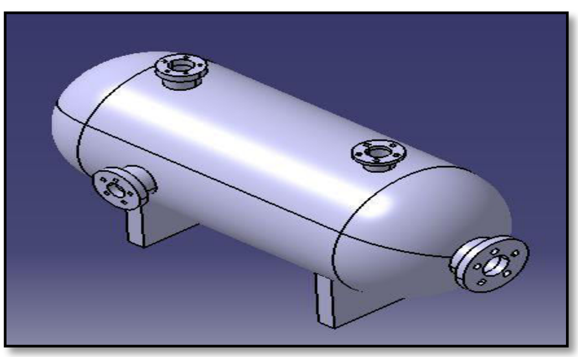


FIGURE 10 ELLIPTICAL HEAD IN CATIA

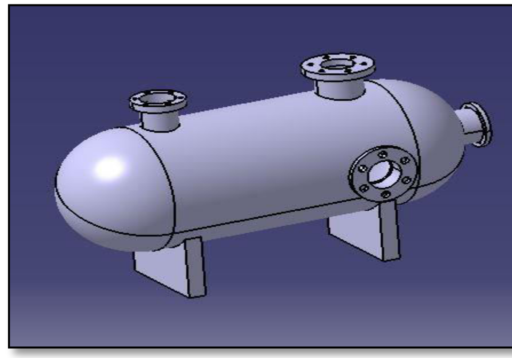


FIGURE 11 ISOMETRIC VIEW IN CATIA WORKBENCH

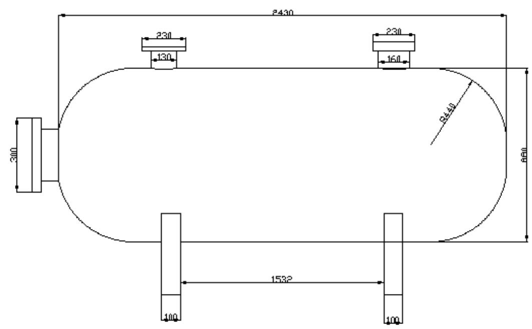


FIGURE 12 ELLIPTICAL HEAD DIMENSIONS

4.2.2 FLATE HEAD : Go to the sketcher workbench select xy plane create the sectional view as per the dimensions length is 1520 height is 440 and create the offset distance is 440 apply fillet radius is 50 after go to the part design workbench apply shaft 360^{o} .

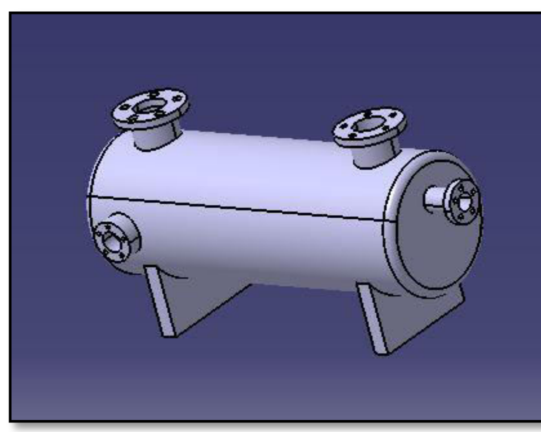


FIGURE 13 ISOMETRIC VIEW FLATE HEAD

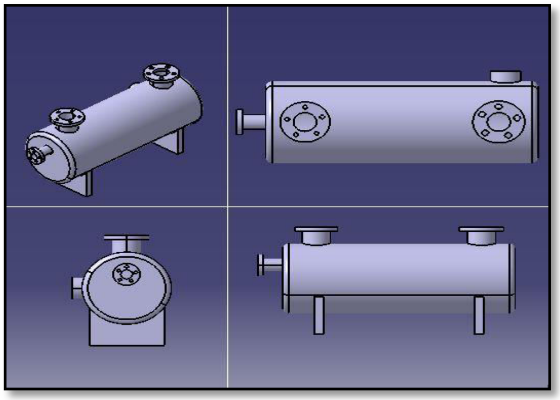


FIGURE 14 FLAT HEAD MULTIVIEWS

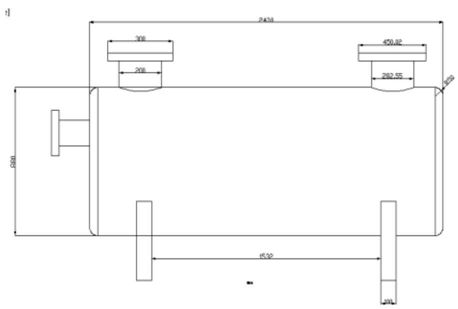


FIGURE 15 DIMENSIONS OF FLATE HEAD PRESSURE VESSEL

5. INTRODUCTION TO ANSYS

5.1 INTRODUCTION TO ANSYS:

ANSYS is a large-scale multipurpose finite element program developed and maintained by ANSYS Inc. to analyze a wide spectrum of problems encountered in engineering mechanics**.**

5.2 PROGRAM ORGANIZATION:

The ANSYS program is organized into two basic levels:

- Begin level
- Processor (or Routine) level

The Begin level acts as a gateway into and out of the ANSYS program. It is also used for certain global program controls such as changing the job name, clearing (zeroing out) the database, and copying binary files. When you first enter the program, you are at the Begin level. At the Processor level, several processors are available.

6. FINITE ELEMENT METHOD

6.1 INTRODUCTION:

The Basic concept in FEA is that the body or structure may be divided into smaller elements of finite dimensions called "Finite Elements". The original body or the structure is then considered as an assemblage of these elements connected at a finite number of joints called "Nodes" or "Nodal Points".

Simple functions are chosen to approximate the displacements over each finite element. Such assumed functions are called "shape functions". This will represent the displacement with in the element in terms of the displacement at the nodes of the element.

6.2 ANALYSIS PROCEDURE IN ANSYS:

Designed component in CATIA V5 workbench after imported into ANSYS workbench now select the steady state thermal ANALYSIS.

1. ENGINEEERING MATERIALS (MATERIAL PROPERTIES).

2. CREATE OR IMPORT GEOMENTRY.

3. MODEL (APPLY MESHING).

4. SET UP (BOUNDARY CONDITIONS).

5. SOLUTION.

6. RESULT.

6.3 MESHING:

6.3.1 HEMISPHERICAL HEAD:

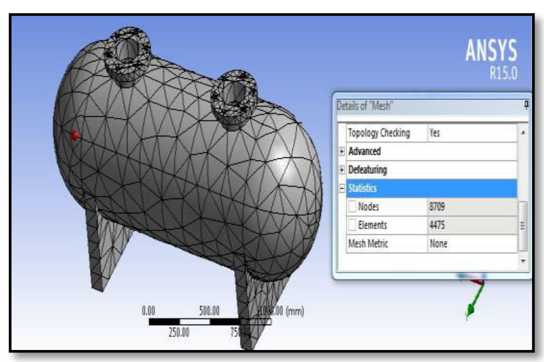


FIGURE 16 HEMISPHERICAL HEAD PRESSURE VESSEL MESH

6.3.2 FLAT HEAD:

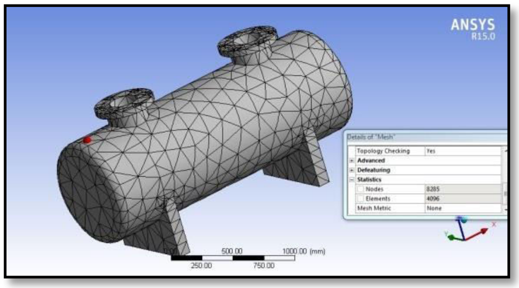


FIGURE 17 FLAT HEAD PRESSURE VESSEL MESH

6.3.3 TORICONICAL HEAD:

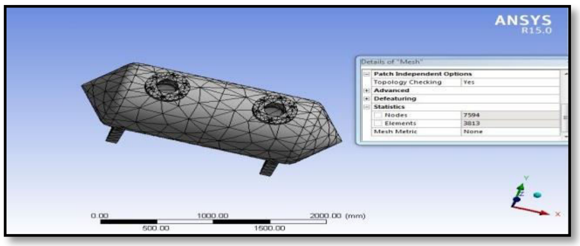


FIGURE 18 TORICONICAL HEAD PRESSURE VESSEL (MESH: NODES 7594, ELEMENTS: 3813)

6.4 BOUNDARY CONDITIONS:

1. Maximum working pressure load apply at inside on pressure vessel surface of the 0.824 Mpa.

2. Temperature apply at the inside on pressure vessel surface top surface 200^{o}c.

3. Fixed the saddles Bottom of the pressure vessel.

6.4.1 HEMISPHERICAL HEAD:

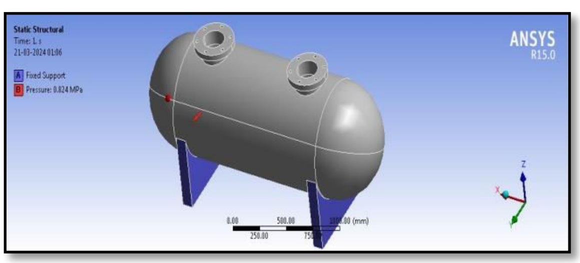


FIGURE 19 BOUNDARY CONDITION OF HEMISPHERICAL HEAD PRESSURE 0.824

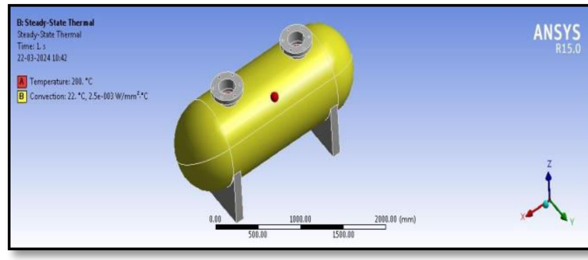


FIGURE 20 BOUNDARY CONDITION OF HEMISPHERICAL TEMPERATURE 200OC

6.4.2 FLAT HEAD:

FIGURE 21 BOUNDARY CONDITION OF FLAT HEAD PRESSURE 0.824

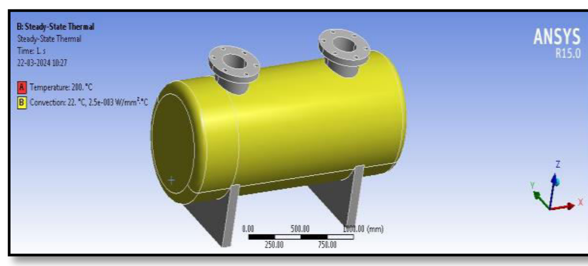


FIGURE 22 BOUNDARY CONDITION OF FLAT HEAD TEMPERATURE 200OC

6.4.3 TORICONICAL HEAD:

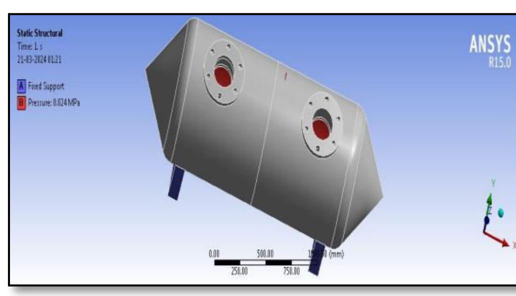


FIGURE 23 BOUNDARY CONDITION OF TORICONICAL HEAD PRESSURE 0.824

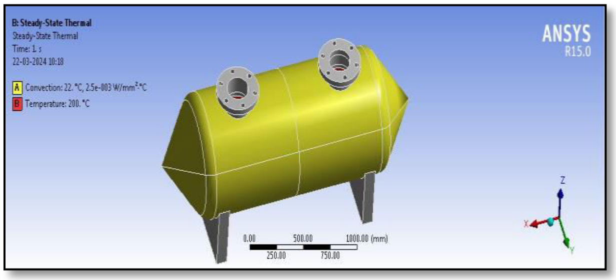


FIGURE 24 BOUNDARY CONDITION OF TORICONICAL HEAD TEMPERATURE 200OC

7. RESULTS AND DISCUSSIONS

7.1 FLAT HEAD PRESSURE VESSEL

7.1.1 CARBON STEEL MATERIAL

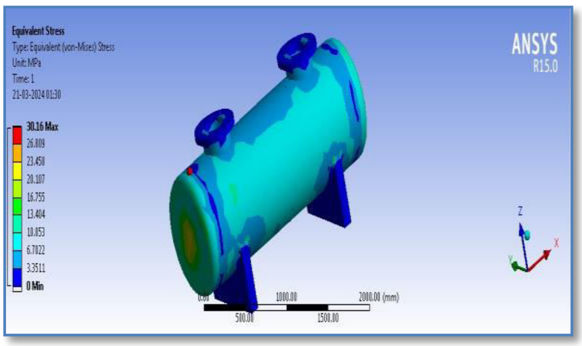


FIGURE 25 VON MISSES STRESSES OF FLAT HEAD CARBON STEEL MATERIAL

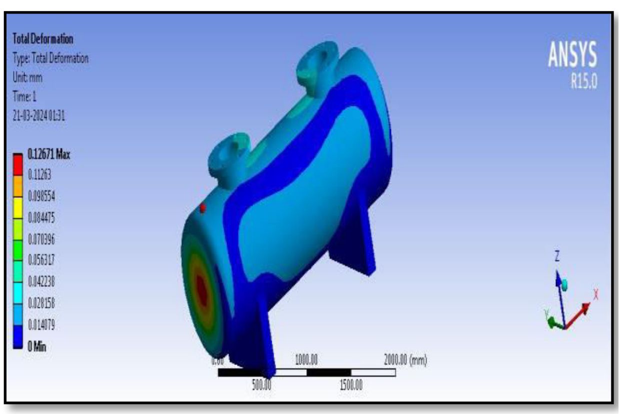


FIGURE 26 TOTAL DEFORMATION OF FLAT HEAD CARBON STEEL MATERIAL

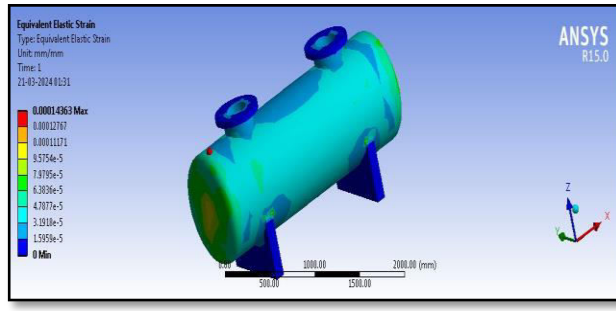


FIGURE 27 STRAIN OF FLAT HEAD CARBON STEEL MATERIAL

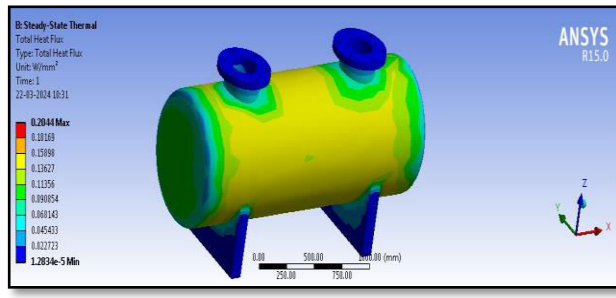


FIGURE 28 TOTAL HEAT FLUX OF FLAT HEAD CARBON STEEL MATERIAL

7.1.2 HSLA

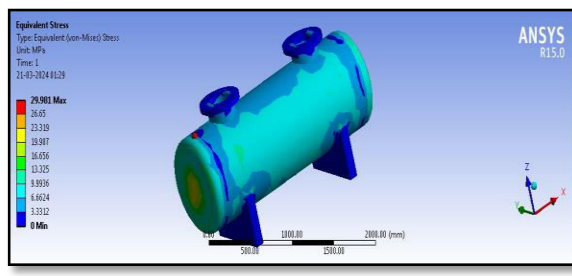


FIGURE 29 VON MISSES STRESSES OF FLAT HEAD HSLA MATERIAL

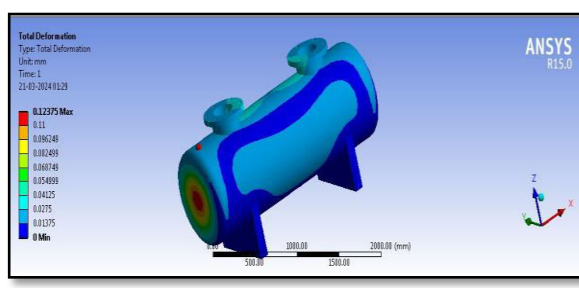


FIGURE 30 TOTAL DEFORMATION OF FLAT HEAD HSLA MATERIAL

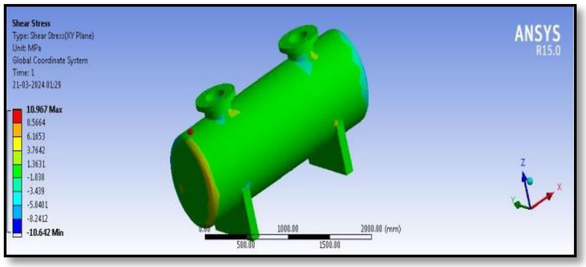


FIGURE 31 SHEAR STRESS OF FLAT HEAD HSLA MATERIAL

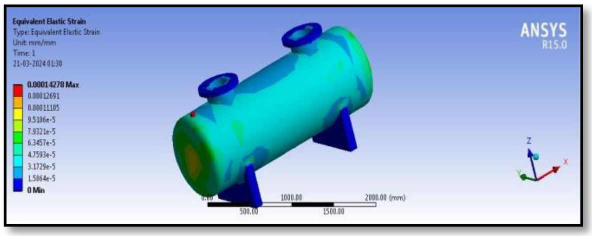


FIGURE 32 STRAIN OF FLAT HEAD HSLA MATERIAL

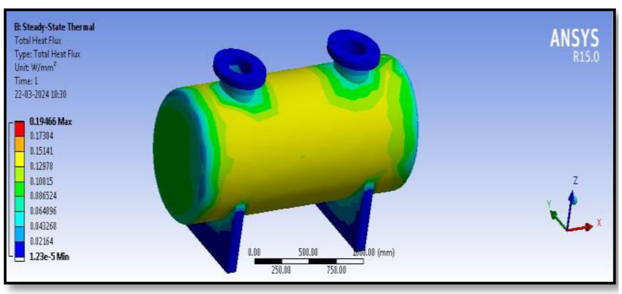


FIGURE 33 TOTAL HEAT FLUX OF FLAT HEAD HSLA MATERIAL

7.1.3 NIMONIC 75 MATERIAL

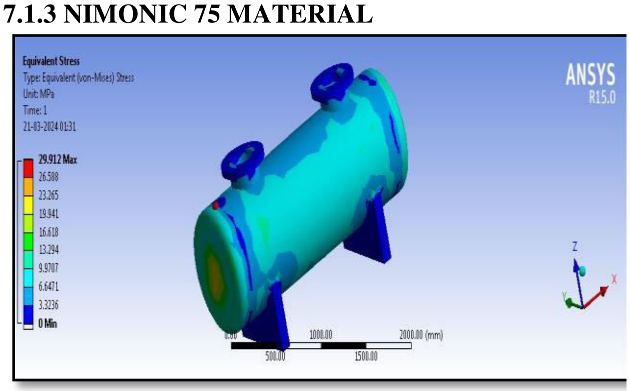


FIGURE 34 VON MISSES STRESSES OF FLAT HEAD NIMONIC 75 MATERIAL

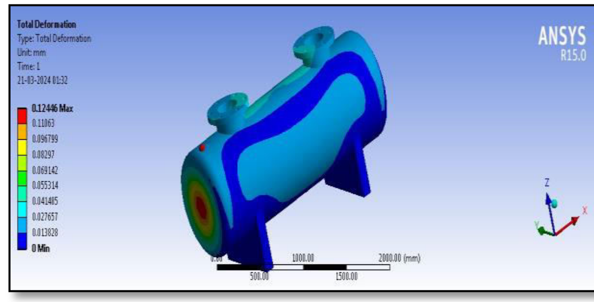


FIGURE 35 TOTAL DEFORMATION OF FLAT HEAD NIMONIC 75 MATERIAL

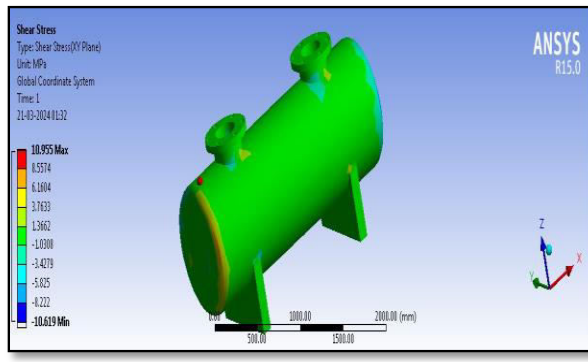


FIGURE 36 SHEAR STRESS OF FLAT HEAD NIMONIC 75 MATERIAL

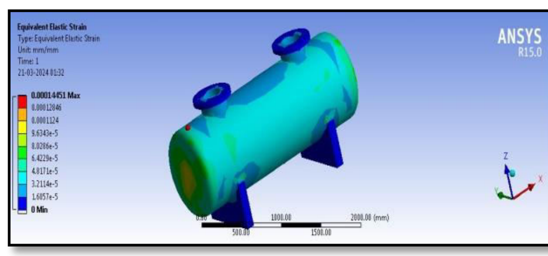


FIGURE 37 STRAIN OF FLAT HEAD NIMONIC 75 MATERIAL

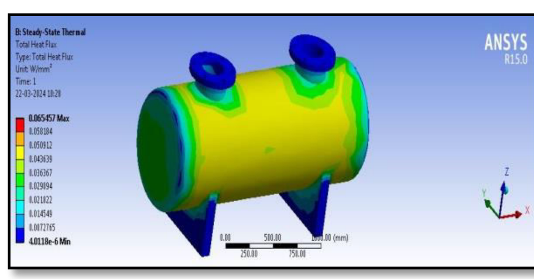


FIGURE 38 TOTAL HEAT FLUX OF FLAT HEAD NIMONIC 75 MATERIAL

7.1.4 NIMONIC 80A MATERIAL

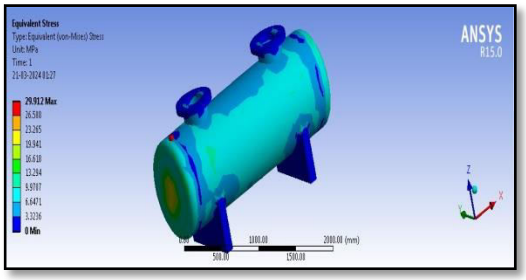


FIGURE 39 VON MISSES STRESSES OF FLAT HEAD NIMONIC 80 MATERIAL

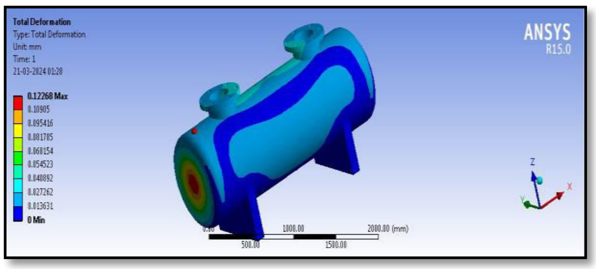


FIGURE 40 TOTAL DEFORMATION OF FLAT HEAD NIMONIC 80 MATERIAL

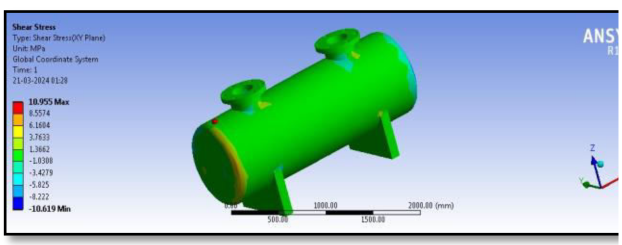


FIGURE 41 SHEAR STRESS OF FLAT HEAD NIMONIC 80 MATERIAL

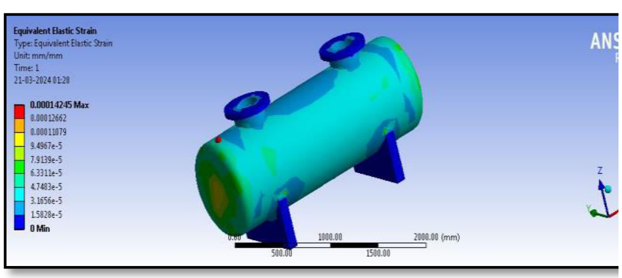


FIGURE 42 STRAIN OF FLAT HEAD NIMONIC 80 MATERIAL

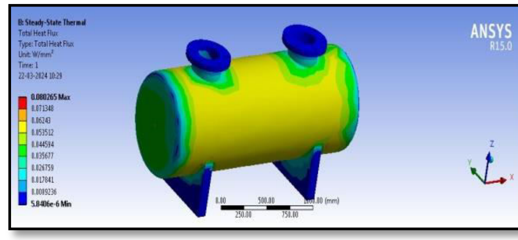


FIGURE 43 TOTAL HEAT FLUX OF FLAT HEAD NIMONIC 80 MATERIAL

7.2 HEMISPHERICAL HEAD:

7.2.1 CARBON STEEL

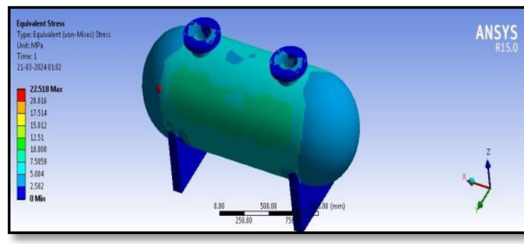


FIGURE 44 VON MISSES STRESSES OF HEMISPHERICAL HEAD CARBON STEEL MATERIAL

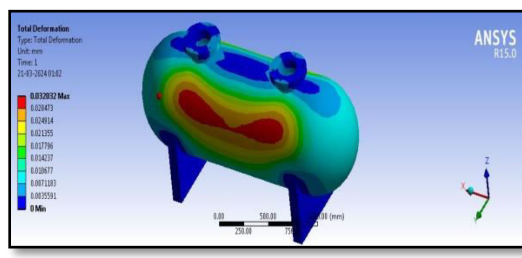


FIGURE 45 TOTAL DEFORMATION OF HEMISPHERICAL HEAD CARBON STEEL MATERIAL

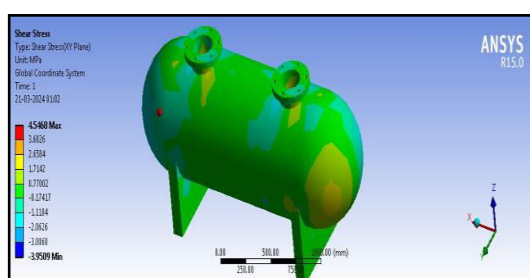


FIGURE 46 SHEAR STRESS OF HEMISPHERICAL HEAD CARBON STEEL MATERIAL

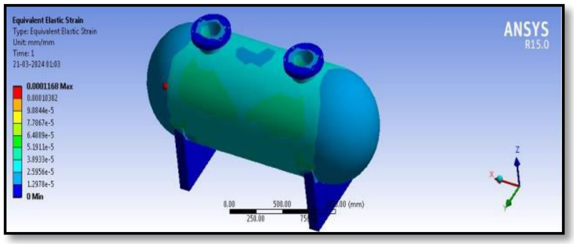


FIGURE 47 STRAIN OF HEMISPHERICAL HEAD CARBON STEEL MATERIAL

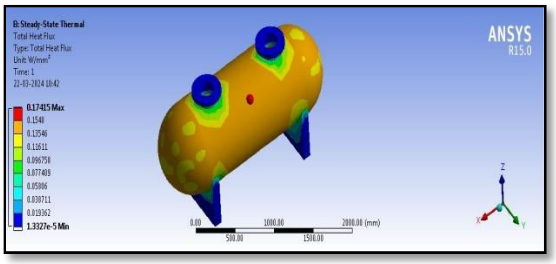


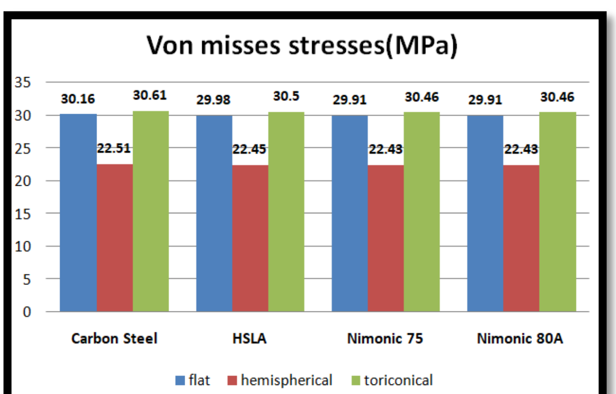
FIGURE 48 TOTAL HEAT FLUX OF HEMISPHERICAL HEAD CARBON STEEL MATERIAL

7.3 STATIC STRUCTURAL RESULTS

The below tabulated data is the maximum results obtained by all materials with the all three head shapes like flat, hemi spherical and toriconical shapes. The static structural von misses stresses, total deformation, shear stress and total deformation values are tabulated in below.

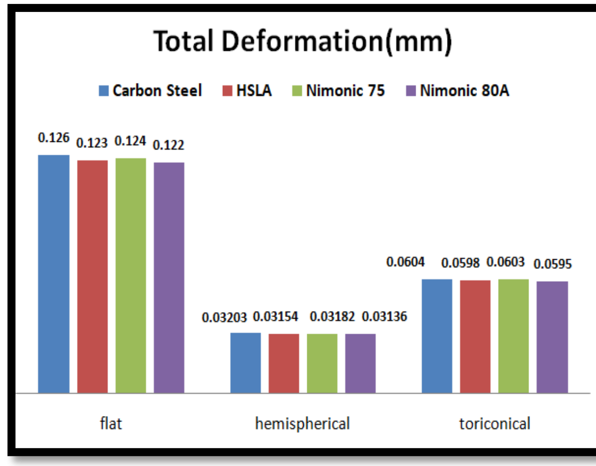
7.4 GRAPHS

7.4.1 VON MISSES STRESSES GRAPH:



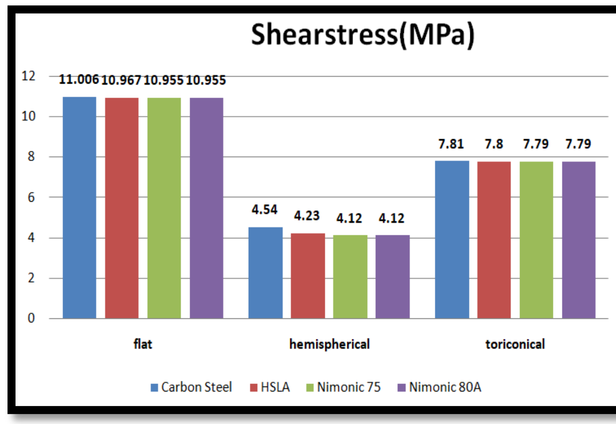
GRAPH:1VON MISSES STRESSES

7.4.2 TOTAL DEFORMATION GRAPH:



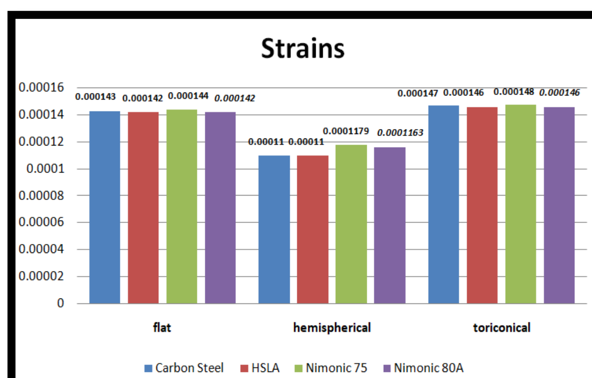
GRAPH:2 TOTAL DEFORMATION

7.4.3 SHEAR STRESS GRAPH:



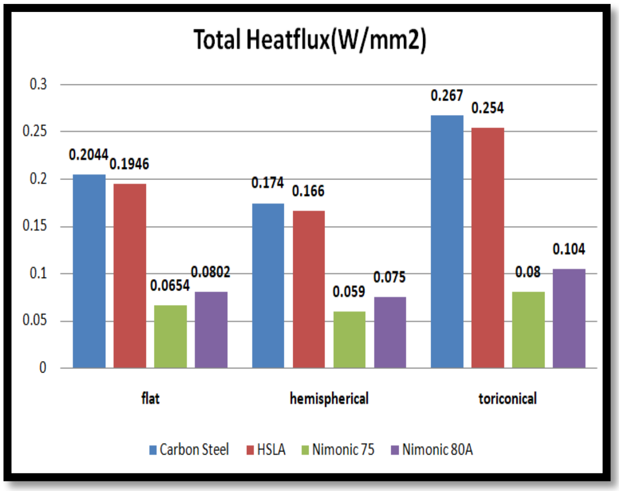
GRAPH:3 SHEAR STRESSES

7.4.4 STRAIN GRAPH



GRAPH:4 STRAIN GRAPH

7.4.5 TOTAL HEATFLUX GRAPH



GRAPH:5 TOTAL HEAT FLUX

8 CONCLUSION

The following conclusions have been drawn from the present work.

The pressure may be from an external source, or by application of heat from a direct or indirect source. Modeling of horizontal pressure vessels Flat head, Hemispherical head, Toriconicalhead is done by using CATIA Software and then the model is imported into ANSYS Software for Structural and thermal analysis on pressure vessel to check the quality of materials such as, Four different materials SA-516 GR.70 (CARBON STEEL) MATERIAL, NIMONIC 75 MATERIAL, HSLA, NIMONIC 80A, Generally pressure vessels are made up of haste alloy materials. From the obtained Von-misses stresses, strain, total deformation, shear stress and heat flux for the materials, respectively Compared with four different materials with different heads. Finally Nimonic80A material have less stresses, deformations, and heat flux values .Finally from structural analysis and thermal analysis based on results it is concluded that with holes Nimonic80A material is suitable material for pressure vessel material because of NIMONIC alloys are primarily composed of nickel and chromium. These alloys are known for their high-temperature low-creep and high performance. NIMONIC alloy 80A is a wrought, age-hardened alloy that is strengthened by additives like titanium, aluminum and carbon. It is manufactured by high-frequency melting and casting in air. It is similar to NIMONIC alloy 80A it has good corrosion and oxidation resistance than it is suitable for manufacturing process

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