PRESSURE ANALYSIS OF THIN-CYLINDRICAL PRESSURE VESSEL

A PROJECT REPORT SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE AWARD OF THE DEGREE OF BACHELOR OF ENIGINEERING IN MECHANICAL ENIGINEERING

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DEPARTMENT OF MECHANICAL ENGINEERING

ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY & SCIENCES
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Sangivalasa, Bheemunipatnam (Mandal), Visakhapatnam (Dist.)

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ANIL NEERUKONDA INSTITUTE OF TECHNOLOGY & SCIENCES (A)
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CERTIFICATE

This is to certify that the Project Report entitled "**PRESSURE ANALYSIS OF THIN CYLINDRICAL PRESSURE VESSEL**" being submitted by POSNI PUNEETH (318126520L01), VOONNA AKHIL (317126520059), POLISETTI SRIKAR (317126520041), GADEPALLI SAI AVINASH KUMAR (317126520019), AKUMARTHI PRAVEEN KUMAR (317126520001) in partial fulfillments for the award of degree of **BACHELOR OF TECHNOLOGY** in **MECHANICAL ENGINEERING, ANITS**. It is the work of bona-fide, carried out under the guidance and supervision of **DR.RAJESH GHOSH**, Associate Professor, Department Of Mechanical Engineering, ANITS during the academic year of 2017-2021.

PROJECT GUIDE	Approved By HEAD OF THE DEPARTMENT
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Last but not least, we would like to convey our thanks to all who have contributed either directly or indirectly for the completion of work.

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CHAPTER-1

INTRODUCTION

1.1 ABSTRACT

A pressure vessel is a closed container designed to hold gases or liquids at a pressure and temperature substantially different from ambient pressure and temperature. The cross-section of the pressure vessel may be circular or square with flat end covers, reinforced by a gate mechanism on both sides. In the present study the vessel has been optimized for thickness by considering stress level for different materials on the shell areas for cylindrical shape.

The pressure vessel designed as per the ASME code Section VIII and then checked for the stress patterns across the walls of vessel for the applied pressure. The complete analysis i.e., pressure tests are carried out using FEA based software platform (Solid works 3D design & Analysis platform). At first on the basis of observation it has been tried to compare the validity of pressure vessel shape. Then tried to increase the thickness of the shell by applying the same amount of pressure and for different materials, so as to obtain an optimal thickness of pressure vessel with suitable material.

Thus, observing both the results we have come to a conclusion to decide the most valid shape & thickness of shell required for an optimal pressure vessel. The literature survey indicates that so far, many works has been done on different topics & subjects related to pressure vessel optimization by FEA based technique of analysis, but there are very few works done to compare the optimality of shape of pressure vessel shell by FEA analysis. The discussion on the results, conclusion & the scope of further work has also been manifested at the end of the work.

1.2 PRESSURE VESSELS

A P**ressure Vessel** is a container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. Vessels, tanks and pipelines that carry, store, or receive fluids are called Pressure Vessels.

A pressure vessel is defined as a container with a pressure differential between inside and outside. The inside pressure is usually higher than the outside, except for some isolated situations. When discussing pressure vessels, we must also consider tanks. Pressure vessels and tanks are significantly different in both design and construction: tanks, unlike Design and Analysis of Pressure Vessel Using Ansys, Pressure vessels, are limited to atmospheric pressure; and pressure vessels often have internals while most tanks do not (and those that do are limited to heating coils or mixers).

The pressure differential is dangerous, and fatal accidents have occurred in the history of pressure vessel development and operation. Consequently, pressure vessel design, manufacture, and operation are regulated by engineering authorities backed by legislation.
For these reasons, the definition of a pressure vessel varies from country to country, but involves parameters such as maximum safe operating pressure and temperature, and are engineered with a safety factor, corrosion allowance, minimum design temperature (for brittle fracture), and involve non-destructive testing, such as ultrasonic testing, radiography, and pressure tests, usually involving water, also known as a hydrotest, but could be pneumatically tested involving air or another gas.

The preferred test is hydrostatic testing because it’s a much safer method of testing as it releases much less energy if fracture were to occur (water does not rapidly increase its volume while rapid depressurization occurs, unlike gases like air, i.e., gasses fail explosively). In the United States, as with many other countries, it is the law that vessels over a certain size and pressure (15 PSIg) be built to Code, in the United States that Code is the ASME Boiler and Pressure Vessel Code (BPVC), these vessels also require an Authorized Inspector to sign off on every new vessel constructed and each vessel has a nameplate with pertinent information about the vessel such as maximum allowable working pressure, maximum temperature, minimum design metal temperature, what company manufactured it, the date, it's registration number (through the National Board), and ASME's official stamp for pressure vessels (U-stamp), making the vessel traceable and officially an ASME Code vessel.

1.3 TERMINOLOGY

Code: The complete rules for construction of pressure vessels as identified in ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, Pressure Vessels.

Construction: The complete manufacturing process, including design, fabrication, inspection, examination, hydrotest, and certification. Applies to new construction only.

Hoop membrane stress: The average stress in a ring subjected to radial forces uniformly distributed along its circumference.

Longitudinal stress: The average stress acting on a cross section of the vessel.

Pressure vessel: A leak-tight pressure container, usually cylindrical or spherical in shape, with pressure usually varying from 15 psi to 5000 psi.

Stress concentration: Local high stress in the vicinity of a material discontinuity such as a change in thickness or an opening in a shell.

Weld efficiency factor: A factor which reduces the allowable stress. The factor depends on the degree of weld examination performed during construction of the vessel.

ASME: American Society of Mechanical Engineers

1.4 TYPES OF PRESSURE VESSELS

Pressure vessels can be classified according to their intended service, temperature and pressure, materials and geometry. Different types of pressure vessels can be classified as follows.

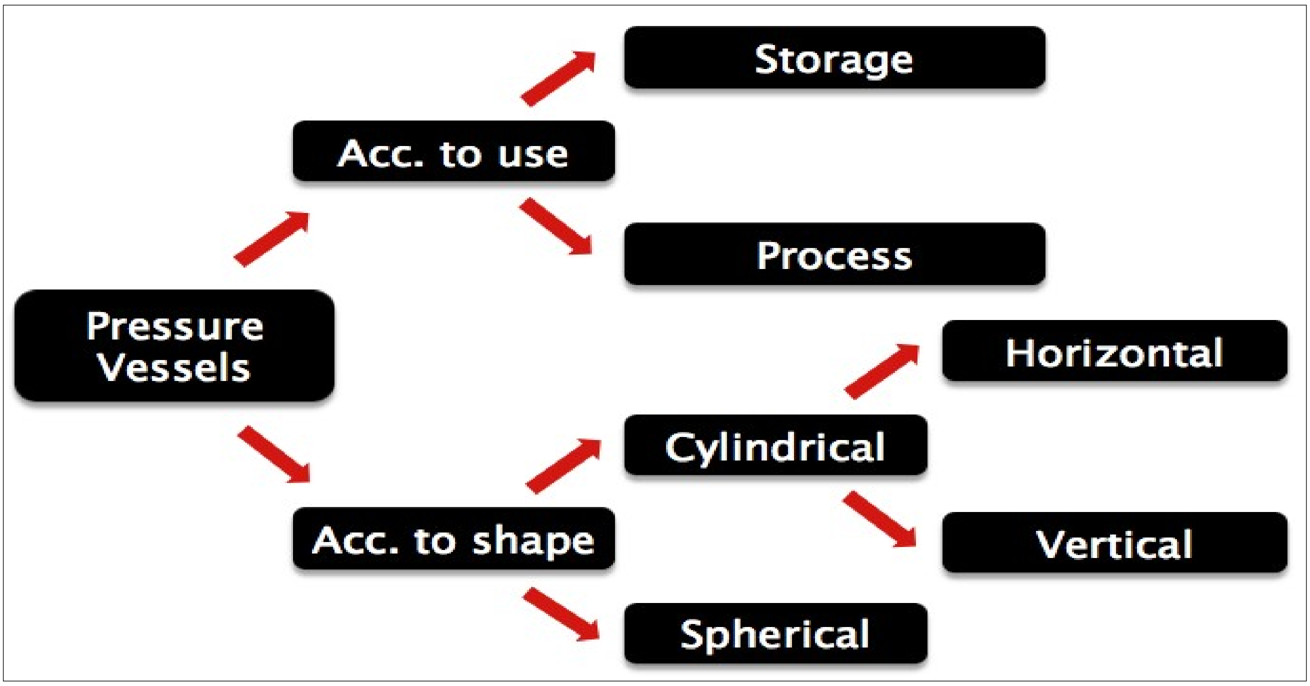


Figure 1: Types based on Categories

According to the intended use of the pressure vessel, they can be divided into storage containers and process vessels.

The first classes are only used for storing fluids under pressure, and in accordance with the service are known as storage tanks. Process pressure vessels have multiple and varied uses, among them we can mention heat exchangers, reactors, fractionating towers, distillation towers, etc.

According to the shape, pressure vessel may be cylindrical or spherical.

The former may be horizontal or vertical, and in some cases may have coils to increase or lower the temperature of the fluid.

Figure 2: Spherical Pr. Vessel

Figure 3: Cylindrical Pr. Vessel

Spherical pressure vessels are usually used as storage tanks, and are recommended for storing large volumes. Since the spherical shape is the "natural" form bodies adopt when subjected to internal pressure, this would be the most economical way to store pressurized fluids. However, the manufacture of such containers is much more expensive compared with cylindrical containers.

1.5 PRESSURE VESSEL PARTS

The following two sample vessels are presented: vertical and horizontal. In both cases the main parts are shown:

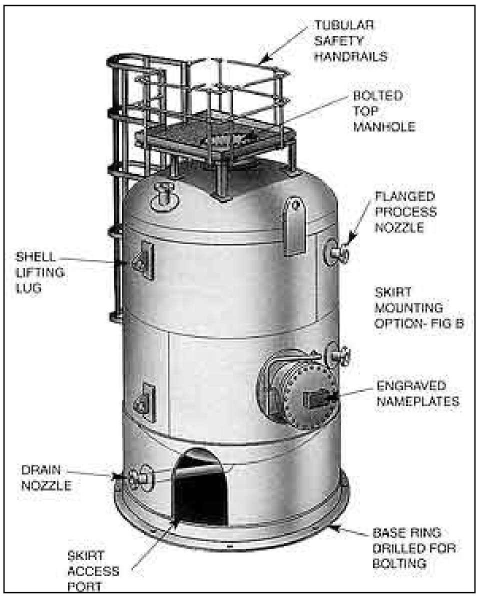


Figure 4. Vertical Pr. Vessel

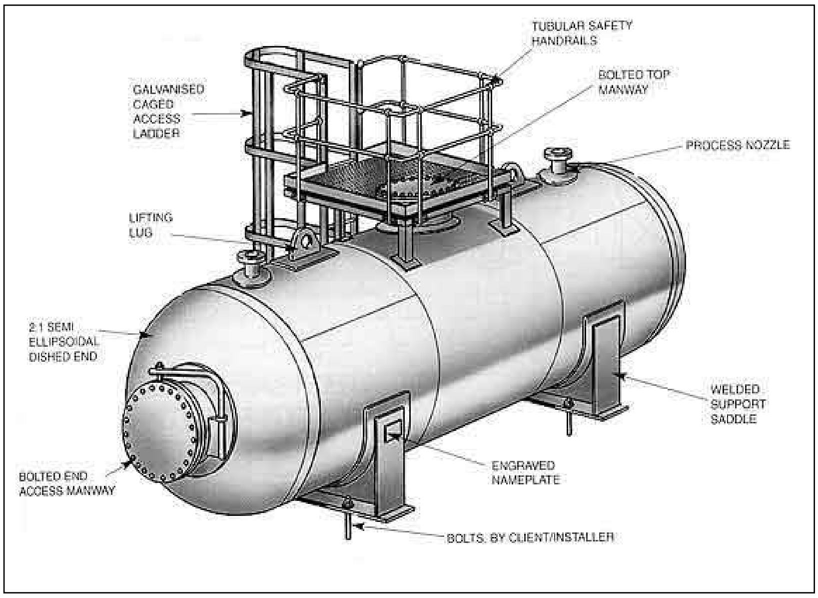


Figure 5. Horizontal Pr. Vessel

Geometry definition

To define the geometry of a pressure vessel, the inner diameter of the equipment and the distance between tangent lines is used.

The inner diameter should be used, since this is a process requirement.

- **Welding line**: point at which the head and shell are welded
- **Tangent line**: point at which the curvature of the head begins

Depending on the head fabrication method, heads come with a straight skirt.

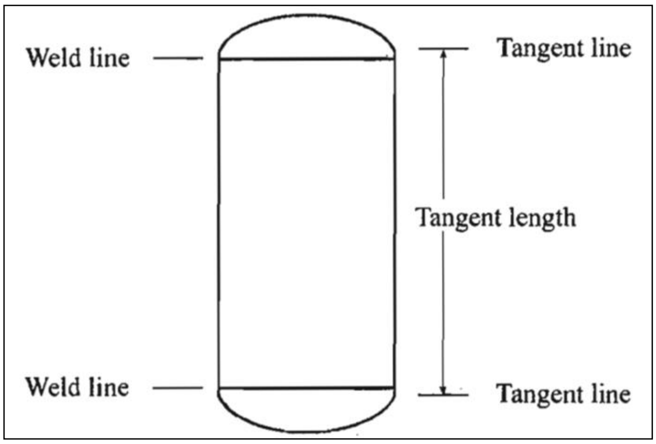


Figure 6.

To set the length of the pressure vessel (regardless the type of heads), the distance between tangent lines is used since this distance is not dependent on the head manufacturing method. It is very rare that the weld and tangent lines coincide.

1.6 CAUSES OF FAILURE:

The pressure differential in pressure vessel is dangerous and many fatal accidents have occurred in the history of pressure vessel development and operation. So, we have to design the shell wall thick enough & check the stress level on shell wall so as to avoid failure of pressure vessel. Also, we should keep in mind that due to heat transfer, there will be condensation of steam inside vessel, which we have to avoid by placing suitable insulation layer around the vessel exterior walls.

The main causes of failure of a pressure vessel are as follows:

- 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

CHAPTER-2

MODELLING AND ANALYSIS OF PRESSURE VESSEL

2.1 3D CAD SOLIDWORKS: -

Solid works mechanical design automation software is a feature-based parametric solid modelling design tool which takes advantage of the easy to learn Windows graphical user interface. You can create fully associative 3-D solid models with or without constraints while utilizing automatic or user defined relations to capture design intent.

2.1.1. FEATURE-BASED

Just as an assembly is made up of a number of individual pieces parts, a Solid works model is also made up of individual constituent elements. These elements are called Features.

When you create a model using the Solid works software, you work with intelligent, easy to understand geometric features such as bosses, cuts, holes, ribs, fillets, chamfers and drafts. As the feature are created, they are applied directly to work piece.

Features can be classified as sketched or applied: -

- Sketched features: Based upon a 2-D sketch. Generally, that sketch is transformed into a solid by extrusion, rotation, sweeping or lofting.
- Applied Features: Created directly on solid model. Fillets and chamfers are example of this type of feature.

The Solid works software graphically shows you the feature-based structure of your model in a special window called the Feature Manager design tree. The Feature Manager design tree not only shows you the sequence in which features were created, it gives you easy access to all the underlying associated information.

2.1.2. PARAMETRIC: -

The dimensions and relations used to create a feature are capture and stored in the model. This is not only enabling you to capture

your design intent, it also enables you to quickly and easily make changes to model.

- Driving dimensions: These are dimensions used when creating a feature. They include the dimensions associated with the sketch geometry, as well as those associated with the feature itself. A simple example of this would be a feature like cylindrical boss. The diameter of the boss is controlled by the diameter of sketched circle. The height of the boss is controlled by the depth to which that circle was extruded when the feature was made.

- Relations: These include such information as parallelism, tangency and concentricity. Historically this type of information has been communicated on drawings via feature-controlled symbols. By capturing this in the sketch, Solid works enables you to fully capture your design intent upfront, in the model.

2.1.3. 3D SOLID MODELING OVERVIEW: -

A solid model is the most complete type of geometric model used in the CAD systems. It contains all the wireframe and surface geometry necessary to fully describe the edges and faces of the model. In addition to the geometric information, it has the information called topology that relates the geometry altogether. An example of topology would be which faces (surfaces) meet at which edge (curve). This intelligence makes operations such as filleting an easy as selecting an edge and specifying a radius.
3D solid modelling with SOLIDWORKS speeds the creation of complex parts and large assemblies.

Creating 3D solid models of your designs instead of 2D drawings:

- speeds design development and detailing
- improves visualization and communication
- eliminates design interference issues
- checks design functionality and performance (without the need for physical prototypes)
- automatically provides manufacturing with 3D solid models that are required when programming CNC machine tools and rapid prototyping equipment.

With SOLIDWORKS automatic drawing updates, you don’t have to worry about modifications.
All 2D drawing views are automatically created from, and linked to, the 3D solid model. If the 3D solid model is modified, the 2D drawing views and details automatically update. This automatic associativity means that the solid model is always synchronized with your 2D documentation.
Key SOLIDWORKS 3D solid modelling features enable you to:

- Create 3D solid models of any part and assembly, no matter how large or complex
- Keep all 3D models, 2D drawings, and other design and manufacturing documents synchronized with associativity that automatically tracks and makes updates
- Quickly make variations of your designs by controlling key design parameters
- Directly edit your model by simply clicking and dragging model geometry
- Generate surfacing for any 3D geometry, even complex organic and stylized shapes
- Instantly analyze your 3D model for any solid mass properties and volume (mass, density,volume, moments of inertia, and so forth.

2.2 FINITE ELEMENT MODELING AND ANALYSIS PROCESS OF PRESSURE VESSEL: -

FINITE ELEMENT MODELING

SOLIDWORKS Simulation uses the displacement formulation of the finite element method to calculate component displacements, strains, and stresses under internal and external loads. The geometry under analysis is discretized using tetrahedral (3D), triangular (2D), and beam elements, and solved by either a direct sparse or iterative solver. SOLIDWORKS Simulation also offers the 2D simplification assumption for plane stress, plane strain, extruded, or axisymmetric options. SOLIDWORKS Simulation can use either an or p adaptive element type, providing a great advantage to designers and engineers as the adaptive method ensures that the solution has converged. In order to streamline the model definition, SOLIDWORKS Simulation automatically generates a shell mesh (2D) for the following geometries.

2.2.1. SHEET METAL BODY:

SOLIDWORKS Simulation assigns the thickness of the shell based on the 3D CAD sheet metal thickness, so, Product Designers can leverage the 3D CAD data for Simulation purposes.

2.2.2. SURFACE BODY:

For shell meshing, SOLIDWORKS Simulation offers a productive tool, called the Shell Manager, to manage multiple shell definitions of your part or assembly document. It improves the workflow for organizing shells according to type, thickness, or material, and allows for a better visualization and verification of shell properties. SOLIDWORKS Simulation also offers the 2D simplification assumption for plane stress, plane strain, extruded, or axisymmetric options. Product Engineers can simplify structural beams to optimize performance in Simulation to be modelled with beam elements. Straight, Curved, and tapered Beams are supported. SOLIDWORKS Simulation automatically converts structural members that are created as weldment features in 3D CAD as beam elements for quick setup of the simulation model.

SOLIDWORKS Simulation can use either an h or p adaptive element type, providing a great advantage to designers and engineers, as the adaptive method ensures that the solution has converged. Product Engineers can review the internal mesh elements with the
Mesh Sectioning Tools to check the quality of the internal mesh and make adjustments to mesh settings before running the study. Users can specify local mesh control at vertices, edges, faces, components and beams for a more accurate representation of the geometry. Integrated with SOLIDWORKS 3D CAD, finite element analysis using SOLIDWORKS Simulation knows the exact geometry during the meshing

process. And the more accurately the mesh matches the product geometry, the more accurate the analysis results will be.

2.3. FINITE ELEMENT ANALYSIS (FEA)

Since the majority of industrial components are made of metal, most FEA calculations involve metallic components. The analysis of metal components can be carried out by either linear or nonlinear stress analysis. Which analysis approach you use depends upon how far you want to push the design:
If you want to ensure the geometry remains in the linear elastic range (that is, once the load is removed, the component returns to its original shape), then linear stress analysis may be applied, as long as the rotations and displacements are small relative to the geometry. For such an analysis, factor of safety (FoS) is a common design goal. Evaluating the effects of post yield load cycling on the geometry, a nonlinear stress analysis should be carried out. In this case, the impact of strain hardening on the residual stresses and permanent set (deformation) is of most interest.
The analysis of non-metallic components (such as, plastic or rubber parts) should be carried out using nonlinear stress analysis methods, due to their complex load deformation relationship. SOLIDWORKS Simulation uses FEA methods to calculate the displacements and stresses in your product due to operational loads such as:

- Forces
- Pressures
- Accelerations
- Temperatures
- Contact between components

Loads can be imported from thermal, flow, and motion Simulation studies to perform Multiphysics analysis.

2.4. MESH DEFINITION

SOLIDWORKS Simulation offers the capability to mesh the CAD geometry in tetrahedral (1st and 2nd order), triangular (1st and 2nd order), beam, and truss elements. The mesh can consist of one type of elements or multiple for mixed mesh. Solid elements are naturally suitable for bulky models. Shell elements are naturally suitable for modelling thin parts (such as sheet metals), and beams and trusses are suitable for modelling structural members. As SOLIDWORKS Simulation is tightly integrated inside SOLIDWORKS 3D CAD, the topology of the geometry is used for mesh type:

- Shell mesh is automatically generated for sheet metal model and surface bodies.
- Beam elements are automatically defined for structural members.

So, their properties are seamlessly leveraged for FEA. To improve the accuracy of results in a given region, the user can define Local Mesh control for vertices, points, edges, faces, and components.

SOLIDWORKS Simulation uses two important checks to measure the quality of elements in a mesh:

- Aspect Ratio Check
- Jacobian Points

In case of mesh generation failure, SOLIDWORKS Simulation guides the users with a failure diagnostics tool to locate and resolve meshing problems. The Mesh Failure Diagnostic tool renders failed parts in shaded display mode in the graphics area.

2.5. ANALYSIS PROCESS: -

2.5.1. PRE-PROCESSING

Pre-processing comprises of building, meshing and loading the model created.

- Define type of Analysis.

Solid works provide wide variety of analysis for real life problem for mechanical and other engineering problems. Static Structural analysis is used for solving current problem.

- Define Engineering Data for Analysis.

The material that is considered for the shell as well as nozzle is SS304; it is having mechanical properties like young's modulus of 193-200MPa

- Define Boundary Condition for Analysis.

All the degrees of freedom of the pressure vessel are arrested at the right-side edges at shell and head joint location for all models of pressure vessel under study throughout the thesis.
The magnitude of the pressure considered for at all internal faces.
Mesh Statics:
Type of Element: Tetrahedrons.

2.5.2. SOLVING THE MODEL:

With all parts of the model defined, nodes, element, restraints and loads, the analysis part of the model is ready to begin. The system can determine approximately the values of stresses, deflection, temperature, pressure and vibration.
An analysis requires the following information:

- Nodal point
- Element connecting the nodal points

- Material and its physical properties
- Boundary conditions, which consists of loads and constraints

Analysis options: how the problem will be evaluated.

2.5.3. POST-PROCESSING:

The post-processing task displays and studies the result of an analysis, which exists in the model as analysis data sets. This task can generate displays of stress contours, deformed geometry, etc.
Assumptions for Finite Element Analysis of pressure vessel:
Analysis type taken is static structural while neglecting effect of loading and boundary condition with time. Only internal pressure is considered as load while neglecting all External loads.

CHAPTER-3

LITERATURE SURVEY

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, onur sayman, tolga dogan, necmettin tarakcioglu selcuk 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. Nikam Bachelor 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 is to 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. Levend Parnas, Nuran Katirci

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.

11. Shafique M.A. Khan Department of Mechanical Engineering, King Fahd University of Petroleum and Minerals.

Stress distributions in a horizontal pressure vessel.

This paper presents analysis results of stress distributions in a horizontal pressure vessel and the saddle supports. The results are obtained from a 3D finite element analysis. In addition to presenting the stress distribution in the pressure vessel, the results provide details of stress distribution in different parts of the saddle separately, i.e., wear, web, flange and base plates. The effect of changing the load and various geometric parameters is investigated and recommendations are made for the optimal values of ratio of the distance of support from the end of the vessel to the length of the vessel and ratio of the length of the vessel to the radius of the vessel for minimum stresses both in the pressure vessel and the saddle structure. Physical reasons for favouring of a particular value of ratio of the distance of support from the end of the vessel to the length of the vessel are also outlined.

12. Vinod Kumar, Navin Kumar, Surjit Angra, Prince Sharma

Design of Saddle Support for Horizontal Pressure Vessel.

This paper presents the design analysis of saddle support of a horizontal pressure vessel. Since saddle have the vital role to support the pressure vessel and to maintain its stability, it should be designed in such a way that it can afford the vessel load and internal pressure of the vessel due to liquid contained in the vessel. A model of horizontal pressure vessel and saddle support is created in ANSYS. Stresses are calculated using mathematical approach and ANSYS software. The analysis reveals the zone of high localized stress at the junction part of the pressure vessel and saddle support due to operating conditions. The results obtained by both the methods are compared with allowable stress value for safe designing.

13. M.R. Baum, Berkeley Centre, Berkeley

Failure of a horizontal pressure vessel containing a high temperature liquid: the velocity of end-cap and rocket missiles Magnox Electric plc.

Many process plant installations include cylindrical vessels which contain high temperature liquids with the remaining space above occupied by vapour or a vapour/gas mixture. If such a pressure vessel were to be ruptured, missiles (i.e., fragments) may be generated and equipment in the vicinity put at risk. There is a particular threat from large missiles. Theoretical models have been developed to describe the peak velocity achieved by end-caps and 'rocket' missiles generated by the circumferential failure of a vessel. The end-cap missile model assumes that the action of the escaping vapour/liquid on the end-cap is analogous to a missile driven by a gas jet from a constant pressure source.

14. K. Magnesia, P. Stasiewicza, W. Szyca

Institute of Applied Mechanics, Poznan University of Technology.

Flexible saddle support of a horizontal cylindrical pressure vessel.

The subject of this paper is the supporting saddle of a horizontal cylindrical pressure vessel filled with liquid. A parametric model of the saddle support has been developed; the effect of the geometrical parameters on the stress values arising in the structure has been examined by means of the Finite Element Method. The shape and location of the supporting saddle have been determined with a view to minimizing the concentration of stresses. Results of numerical analysis allow determination of the effective proportions of the geometrical parameters of the vessel.

15. Vijay Kumar, Pardeep Kumar

Mechanical design of pressure vessel by using PV-ELITE software.

The safety factor of a pressure vessel is related to both the tensile stress and yield strength for material allowance. ASME code section VIII has fully covered these two on the construction code for pressure vessel. This code section addressed mandatory and non-mandatory appendixes requirement, specific prohibition, vessel materials, design, fabrication, examination, inspection, testing, certification, and pressure relief. Mechanical design of a horizontal pressure vessel based on this standard had been done incorporating PV ELITE software. Analyses were carried out on head, shell, nozzle and saddle. The input parameters are type of material, pressure, temperature, diameter, and corrosion allowance. Analysis performed the calculations of internal and external pressure, weight of the element, allowable stresses, vessel longitudinal stress check, nozzle check and saddle check.

16. V. Mohanavel (Modelling and stress analysis of aluminium alloy-based composite pressure vessel through ANSYS software).

In Industries are extensively required for pressure vessels, which will have low weight to strength ratio without affecting the strength. In recent years, most of the sectors replace conventional materials with aluminium matrix composite materials. On the other hand, Aluminium matrix composite (AMCs) materials with their higher specific strength and these characteristics will reduce the structure's weight. In this research paper, the AA6082 alloy based aluminium matrix composites have been prepared by stir casting technique to test their mechanical properties under different weight percentages of reinforcement. Various mechanical studies have been done, such as tensile, impact, flexural, and hardness. For the same geometrical parameters of the steel pressure vessel, FE Analysis of AMCs composite pressure vessel is carried out, and stresses for different internal pressures are determined. And the design is carried out in design software solid works and analyse in ANSYS workbench. Then the results of steel pressure vessel and composite pressure vessel are compared for stress results.

17. A. M. Senthil Anbazhagan and M. Dev Anand

Department of Mechanical Engineering, Thuckalay, Kanyakumari District, Tamilnadu State, India.

Design and Crack Analysis of Pressure Vessel Saddles Using Finite Element Method.

The main intention of this work was to avoid the repeated failures of saddles during operation in energy development industries and wherever it is used. Methods/Statistical Analysis**:** Two different types of saddles were considered and fabricated using IS2060 Grade B material. The saddle parts were welded as per the code rule of API. Findings**:** Normally, welding in inclined saddle is difficult in comparison with straight saddle. This may be reason; the failure rate of inclined saddle is high in comparison with straight saddles during operation and loading conditions. The other possibility of failure is the gap formation inside the weld during joining the plates. This is due to non-deposition of weld materials. The gap would grow during operation and loaded conditions. To avoid these types of failures, external and internal crack inspections were done. Once the inspection was done, it was examined the load carrying of the fabricated saddles using FEM.

18. Goeun Han (Faculty of Purdue University)

A study on the failure analysis of the neutron embrittled reactor pressure vessel support using finite element analysis.

This study provides the failure assessment analysis of irradiated steel with prediction of the failure modes and safety margin. Through the failure assessment diagram, we could understand the effects of different levels of irradiation and loadings. Also, this study provides an alternative structural stress determination method, dividing the 3D solid element model into two 2D models, using the finite element analysis. Reconstructing the structural stress in 3D was carried by the 3x3 stress matrix and compared to the 3D FEA results. The difference in 2D FEA structural stress results were eliminated by the constructing the stress in 3D.

19. P. BOWEN,

EFFECTS OF MICROSTRUCTURE ON CLEAVAGE FRACTURE IN PRESSURE VESSEL STEEL.

This paper describes studies made on a wide range of microstructures in A533B pressure-vessel steel, to explore the relationships between microstructural parameters and toughness, as characterized by both the critical stress intensity factor, K, and the microscopic (local) cleavage fracture stress, large variations in toughness are obtained as a function of microstructure. The results show that auto-tempered martensite's possess toughness's superior to those for mixed lower-and-upper bainites, or for upper bainites. The carbide size distribution is found to be the most important single microstructural feature that controls cleavage fracture in these heat-treated conditions. The coarsest carbides in the distribution are the most deleterious to toughness.

20. N. Karthik, M. Jaypal Reddy, M. NagaKiran

Design Optimization and Buckling Analysis of Pressure Vessel.

The modelling was done by using Creo 2.0. Which is an advanced modelling software used in almost all the manufacturing industries. After the modelling the model was imported into the ANSYS 14.0. The linear buckling analysis of the pressure vessel will be done before and after the winding coil was placed at different load conditions and the maximum buckling load will be estimated. By considering some different type of materials to the model for showing the differences. Finally, the results will be tabulated and graphs will be plotted.

CHAPTER-4

DESIGN CONSIDERATIONS AND FORMULAS

4.1. ASME BOILER AND PRESSURE VESSEL CODE (BPVC) FORMULAS ARE:

Cylindrical shells:

$\sigma\theta = p(r+0.6t)/tE$

$\sigma l = p(r-0.4t)/2tE$

where E is the joint efficient, and all others variables as stated above.

The factor of safety is often included in these formulas as well, in the case of the ASME BPVC this term is included in the material stress value when solving for pressure or thickness.

4.2. MAXIMUM ALLOWABLE WORKING PRESSURE

When the thickness of the shell does not exceed one half of the inside radius, the maximum allowable working pressure on the cylindrical shell of a steam boiler, pressure vessel or drum shall be determined by the strength of the weakest course computed from the thickness of the plate, the efficiency of the longitudinal joint, or of the ligament between openings (whichever is the least), the inside radius of the course, and the maximum allowable unit working stress.

$P = (S\ E\ t)/\ (R + 0.6t)$ or

$t = PR/\ (SE - 0.6P)$

Where,

P = maximum allowable working pressure, pounds per square inch,

S = maximum allowable unit working stress, pounds per square inch, from, A.S.M.E. except for shells or headers of seamless or fusion welded construction exceeding 1/2 inch in thickness, which shall be built under the provisions of A.S.M.E.,

E = efficiency of longitudinal joints or of ligaments between openings: for rivetted joints calculated riveted efficiency; for fusion welded joints efficiency specified in A.S.M.E.; for seamless shells 100 percent (unity); for ligaments between openings, the efficiency shall be calculated by the rules given in A.S.M.E.,

t = minimum thickness of shell plates in weakest course, inches,

R = inside radius of the weakest course of the shell or drum, inches.

The maximum allowable working pressure for shells other than cylindrical, and for heads and other parts, shall be determined in a similar manner using the formulas appropriate for the parts, as otherwise given in the A.S.M.E. Code or some other acceptable formula.

4.3. ASSUMPTIONS & BOUNDARY CONDITIONS: -

Here the vessel has the following design characteristics:

Inside diameter of shell – 406.4mm

Length – 1.5494 m

Shell material – ALLOY STEEL

(Yield strength – 620422kpa)

Fluid inside pressure vessel – Steam

Working pressure – 1200kpa

Insulation material – Glass wool

Weld condition – Fillet weld double sided

For shell area we used Alloy Steel to weld the joints.

For other parts made of mild steel, we used mild steel as filler material to weld the joints.

For better weld we have used CO2 – MIG welding in place of conventional arc welding in order to prevent from weak porous weld section of arc welding.

Type – Horizontal circular pressure vessel with flat ends having door mechanisms on both ends.

Thin shell ($t/d = 0.005 < 0.05$)

Shell sheet thickness –

From Table P7, A.S.M.E.

Yield strength = 620422 kpa

Allowable stress(S) = 78534.4 kpa

(Longitudinal butt-welded joint efficiency factor for non-radiographed weld) E = 0.7

(Working pressure) P = 1200 kpa

From ASME Section VIII, Division 1, paragraph UG-27,

(Minimum design wall thickness of shell plates) $t =(PR)/ (SE - 0.6P)$

$t= (1200X203.4)/ (78534.4X0.7- 0.6X1200)$

$= 4.49$ mm

Corrosive allowance = 3 mm

Taking design shell thickness = 8 mm > Minimum design wall thickness of shell plates

To determine the pressure limit of vessel for the chosen design shell thickness,

(Maximum allowable working pressure or design pressure)

$P = (2SEt) / (R - 0.4t)$

$= (2 X 78534.4 X 0.7 X 8) / (203.2 – 0.4X8)$

$= 879585.28 / 200$

$= 4397.92$ kpa > (working pressure)

```python
'''
class material:
    def __init__(self,mname,mstrength,fos):
        self.mname=mname
        self.mstrength=mstrength
        self.fos=fos

class design:
    def __init__(self,mlist):
        self.mlist=mlist

    def minimumthickness(self,dp):
        ans=[]
        r=203.2
        for i in self.mlist:
            print(i.mname.upper()+":")
            s=i.mstrength/i.fos
            mt=((dp*r)/((s*0.7)-(0.6*dp)))
            print("The Minimum Thickness is:",mt)
            if (mt<=8.00):
                print("Design is safe")
                ans.append((i.mname,mt))
            else:
                print("Needed Modification")
        return ans
```

```python
if __name__=="__main__":
    n=int(input())
    mlist=[]
    for i in range(n):
        mname=input()
        mstrength=float(input())
        fos=float(input())
        mlist.append(material(mname,mstrength,fos))
    dp=float(input("enter the design pressure(kpa):"))
    pr=design(mlist)
    ans1=pr.minimumthickness(dp)
    for i in ans1:
        print(*i)
```

MATERIAL	MINIMUM THICKNESS (mm)
Stainless Steel	4.494
Cast Carbon Steel	5.125
Aluminum Alloy	5.129
Copper Alloy	5.139
Titanium Alloy	5.124

CHAPTER-5

ANALYTICAL CALCULATIONS

5.1 MODEL CALCULATIONS

Diameter = 406.4 mm

Thickness = 8 mm

Operating pressure= 102 Mpa

Material properties of alloy steel

young's modulus E = 2×10^{11} pa

Poisson's ratio: 0.28

Hoop strain: (€)

$$\frac{P * d * (2 - \mu)}{4 * t * E}$$

$$\frac{1.2 * 406.4 * (2 - 0.28)}{4 * 8 * 2 * 10^{\wedge}11}$$

$$= 1.31 \times 10^{-4}$$

Hoop Stress:

- € x E = σ_H
- $\sigma_H = 2 \times 10^{11} \times 1.3 \times 10^{-4}$

  $= 26.21$ Mpa

Longitudinal strain:

$$\frac{P * d * (1 - 2\mu)}{4tE}$$

$$\frac{1.2 * 406.4 * (1 - 2(0.28))}{4 * 8 * 2 * 10^{\wedge}11}$$

$$= 3.35 \times 10^{-5}$$

Longitudinal stress:

- σ_H = € x E
- $3.35 \times 10^{-5} \times 2 \times 10^{11}$

= 6.70 Mpa

5.2 PYTHON PROGRAM FOR FINDING STRESSES AND STRAINS

```
main.py
def project(thicknesslist):
    d=406.40
    mplist={"Alloy steel":[2*10**11,0.28],"Titanium alloys":[1.05*10**11,0.33],"carbon steel":[2*10**11,0.32],
    "Alluminium alloy":[6.9*10**11,0.33],"beryllium copper alloy":[1.25*10**11,0.3]}
    p=1.20
    for i in thicknesslist:
        for k in mplist:
            if ((i/d)<20):
                print("Thin cylinder:",k.upper())
                print("Thickness",i,":")
                hst=((p*d*(2-mplist[k][1]))/((4*i)*mplist[k][0]))
                hs=hst*mplist[k][0]
                lst=((p*d*(1-2*mplist[k][1]))/((4*i)*mplist[k][0]))
                ls=lst*mplist[k][0]
                print("Hoop strain=",hst,)
                print("Hoop stress=",hs,"Mpa")
                print("Longitudinal strain=",lst)
                print("Longitudinal stress=",ls,"Mpa")
                print("\n")
            else:
                print("Thick Cylinder,\n")


if __name__=="__main__":
    thicknesslist=[8,10,15,20,25,30,35,40]
    project(thicknesslist)
```

<u>THICKNESS 8MM</u>

- Thin cylinder: ALLOY STEEL

  Thickness: 8
  Hoop strain= 1.31064e-4
  Hoop stress= 26.212799999999998 Mpa
  Longitudinal strain= 3.3527999999999995e-5
  Longitudinal stress= 6.705599999999999 Mpa

- Thin cylinder: TITANIUM ALLOYS

  Thickness: 8
  Hoop strain= 2.423885714285714e-4
  Hoop stress= 25.450799999999994 Mpa
  Longitudinal strain= 4.934857142857142e-5
  Longitudinal stress= 5.181599999999999 Mpa

- Thin cylinder: CARBON STEEL

  Thickness: 8
  Hoop strain= 1.2801599999999998e-4
  Hoop stress= 25.603199999999998 Mpa
  Longitudinal strain= 2.7431999999999994e-5
  Longitudinal stress= 5.486399999999999 Mpa

- Thin cylinder: ALLUMINIUM ALLOY

  Thickness: 8
  Hoop strain= 3.688521739130434e-5
  Hoop stress= 25.450799999999994 Mpa
  Longitudinal strain= 7.509565217391302e-6
  Longitudinal stress= 5.181599999999999 Mpa

- Thin cylinder: BERYLLIUM COPPER ALLOY

  Thickness: 8
  Hoop strain= 2.0726399999999997e-4
  Hoop stress= 25.907999999999998 Mpa
  Longitudinal strain= 4.8768e-5
  Longitudinal stress= 6.096 Mpa

THICKNESS 10mm

- Thin cylinder: ALLOY STEEL

  Thickness:10
  Hoop strain= 1.0485119999999999e-4
  Hoop stress= 20.970239999999997 Mpa
  Longitudinal strain= 2.6822399999999994e-5
  Longitudinal stress= 5.364479999999999 Mpa

- Thin cylinder: TITANIUM ALLOYS

  Thickness: 10
  Hoop strain= 1.9391085714285712e-4 Hoop
  stress= 20.360639999999997 Mpa
  Longitudinal strain= 3.947885714285713e-5
  Longitudinal stress= 4.145279999999999 Mpa

- Thin cylinder: CARBON STEEL

  Thickness: 10
  Hoop strain= 1.0241279999999999e-4
  Hoop stress= 20.482559999999996 Mpa
  Longitudinal strain= 2.1945599999999996e-5
  Longitudinal stress= 4.389119999999999 Mpa

- Thin cylinder: ALLUMINIUM ALLOY

  Thickness: 10
  Hoop strain= 2.9508173913043474e-5 Hoop
  stress= 20.360639999999997 Mpa
  Longitudinal strain= 6.007652173913042e-6
  Longitudinal stress= 4.145279999999999 Mpa

- Thin cylinder: BERYLLIUM COPPER ALLOY

  Thickness: 10
  Hoop strain= 1.658112e-4
  Hoop stress= 20.726399999999998 Mpa
  Longitudinal strain= 3.90144e-5
  Longitudinal stress= 4.8768 Mpa

THICKNESS 15mm

- Thin cylinder: ALLOY STEEL

  Thickness: 15
  Hoop strain= 6.99008e-5
  Hoop stress= 13.98016 Mpa
  Longitudinal strain= 1.7881599999999997e-5
  Longitudinal stress= 3.5763199999999995 Mpa

- Thin cylinder: TITANIUM ALLOYS

  Thickness: 15
  Hoop strain= 1.2927390476190475e-4
  Hoop stress= 13.57376 Mpa
  Longitudinal strain= 2.6319238095238088e-5
  Longitudinal stress= 2.7635199999999993 Mpa

- Thin cylinder: CARBON STEEL

  Thickness: 15
  Hoop strain= 6.827519999999999e-5
  Hoop stress= 13.655039999999998 Mpa
  Longitudinal strain= 1.4630399999999997e-5
  Longitudinal stress= 2.9260799999999993 Mpa

- Thin cylinder: ALLUMINIUM ALLOY

  Thickness: 15
  Hoop strain= 1.9672115942028984e-5 Hoop
  stress= 13.573759999999998 Mpa
  Longitudinal strain= 4.005101449275361e-6
  Longitudinal stress= 2.7635199999999993 Mpa

- Thin cylinder: BERYLLIUM COPPER ALLOY

  Thickness: 15
  Hoop strain= 1.1054079999999998e-4
  Hoop stress= 13.817599999999999 Mpa
  Longitudinal strain= 2.60096e-5
  Longitudinal stress= 3.2512000000000003 Mpa

THICKNESS 20mm

- Thin cylinder: ALLOY STEEL

  Thickness: 20
  Hoop strain= 5.2425599999999993e-5
  Hoop stress= 10.485119999999998 Mpa
  Longitudinal strain= 1.3411199999999997e-5
  Longitudinal stress= 2.6822399999999993 Mpa

- Thin cylinder: TITANIUM ALLOYS

  Thickness: 20
  Hoop strain= 9.695542857142856e-5
  Hoop stress= 10.180319999999998 Mpa
  Longitudinal strain= 1.9739428571428565e-5
  Longitudinal stress= 2.0726399999999994 Mpa

- Thin cylinder: CARBON STEEL

  Thickness: 20
  Hoop strain= 5.1206399999999999e-5
  Hoop stress= 10.241279999999998 Mpa
  Longitudinal strain= 1.0972799999999998e-5
  Longitudinal stress= 2.1945599999999996 Mpa

- Thin cylinder: ALLUMINIUM ALLOY

  Thickness: 20
  Hoop strain= 1.4754086956521737e-5 Hoop
  stress= 10.180319999999998 Mpa
  Longitudinal strain= 3.003826086956521e-6
  Longitudinal stress= 2.0726399999999994 Mpa

- Thin cylinder: BERYLLIUM COPPER ALLOY

  Thickness: 20
  Hoop strain= 8.29056e-5
  Hoop stress= 10.363199999999999 Mpa
  Longitudinal strain= 1.95072e-5
  Longitudinal stress= 2.4384 Mpa

THICKNESS 25mm

- Thin cylinder: ALLOY STEEL

  Thickness: 25
  Hoop strain= 4.194048e-5
  Hoop stress= 8.388095999999999 Mpa
  Longitudinal strain= 1.0728959999999997e-5
  Longitudinal stress= 2.1457919999999993 Mpa

- Thin cylinder: TITANIUM ALLOYS

  Thickness: 25
  Hoop strain= 7.756434285714285e-5
  Hoop stress= 8.144256 Mpa
  Longitudinal strain= 1.5791542857142852e-5
  Longitudinal stress= 1.6581119999999994 Mpa

- Thin cylinder: CARBON STEEL

  Thickness: 25
  Hoop strain= 4.0965119999999996e-5
  Hoop stress= 8.193024 Mpa
  Longitudinal strain= 8.778239999999998e-6
  Longitudinal stress= 1.7556479999999997 Mpa

- Thin cylinder: ALLUMINIUM ALLOY

  Thickness: 25
  Hoop strain= 1.180326956521739e-5
  Hoop stress= 8.144256 Mpa
  Longitudinal strain= 2.4030608695652167e-6
  Longitudinal stress= 1.6581119999999996 Mpa

- Thin cylinder: BERYLLIUM COPPER ALLOY

  Thickness: 25
  Hoop strain= 6.632448e-5
  Hoop stress= 8.29056 Mpa
  Longitudinal strain= 1.560576e-5
  Longitudinal stress= 1.95072 Mpa

THICKNESS 30mm

- Thin cylinder: ALLOY STEEL

  Thickness: 30
  Hoop strain= 3.49504e-5
  Hoop stress= 6.99008 Mpa
  Longitudinal strain= 8.940799999999998e-6
  Longitudinal stress= 1.7881599999999997 Mpa

- Thin cylinder: TITANIUM ALLOYS

  Thickness: 30
  Hoop strain= 6.463695238095238e-5
  Hoop stress= 6.78688 Mpa
  Longitudinal strain= 1.3159619047619044e-5
  Longitudinal stress= 1.3817599999999997 Mpa

- Thin cylinder: CARBON STEEL

  Thickness: 30
  Hoop strain= 3.4137599999999995e-5 Hoop
  stress= 6.827519999999999 Mpa
  Longitudinal strain= 7.315199999999999e-6
  Longitudinal stress= 1.4630399999999997 Mpa

- Thin cylinder: ALLUMINIUM ALLOY

  Thickness: 30
  Hoop strain= 9.836057971014492e-6
  Hoop stress= 6.786879999999999 Mpa
  Longitudinal strain= 2.0025507246376806e-6
  Longitudinal stress= 1.3817599999999997 Mpa

- Thin cylinder: BERYLLIUM COPPER ALLOY

  Thickness: 30
  Hoop strain= 5.527039999999999e-5
  Hoop stress= 6.908799999999999 Mpa
  Longitudinal strain= 1.30048e-5
  Longitudinal stress= 1.6256000000000002 Mpa

THICKNESS 35mm

- Thin cylinder: ALLOY STEEL

  Thickness: 35
  Hoop strain= 2.995748571428571e-5
  Hoop stress= 5.991497142857142 Mpa
  Longitudinal strain= 7.663542857142856e-6
  Longitudinal stress= 1.532708571428571 Mpa

- Thin cylinder: TITANIUM ALLOYS

  Thickness: 35
  Hoop strain= 5.540310204081632e-5
  Hoop stress= 5.8173257142857135 Mpa
  Longitudinal strain= 1.1279673469387753e-5
  Longitudinal stress= 1.184365714285714 Mpa

- Thin cylinder: CARBON STEEL

  Thickness: 35
  Hoop strain= 2.9260799999999994e-5
  Hoop stress= 5.852159999999999 Mpa
  Longitudinal strain= 6.2701714285714276e-6
  Longitudinal stress= 1.2540342857142854 Mpa

- Thin cylinder: ALLUMINIUM ALLOY

  Thickness: 35
  Hoop strain= 8.430906832298136e-6
  Hoop stress= 5.817325714285714 Mpa
  Longitudinal strain= 1.7164720496894406e-6
  Longitudinal stress= 1.184365714285714 Mpa

- Thin cylinder: BERYLLIUM COPPER ALLOY

  Thickness: 35
  Hoop strain= 4.7374628571428565e-5
  Hoop stress= 5.921828571428571 Mpa
  Longitudinal strain= 1.1146971428571429e-5
  Longitudinal stress= 1.3933714285714287 Mpa

THICKNESS 40mm

- Thin cylinder: ALLOY STEEL

  Thickness: 40
  Hoop strain= 2.6212799999999997e-5
  Hoop stress= 5.242559999999999 Mpa
  Longitudinal strain= 6.7055999999999984e-6
  Longitudinal stress= 1.3411199999999996 Mpa

- Thin cylinder: TITANIUM ALLOYS

  Thickness: 40
  Hoop strain= 4.847771428571428e-5
  Hoop stress= 5.090159999999999 Mpa
  Longitudinal strain= 9.869714285714283e-6
  Longitudinal stress= 1.0363199999999997 Mpa

- Thin cylinder: CARBON STEEL

  Thickness: 40
  Hoop strain= 2.5603199999999997e-5 Hoop
  stress= 5.120639999999999 Mpa
  Longitudinal strain= 5.486399999999999e-6
  Longitudinal stress= 1.0972799999999998 Mpa

- Thin cylinder: ALLUMINIUM ALLOY

  Thickness: 40
  Hoop strain= 7.377043478260869e-6
  Hoop stress= 5.090159999999999 Mpa
  Longitudinal strain= 1.5019130434782604e-6
  Longitudinal stress= 1.0363199999999997 Mpa

- Thin cylinder: BERYLLIUM COPPER ALLOY

  Thickness: 40
  Hoop strain= 4.14528e-5
  Hoop stress= 5.1815999999999995 Mpa
  Longitudinal strain= 9.7536e-6
  Longitudinal stress= 1.2192 Mpa

CHAPTER-6

SOLID WORKS ANALYSIS

Thin Cylindrical Horizontal Pressure Vessel subjected to Internal:

we have taken 5 different materials in construction of the pressure vessel and performed stress analysis using SolidWorks Software.

The selected 5 different materials are:

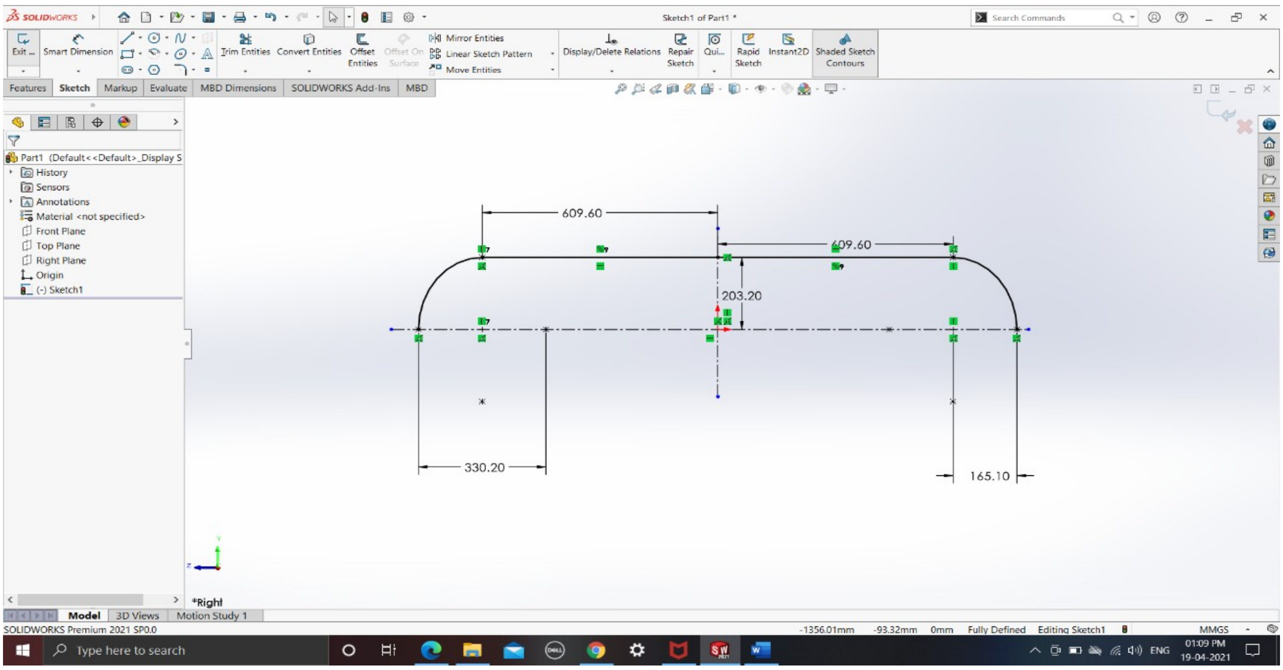
1. Alloy steel
2. Cast carbon steel
3. Aluminum alloy
4. Copper alloy
5. Titanium alloy

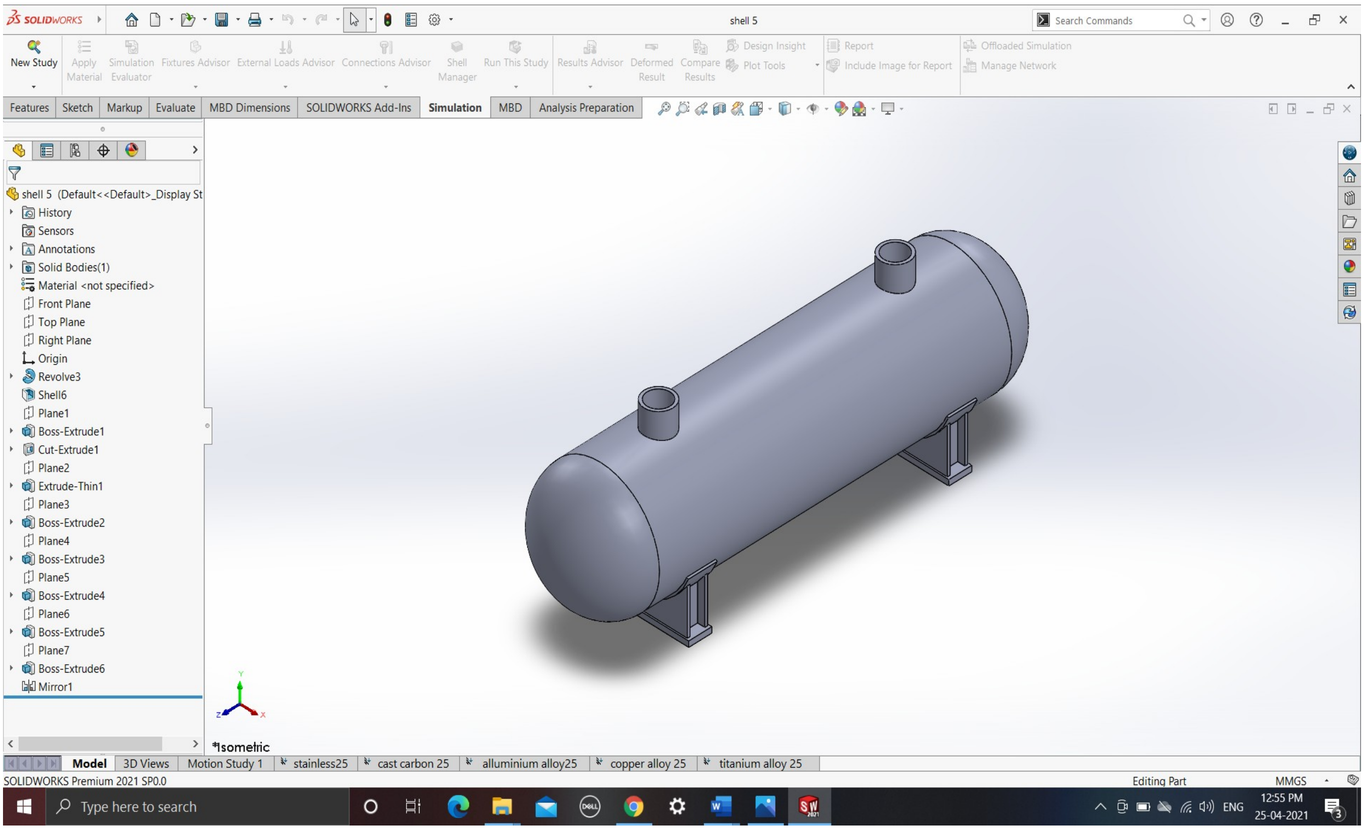
THE DIMENSIONS AND INTERNAL PRESSURE OF PRESSURE VESSEL:

Pressure = 1.2 MPa | Length of Pressure Vessel = 1.5494 m

Thickness of pressure Vessel = 8 to 40 mm

Outer Radius = 406.4mm

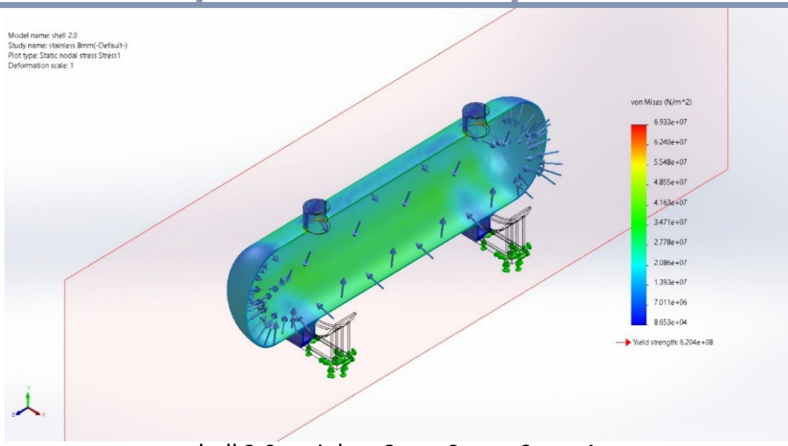




➢ ALLOY STEEL:

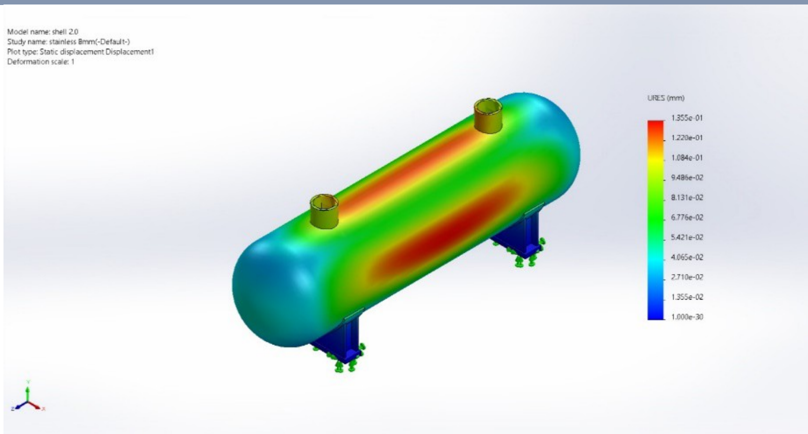
PRESSURE VESSEL OF THICKNESS 8MM:

STRESS ANALYSIS:

Name	Type	Min	Max
Stress1	VON: von Mises Stress	8.653e+04N/m^2 Node: 3216	6.933e+07N/m^2 Node: 10253
 shell 2.0-stainless 8mm-Stress-Stress1			

DISPLACEMENT

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2676	1.355e-01mm Node: 2512

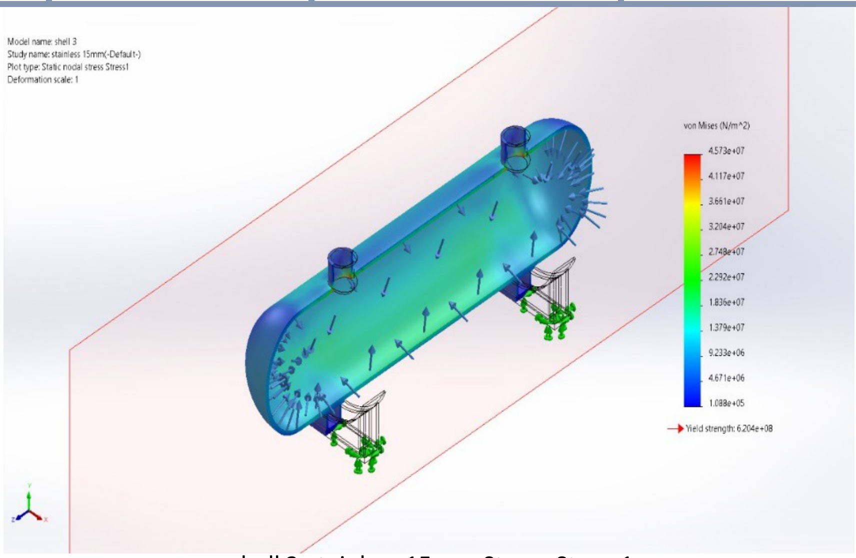


shell 2.0-stainless 8mm-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 15mm:

STRESS ANALYSIS:

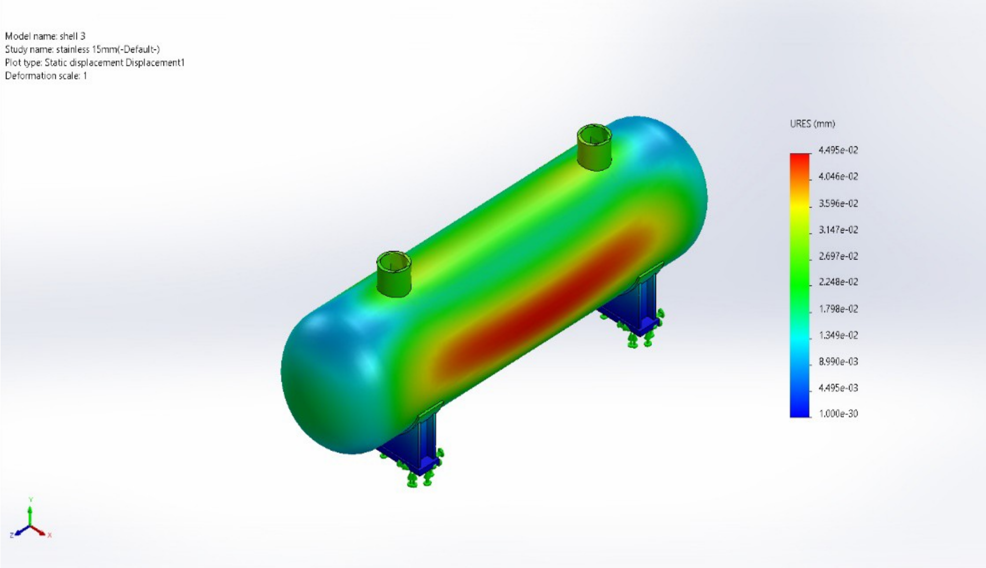
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.088e+05N/m^2 Node: 3140	4.573e+07N/m^2 Node: 9888



shell 3-stainless 15mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2587	4.495e-02mm Node: 475

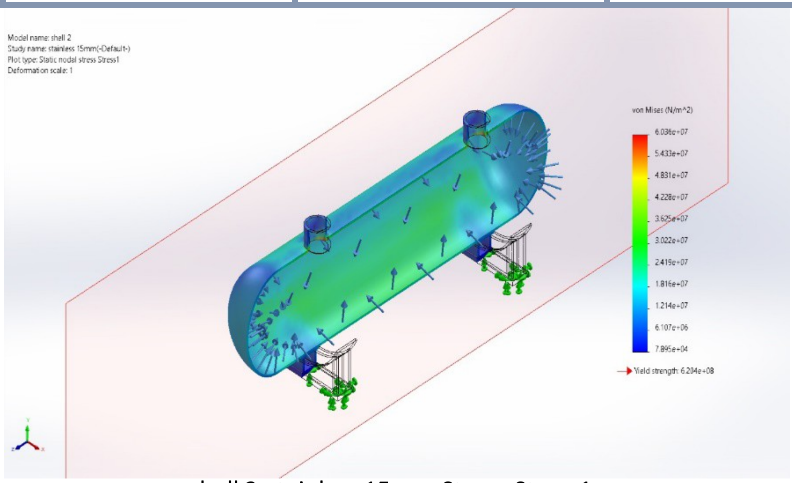


shell 3-stainless 15mm-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 10mm:

STRESS ANALYSIS:

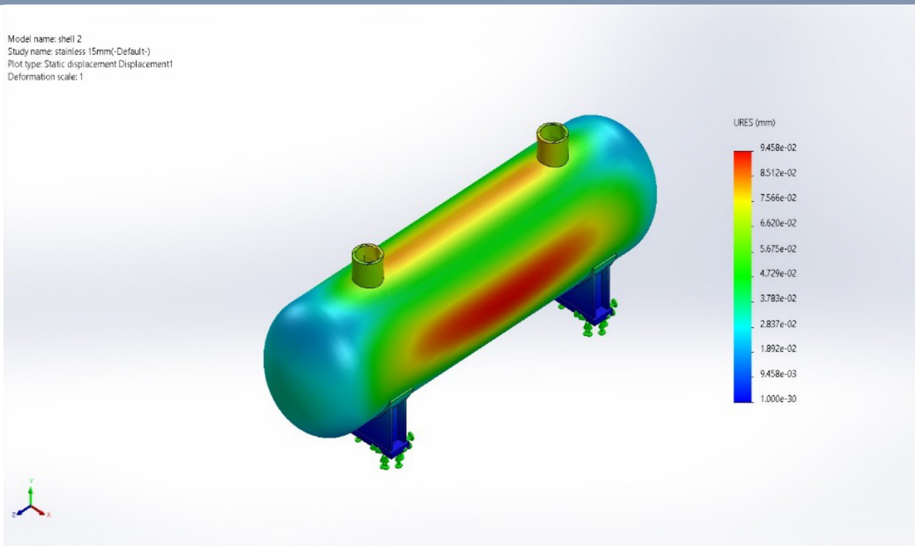
Name	Type	Min	Max
Stress1	VON: von Mises Stress	7.895e+04N/m^2 Node: 3421	6.036e+07N/m^2 Node: 10255



shell 2-stainless 15mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2679	9.458e-02mm Node: 12135

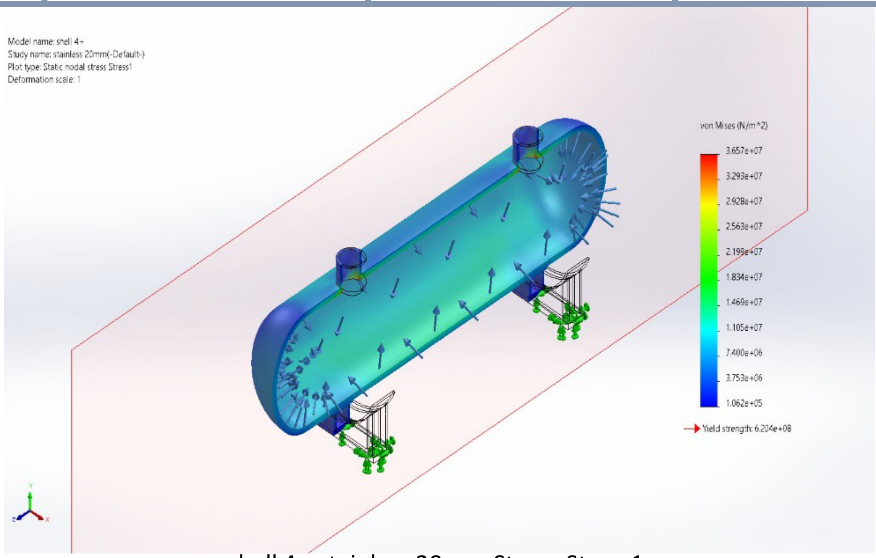


shell 2-stainless 15mm-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 20mm:

STRESS ANALYSIS:

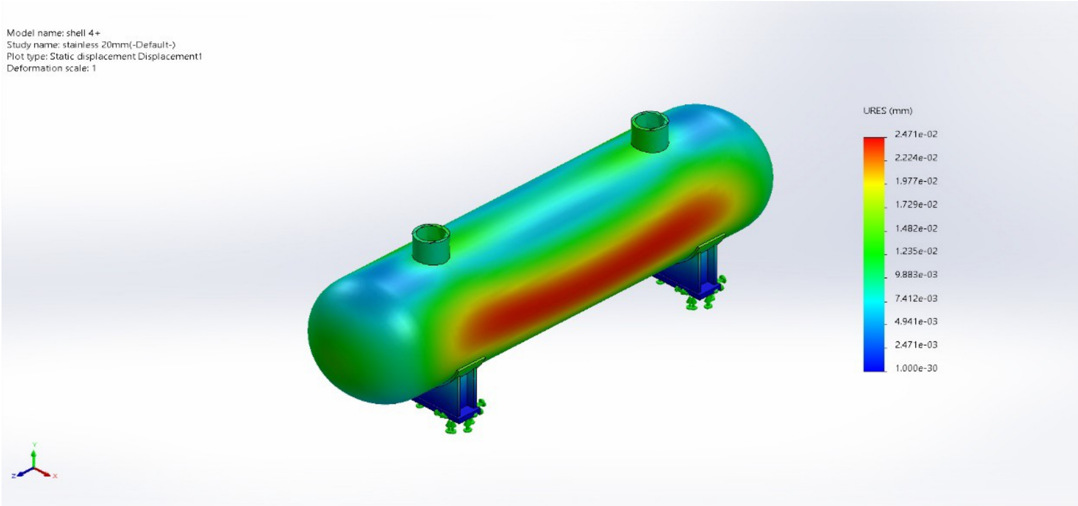
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.062e+05N/m^2 Node: 3105	3.657e+07N/m^2 Node: 9786



shell 4+-stainless 20mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2566	2.471e-02mm Node: 13745

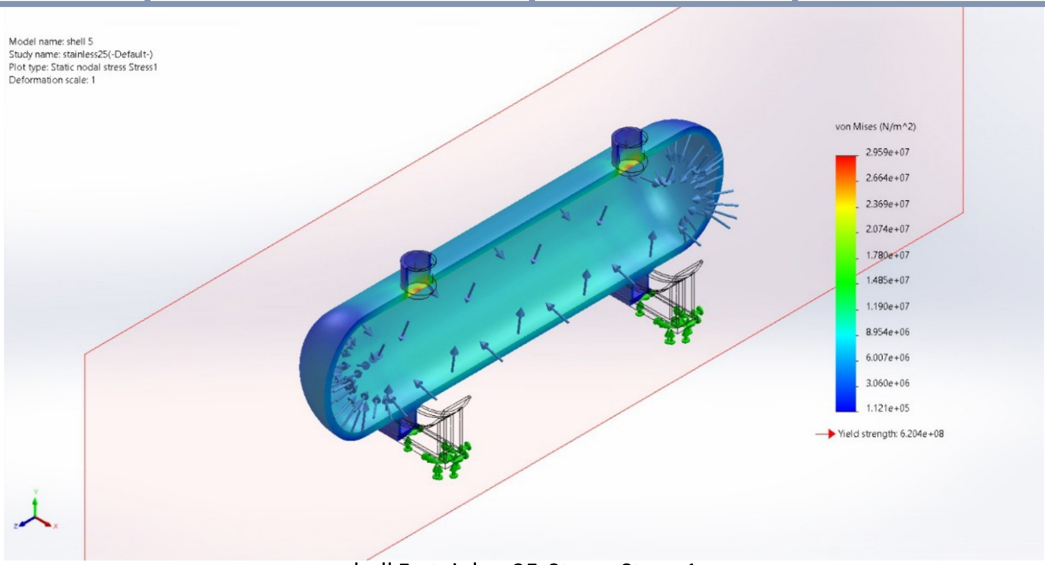


shell 4+-stainless 20mm-Displacement-Displacement1

PRESSUER VESSEL OF THICKNESS 25mm:

STRESS ANALYSIS:

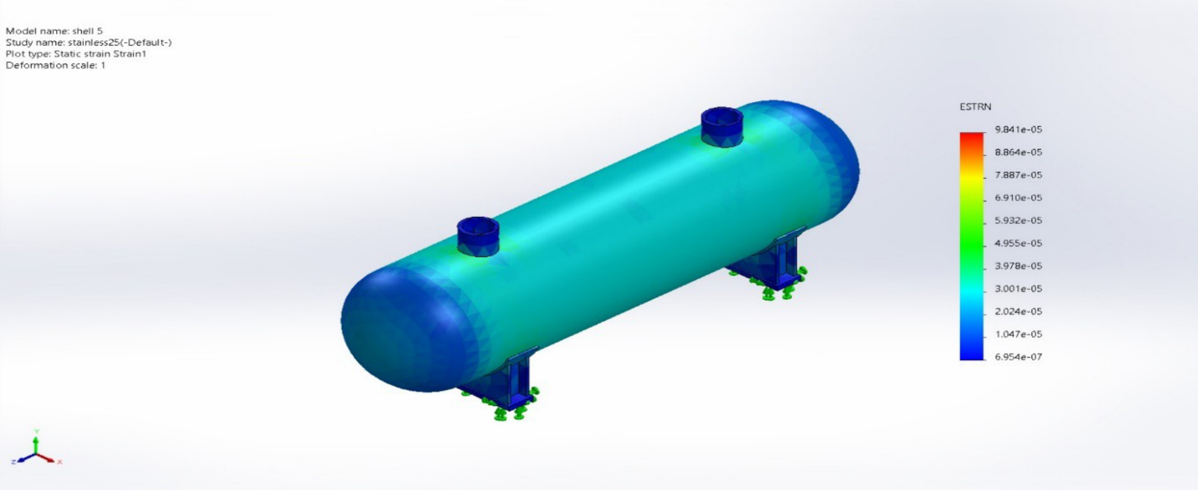
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.121e+05N/m^2 Node: 6376	2.959e+07N/m^2 Node: 1532



shell 5-stainless25-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	6.954e-07 Element: 6183	9.841e-05 Element: 3478

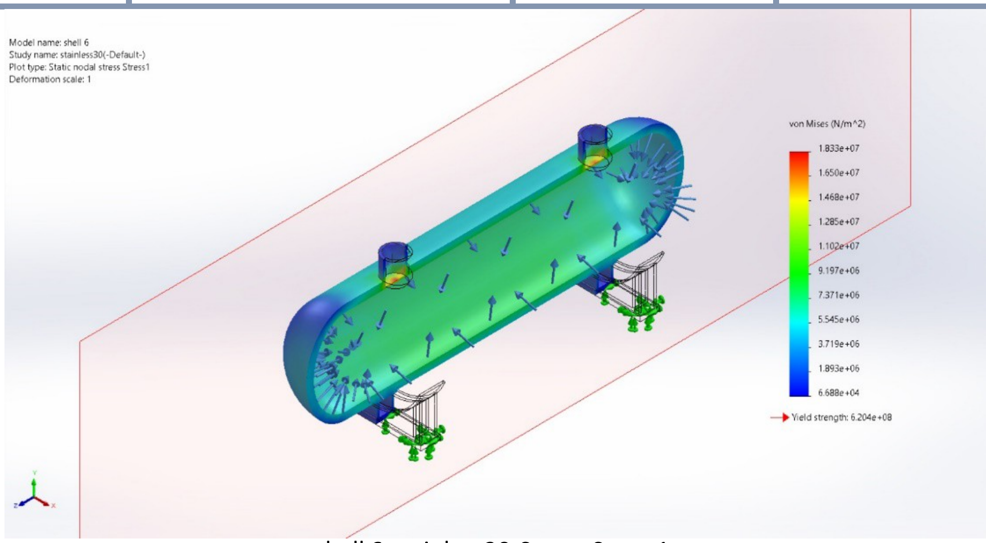


shell 5-stainless25-Strain-Strain1

PRESSURE VESSEL OF THICKNESS 30mm:

STRESS ANALYSIS:

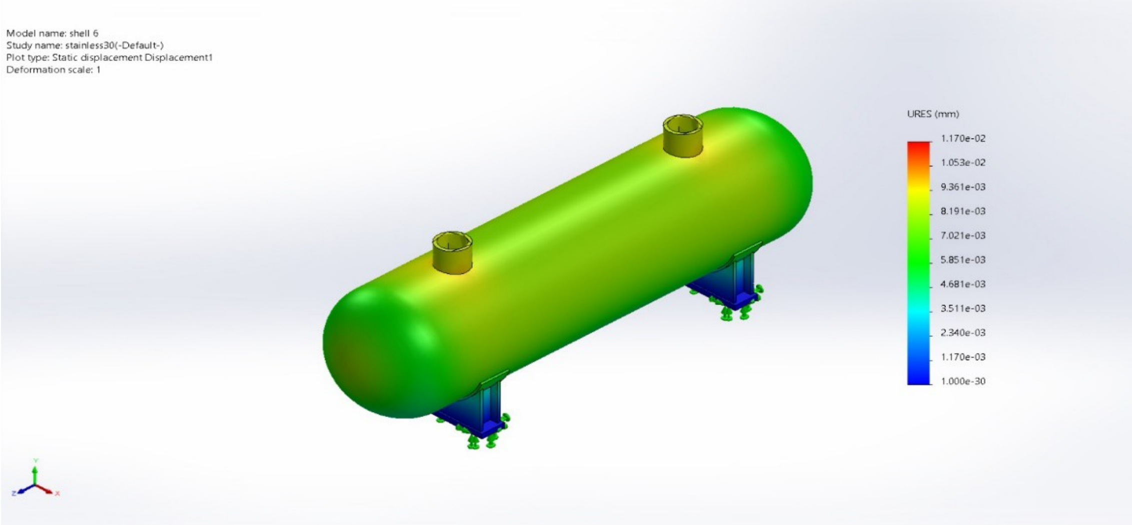
Name	Type	Min	Max
Stress1	VON: von Mises Stress	6.688e+04N/m^2 Node: 995	1.833e+07N/m^2 Node: 1532



shell 6-stainless30-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.170e-02mm Node: 6789

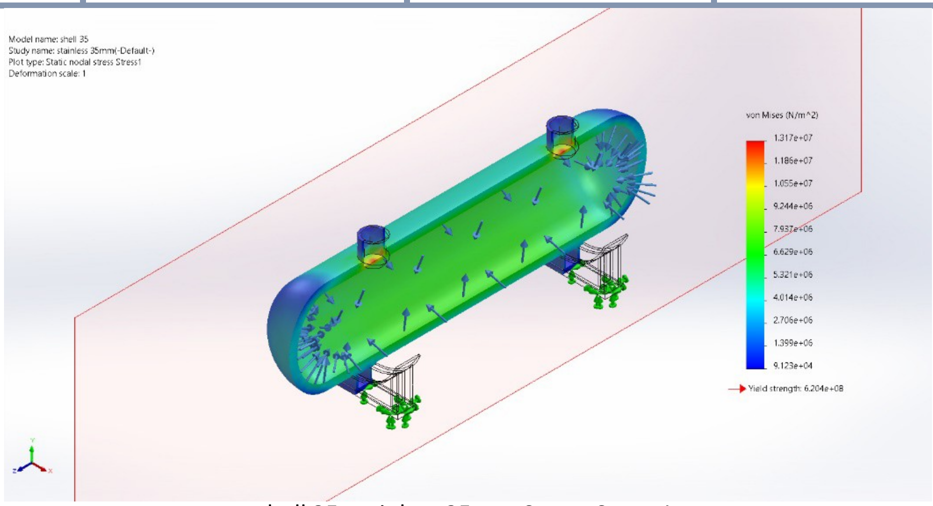


shell 6-stainless30-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 35mm:

STRESS ANALYSIS:

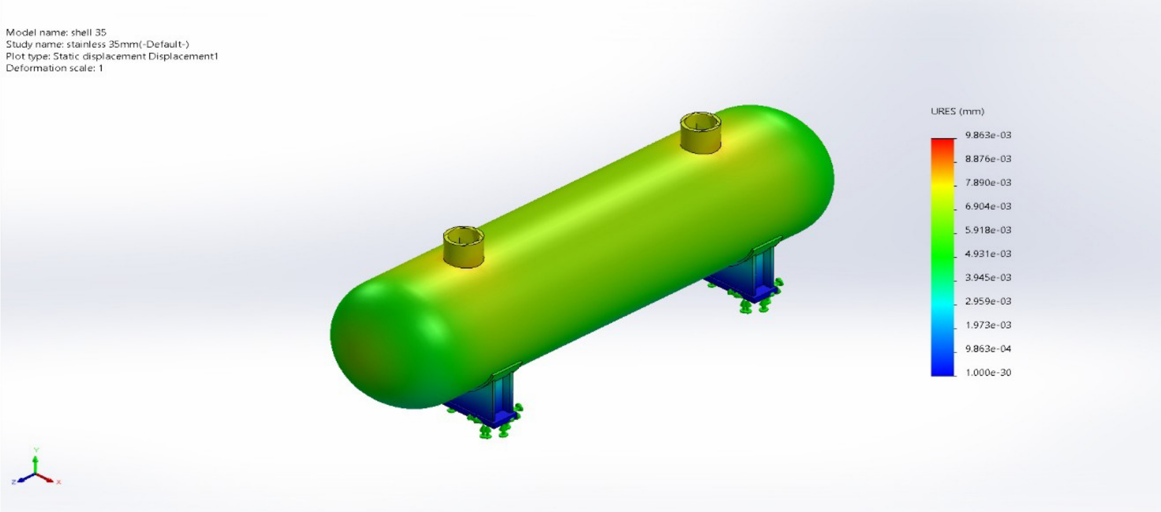
Name	Type	Min	Max
Stress1	VON: von Mises Stress	9.123e+04N/m^2 Node: 6281	1.317e+07N/m^2 Node: 1527



shell 35-stainless 35mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	9.863e-03mm Node: 2441

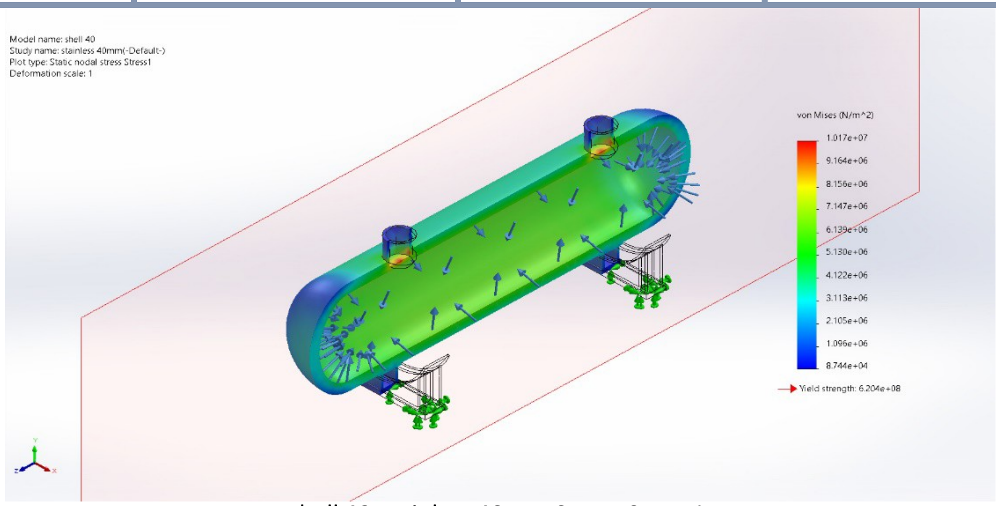


shell 35-stainless 35mm-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 40mm:

STRESS ANALYSIS:

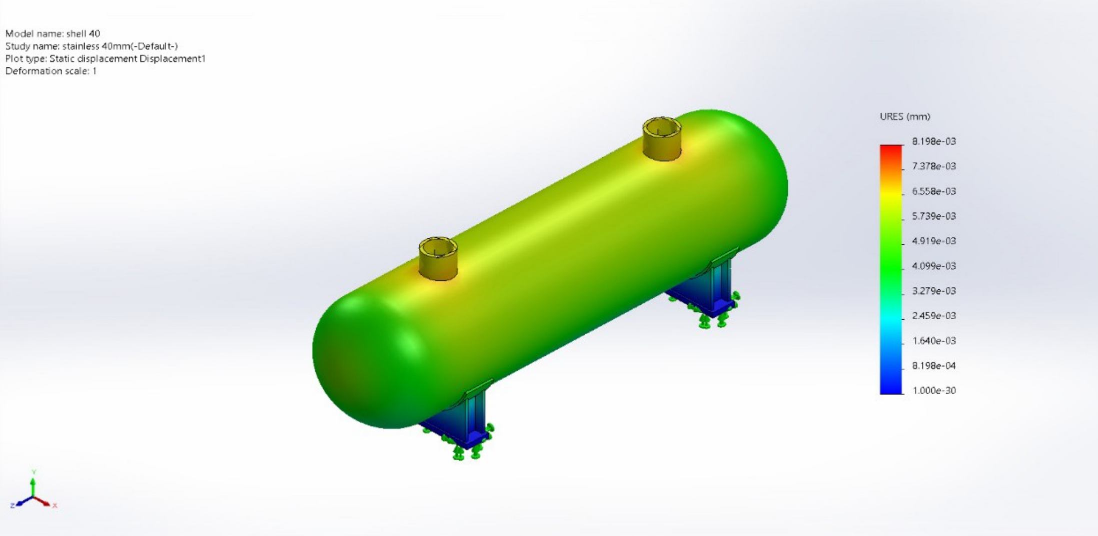
Name	Type	Min	Max
Stress1	VON: von Mises Stress	8.744e+04N/m^2 Node: 8059	1.017e+07N/m^2 Node: 1532



shell 40-stainless 40mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	8.198e-03mm Node: 2429



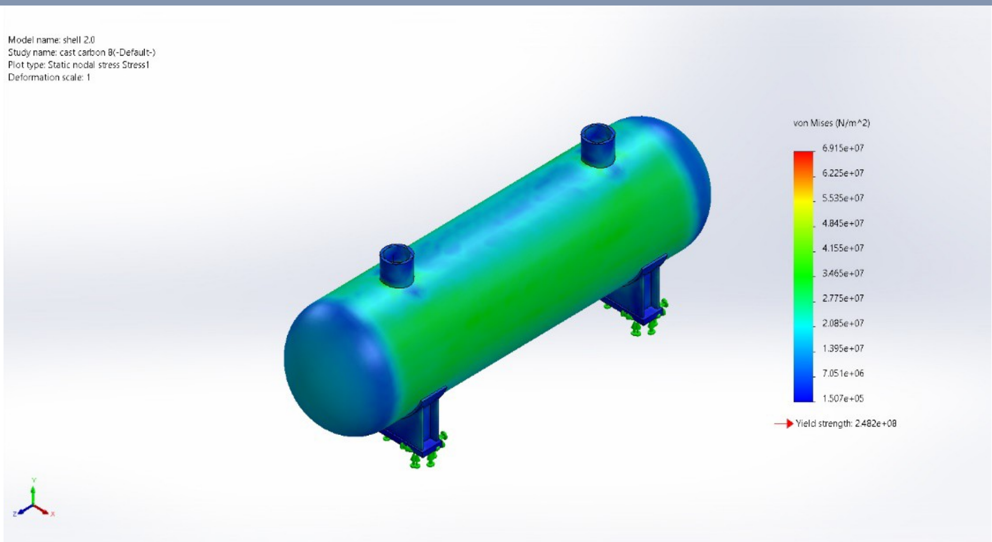
shell 40-stainless 40mm-Displacement-Displacement1

➢ **CAST CARBON STEEL:**

PRESSURE VESSEL OF THICKNESS 8mm:

STRESS ANALYSIS:

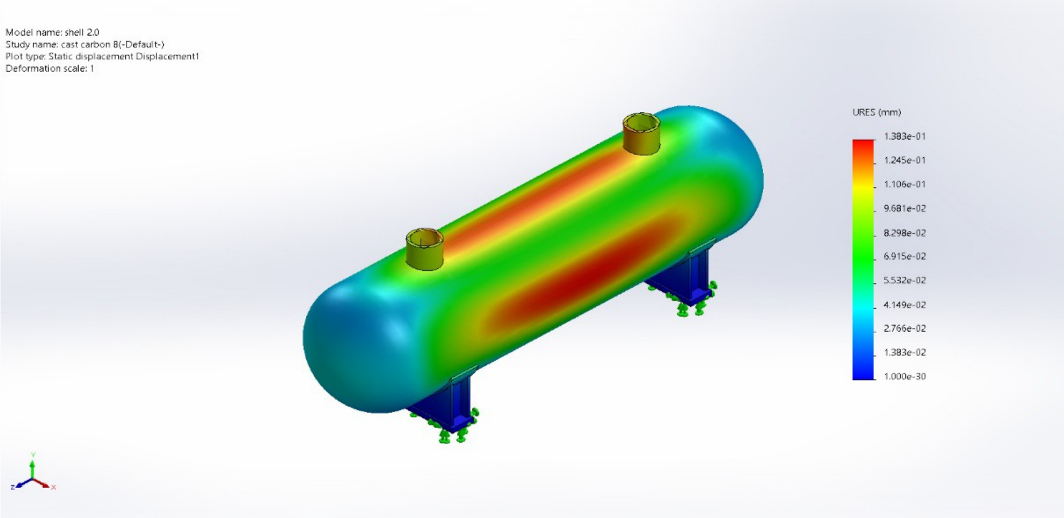
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.507e+05N/m^2 Node: 3590	6.915e+07N/m^2 Node: 10253



shell 2.0-cast carbon 8-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2676	1.383e-01mm Node: 2512

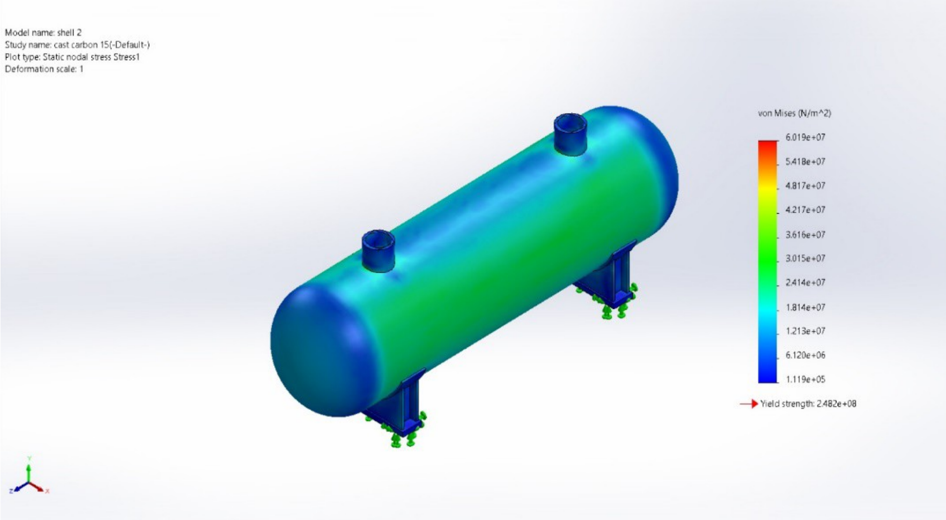


shell 2.0-cast carbon 8-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 10mm:

STRESS ANALSIS:

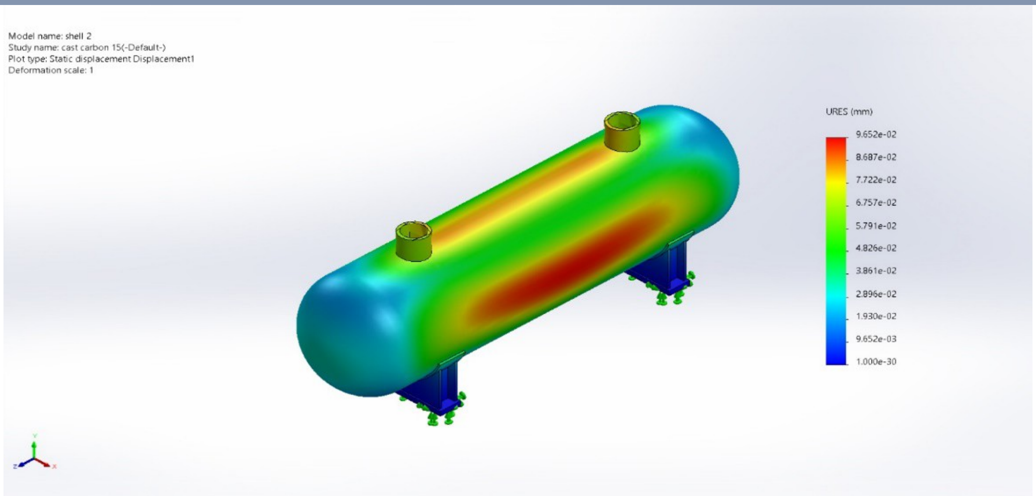
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.119e+05N/m^2 Node: 3489	6.019e+07N/m^2 Node: 10255



shell 2-cast carbon 15-Stress-Stress1

DISPLACEMENT:

Name	**Type**	**Min**	**Max**
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2679	9.652e-02mm Node: 12135

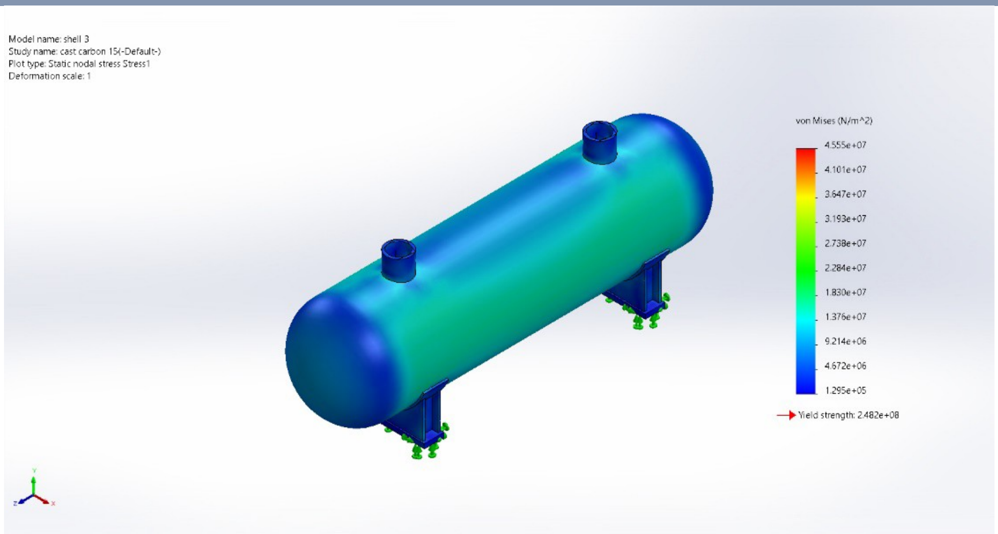


shell 2-cast carbon 15-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 15mm:

STRESS ANALYSIS:

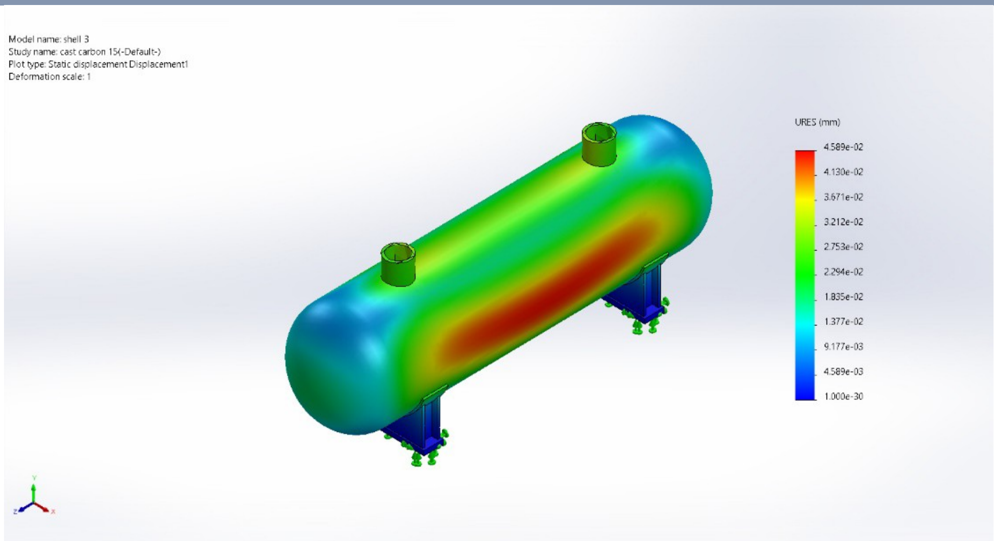
Name	**Type**	**Min**	**Max**
Stress1	VON: von Mises Stress	1.295e+05N/m^2 Node: 3164	4.555e+07N/m^2 Node: 9888



shell 3-cast carbon 15-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2587	4.589e-02mm Node: 475

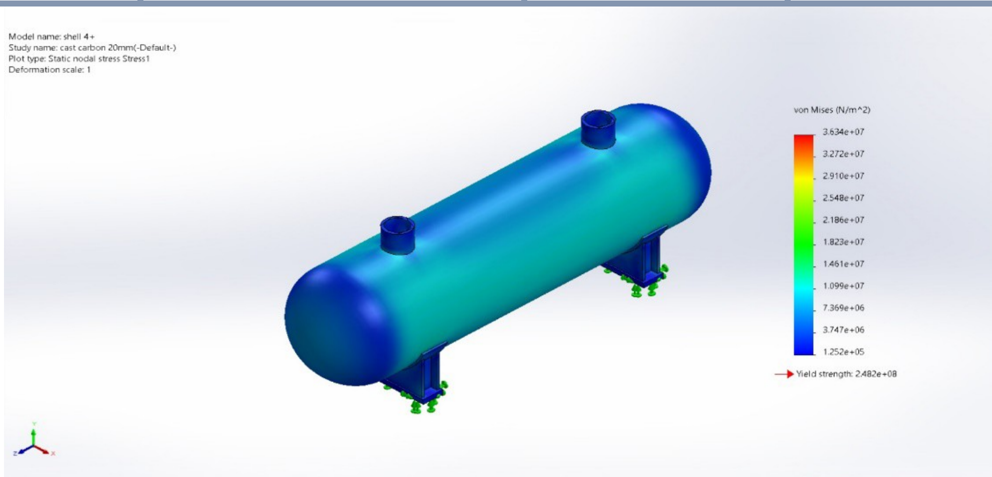


shell 3-cast carbon 15-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 20mm:

STRESS ANALYSIS:

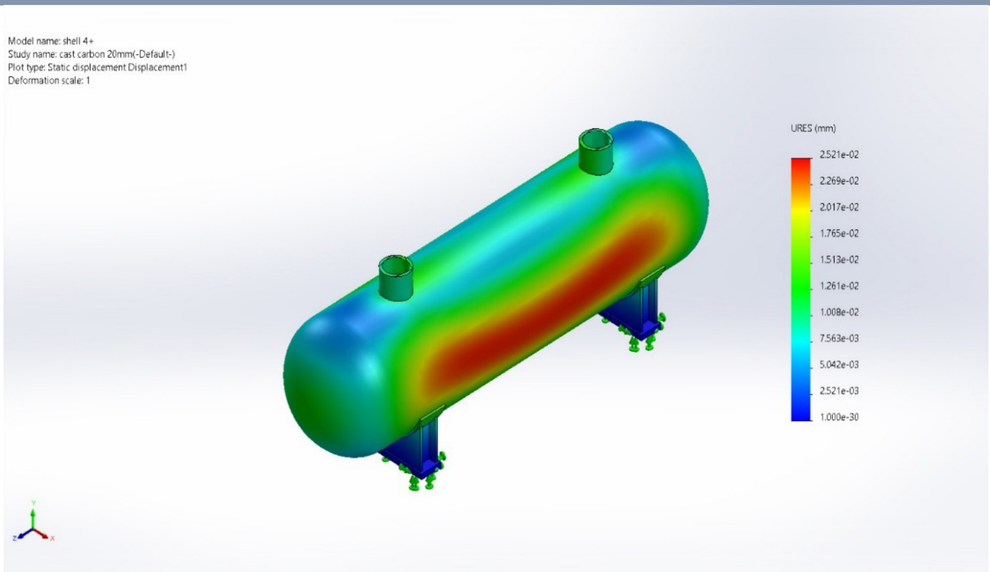
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.252e+05N/m^2 Node: 2645	3.634e+07N/m^2 Node: 9786



shell 4+-cast carbon 20mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2566	2.521e-02mm Node: 13745

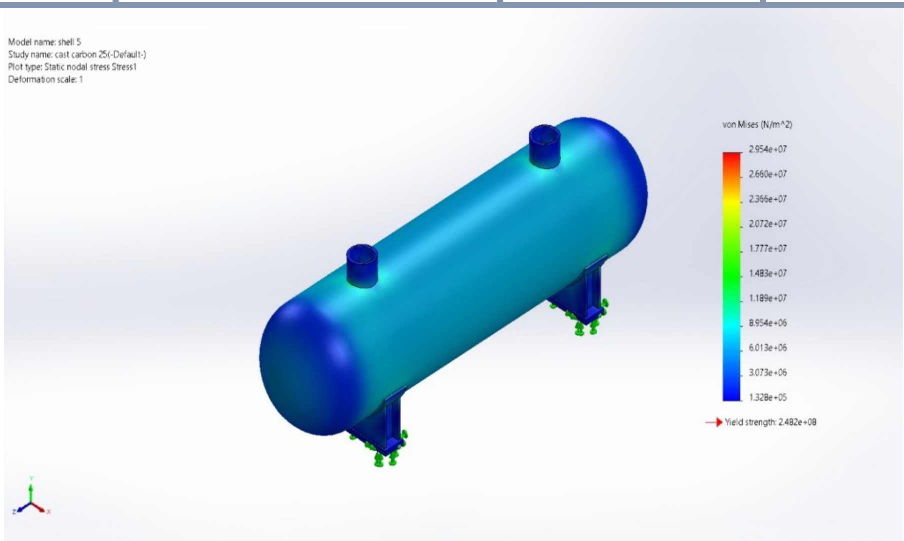


shell 4+-cast carbon 20mm-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESSS 25mm:

STRESS ANALYSIS:

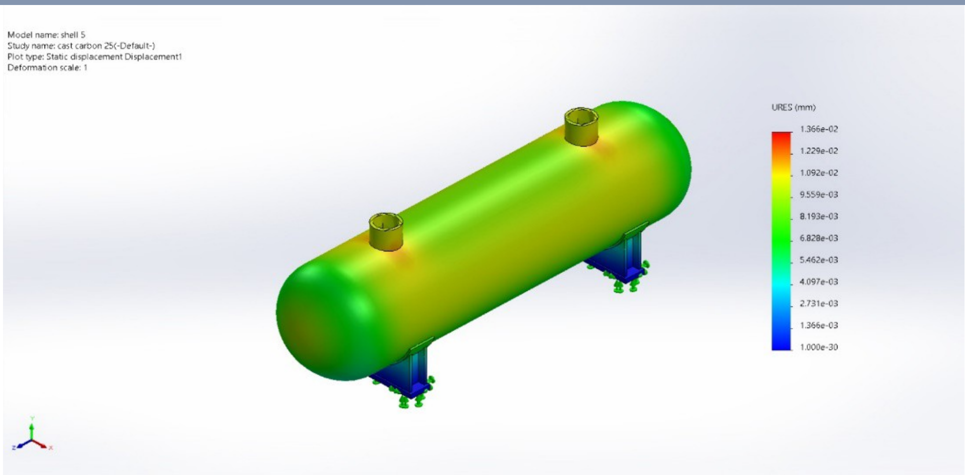
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.328e+05N/m^2 Node: 6641	2.954e+07N/m^2 Node: 1532



shell 5-cast carbon 25-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.366e-02mm Node: 3184

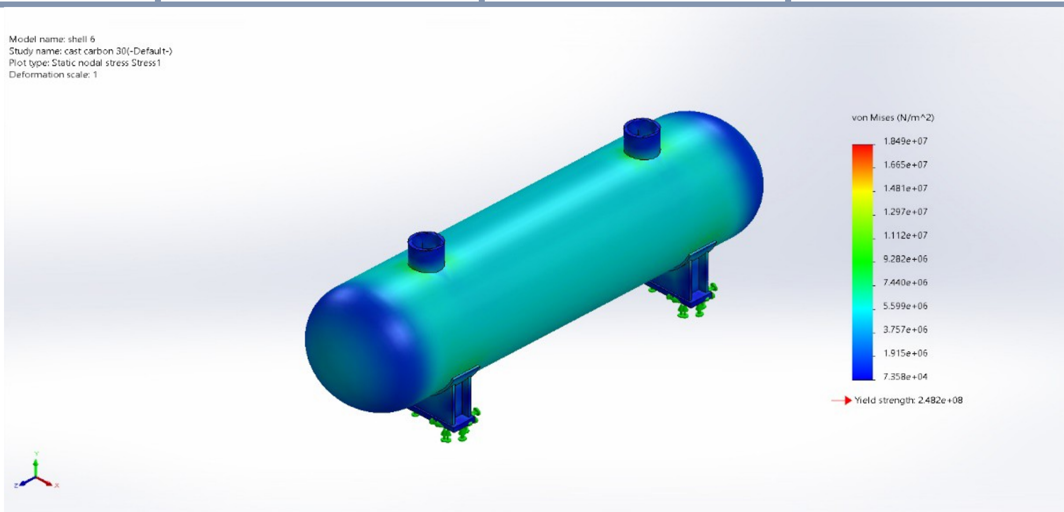


shell 5-cast carbon 25-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 30mm:

STRESS ANALYSIS:

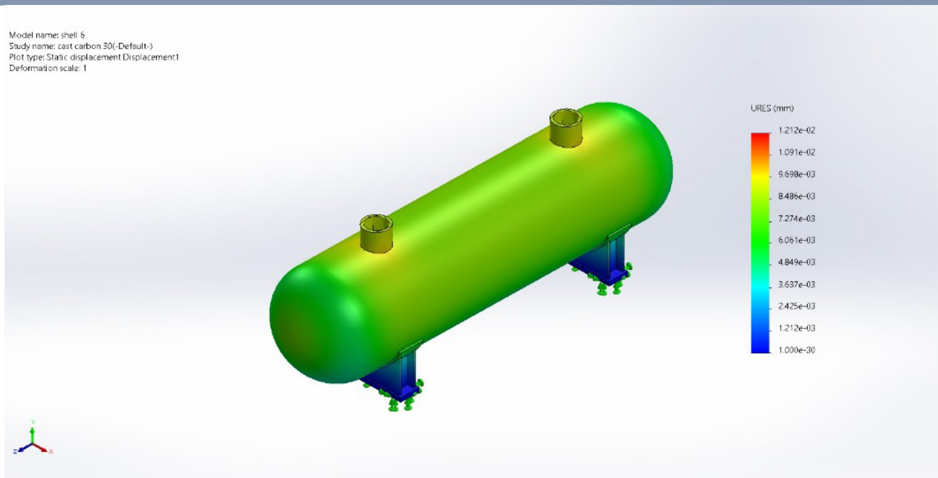
Name	Type	Min	Max
Stress1	VON: von Mises Stress	7.358e+04N/m^2 Node: 995	1.849e+07N/m^2 Node: 1532



shell 6-cast carbon 30-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.212e-02mm Node: 6789

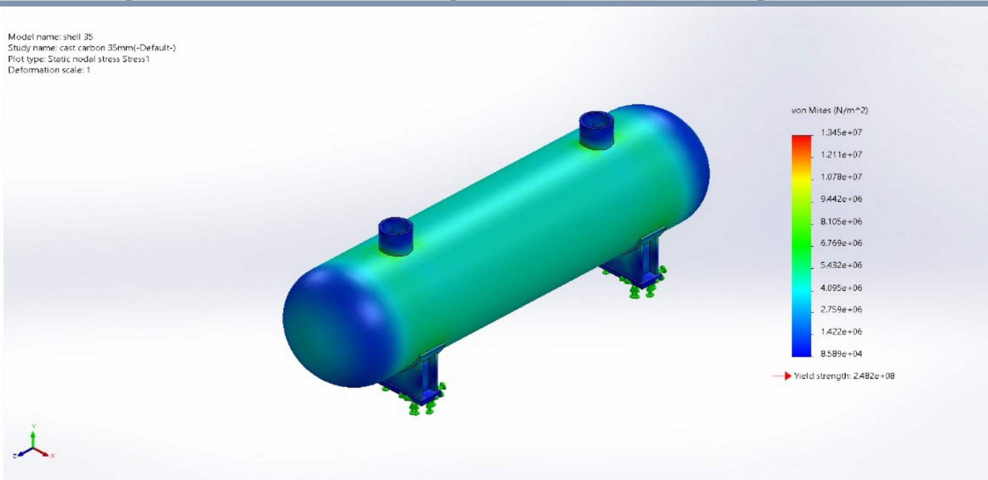


shell 6-cast carbon 30-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 35mm:

STRESS ANALSIS:

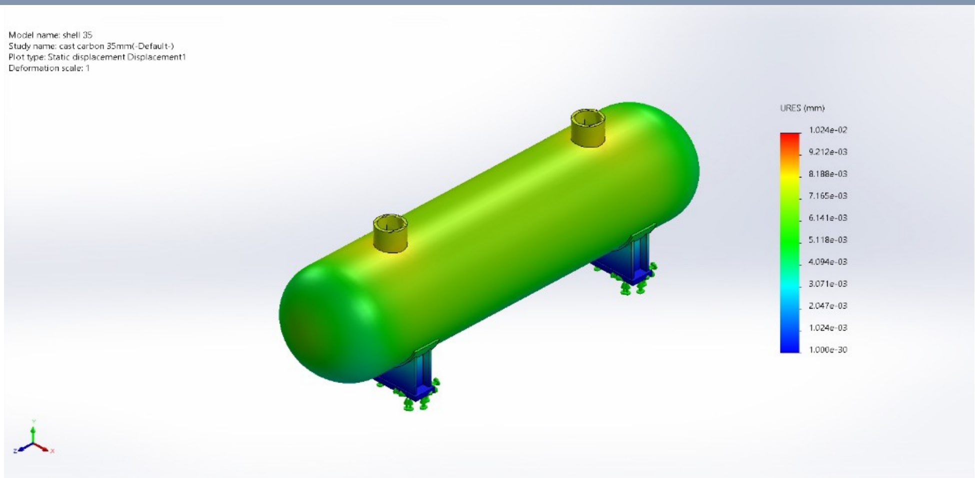
Name	Type	Min	Max
Stress1	VON: von Mises Stress	8.589e+04N/m^2 Node: 6426	1.345e+07N/m^2 Node: 1527



shell 35-cast carbon 35mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.024e-02mm Node: 2441

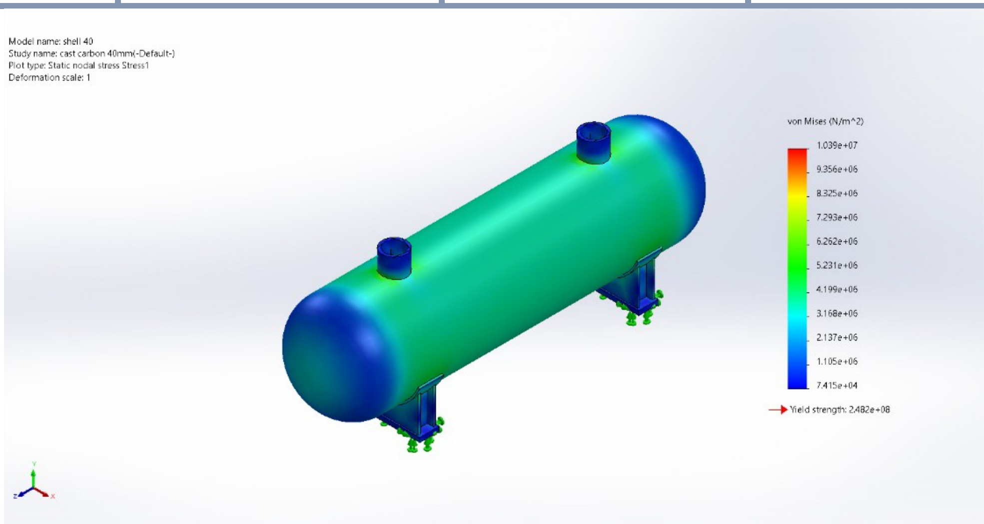


shell 35-cast carbon 35mm-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 40mm:

STRESS ANALYSIS:

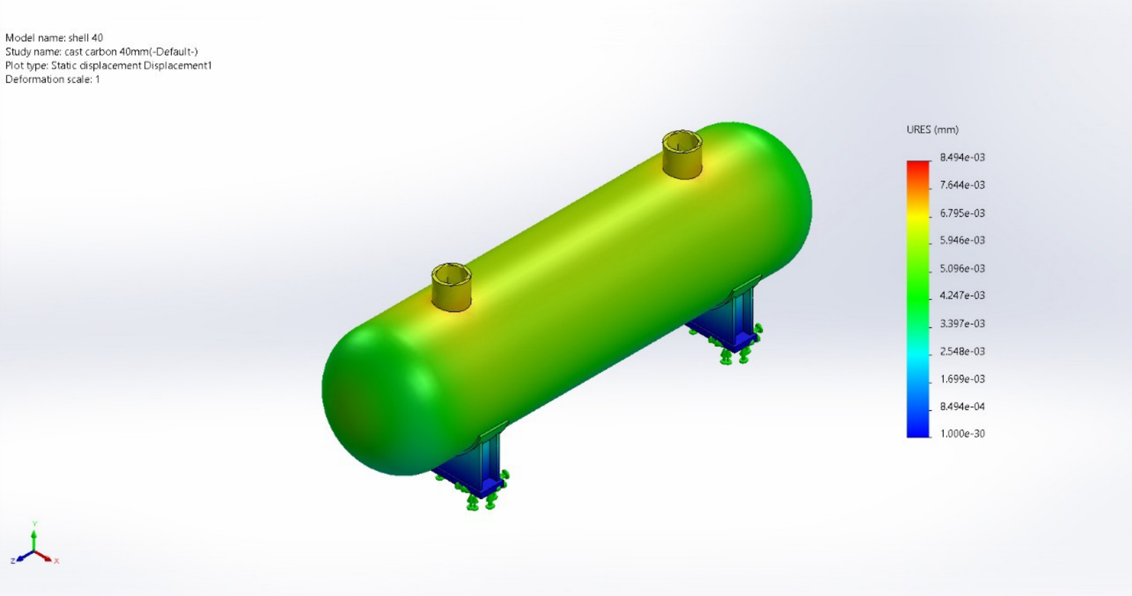
Name	Type	Min	Max
Stress1	VON: von Mises Stress	7.415e+04N/m^2 Node: 8209	1.039e+07N/m^2 Node: 1532



shell 40-cast carbon 40mm-Stress-Stress1

DISPLACEMENTS:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	8.494e-03mm Node: 2429



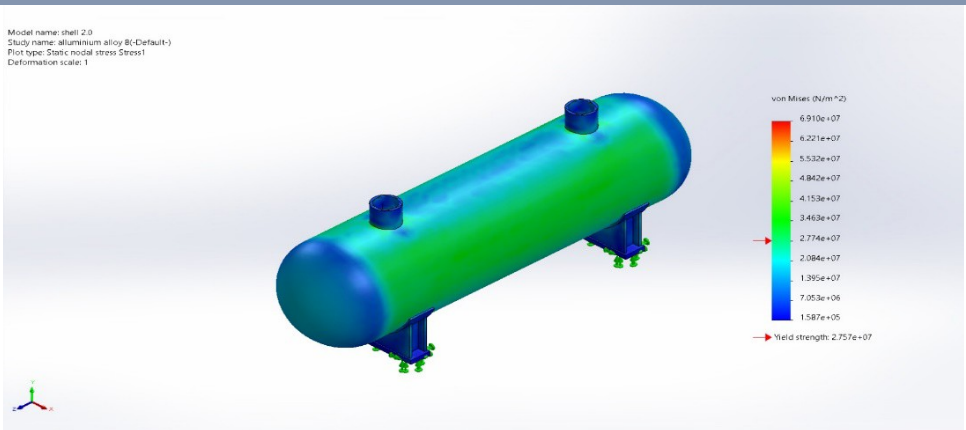
shell 40-cast carbon 40mm-Displacement-Displacement1

➢ **ALUMINIUM ALLOY:**

PRESSURE VESSEL OF THICKNESS 8mm:

STRESS ANALYSIS:

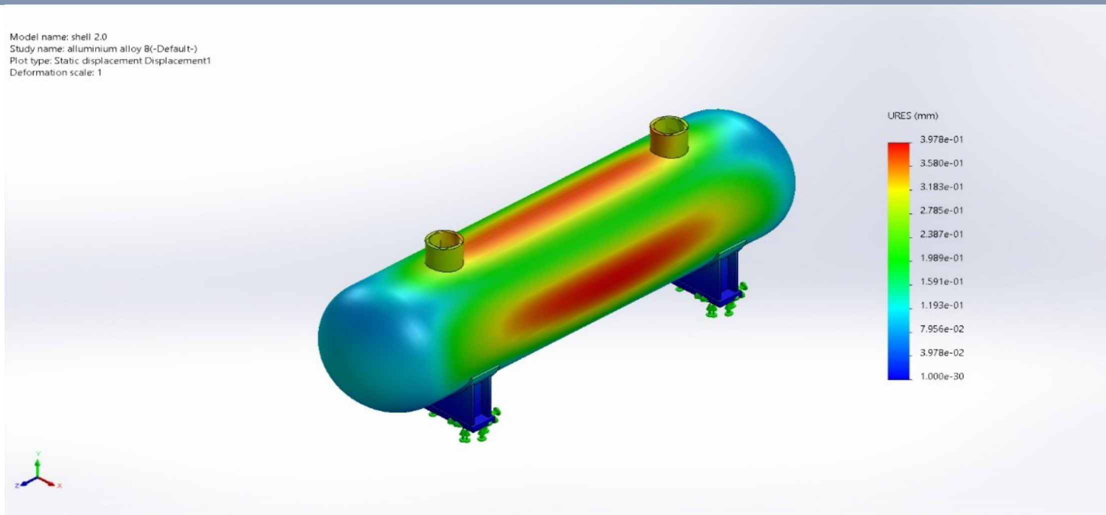
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.587e+05N/m^2 Node: 3404	6.910e+07N/m^2 Node: 10253



shell 2.0-alluminium alloy 8-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2676	3.978e-01mm Node: 2512

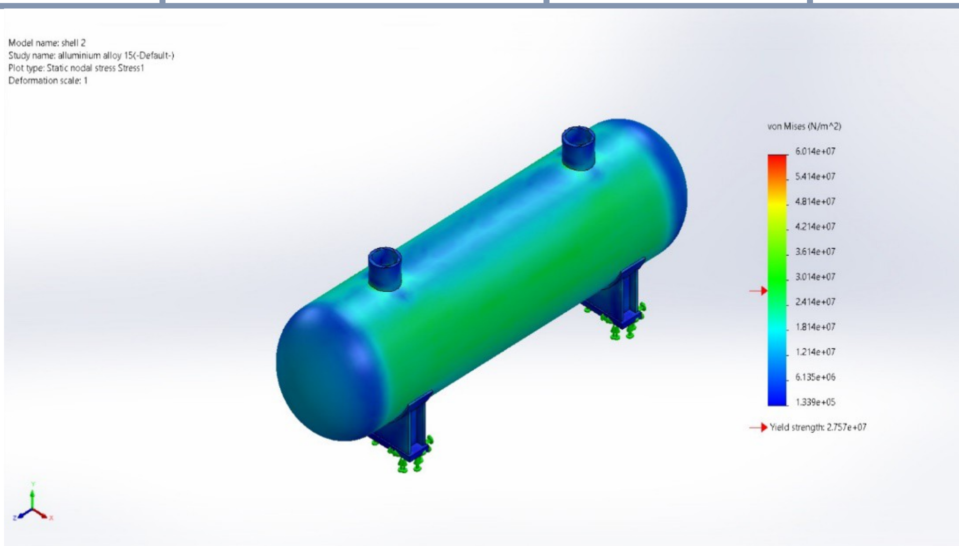


shell 2.0-alluminium alloy 8-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 10mm:

STRESS ANALYSIS:

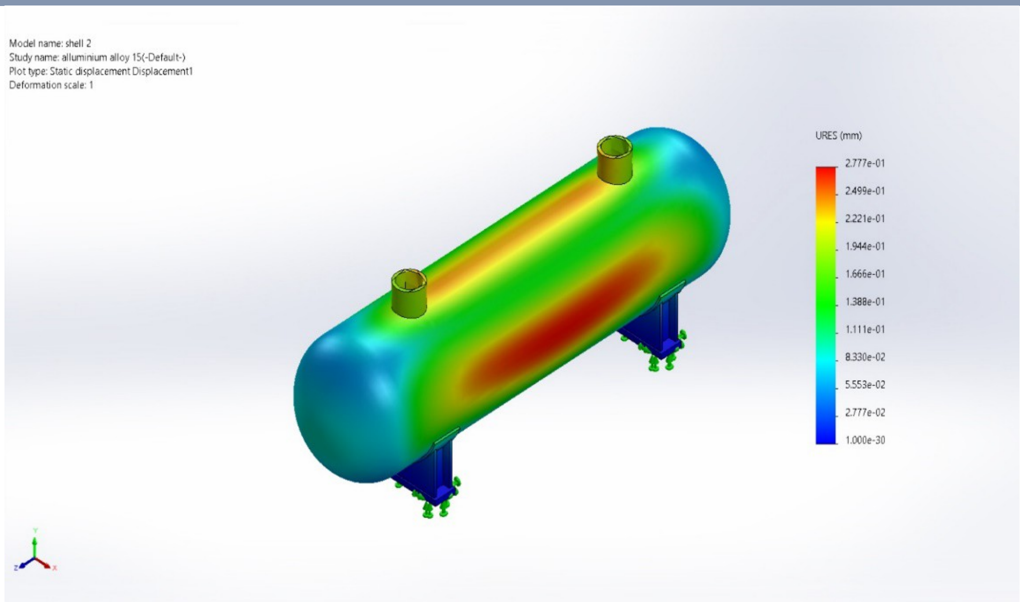
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.339e+05N/m^2 Node: 3235	6.014e+07N/m^2 Node: 10255



shell 2-alluminium alloy 15-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2679	2.777e-01mm Node: 12135

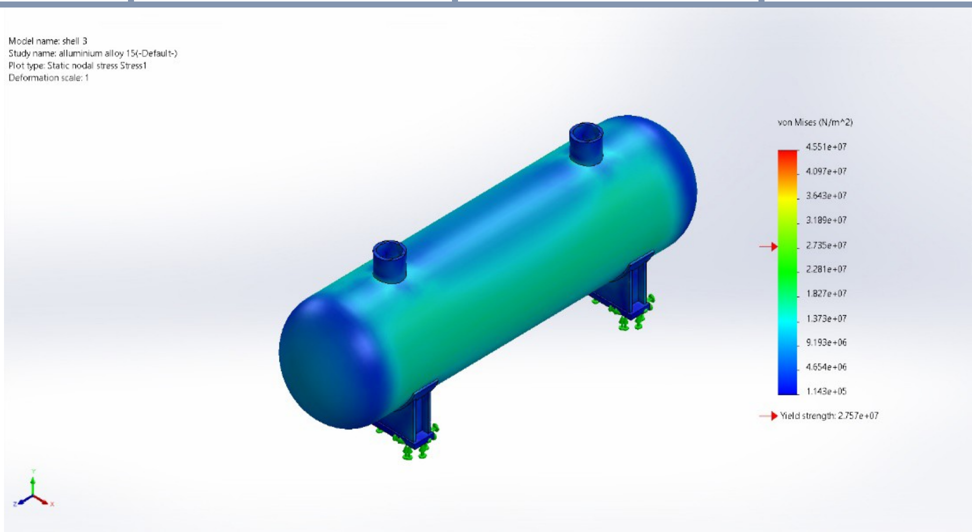


shell 2-alluminium alloy 15-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 15mm:

STRESS ANALYSIS:

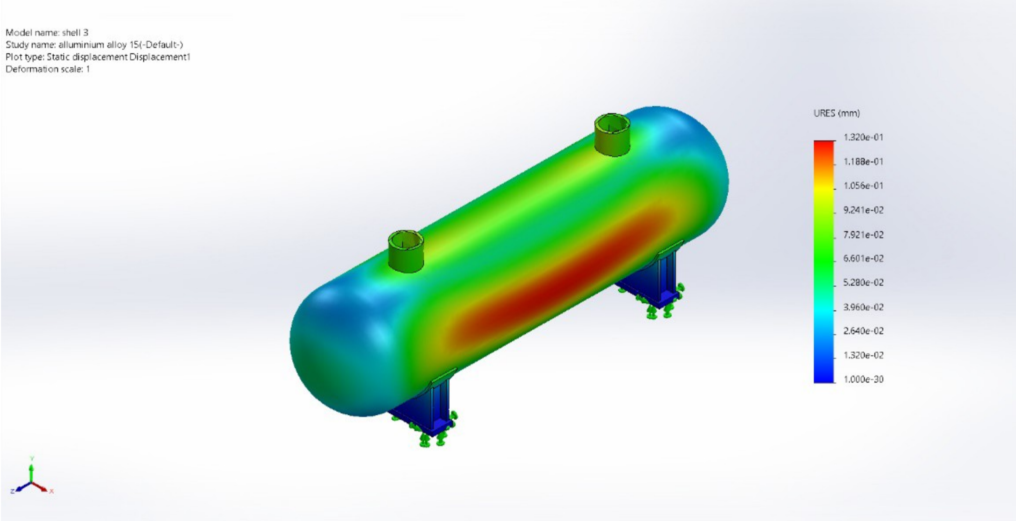
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.143e+05N/m^2 Node: 3164	4.551e+07N/m^2 Node: 9888



shell 3-alluminium alloy 15-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2587	1.320e-01mm Node: 475

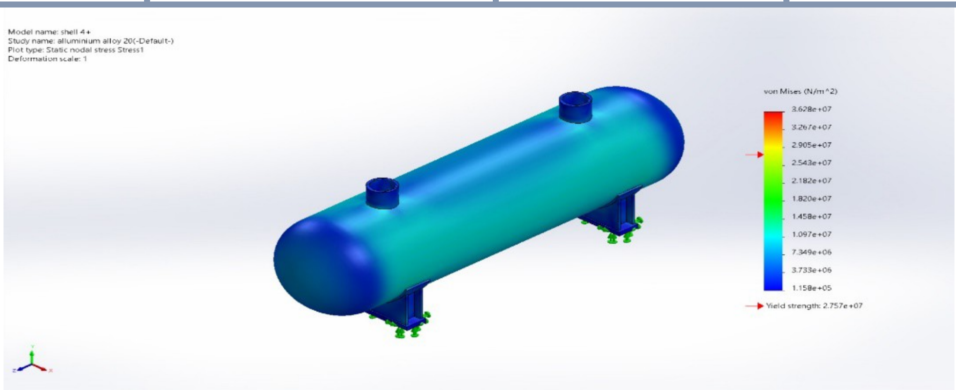


shell 3-alluminium alloy 15-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 20mm:

STRESS ANALYSIS:

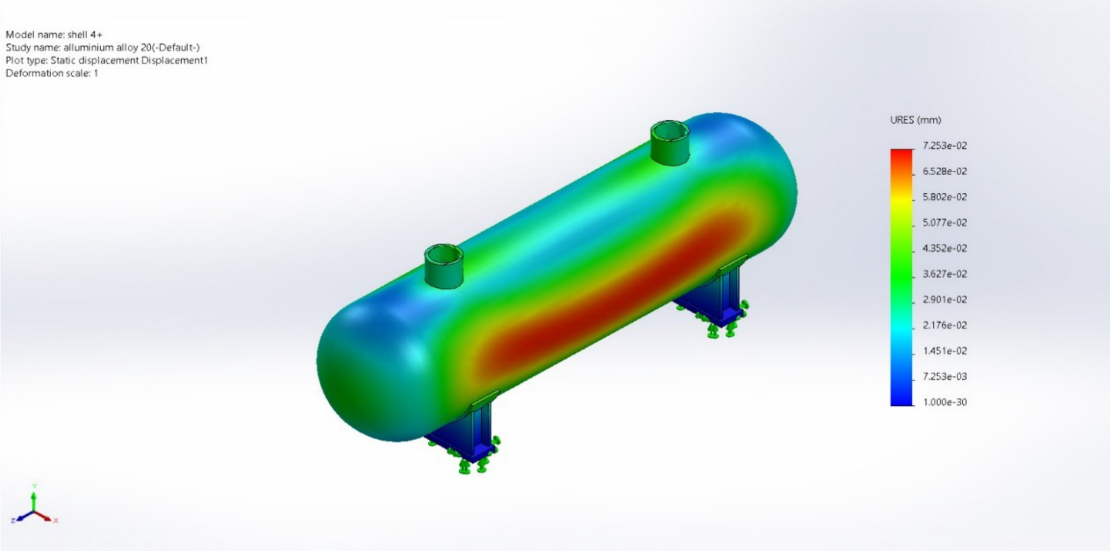
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.158e+05N/m^2 Node: 2645	3.628e+07N/m^2 Node: 9786



shell 4+-aluminum alloy 20-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2566	7.253e-02mm Node: 13745

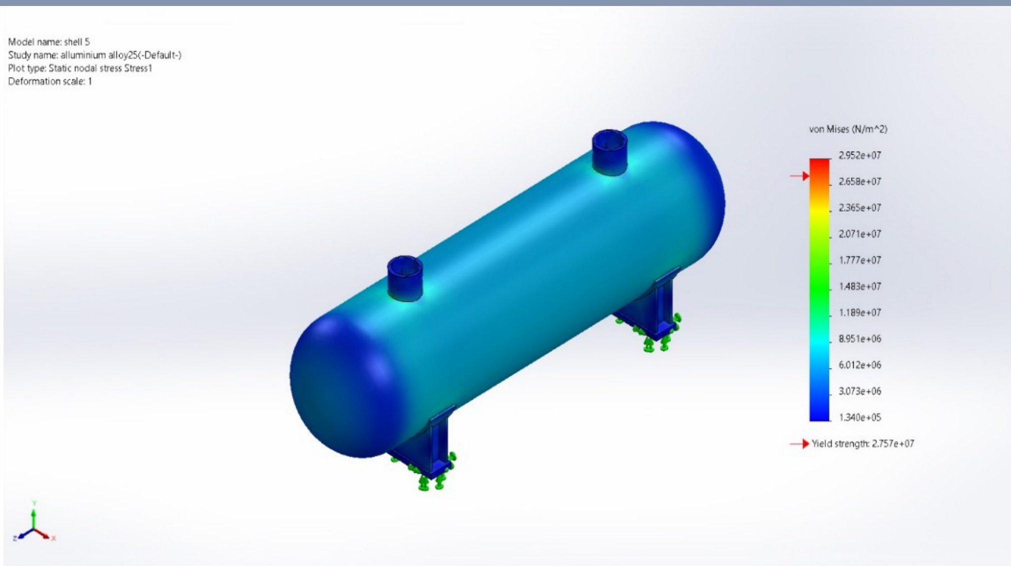


shell 4+-aluminum alloy 20-Displacement-Displacement1

PRESSUERE VESSEL OF THICKNESS 25mm:

STRESS ANALYSIS:

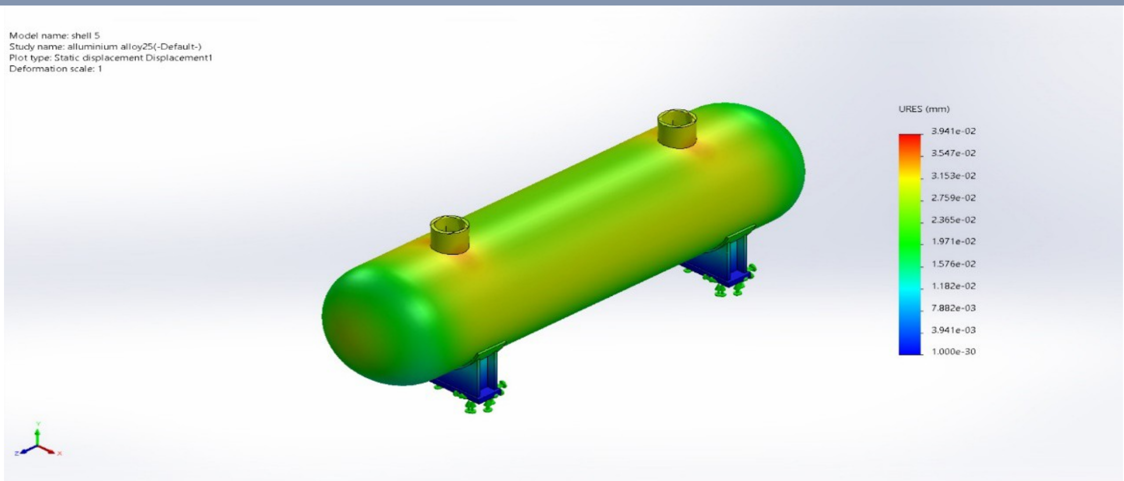
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.340e+05N/m^2 Node: 6517	2.952e+07N/m^2 Node: 1532



shell 5-alluminium alloy25-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	3.941e-02mm Node: 3184

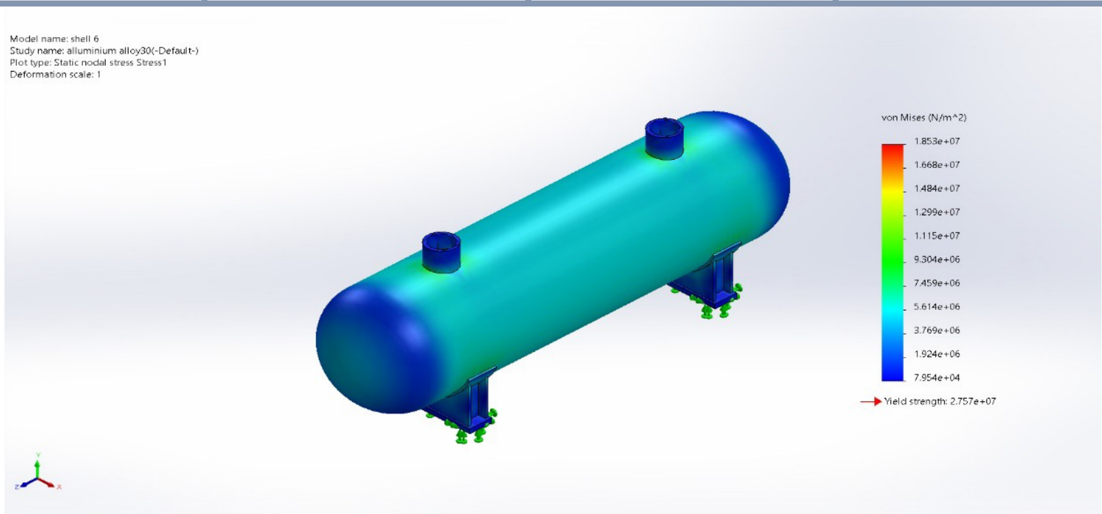


shell 5-alluminium alloy25-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 30mm:

STRESS ANALYSIS:

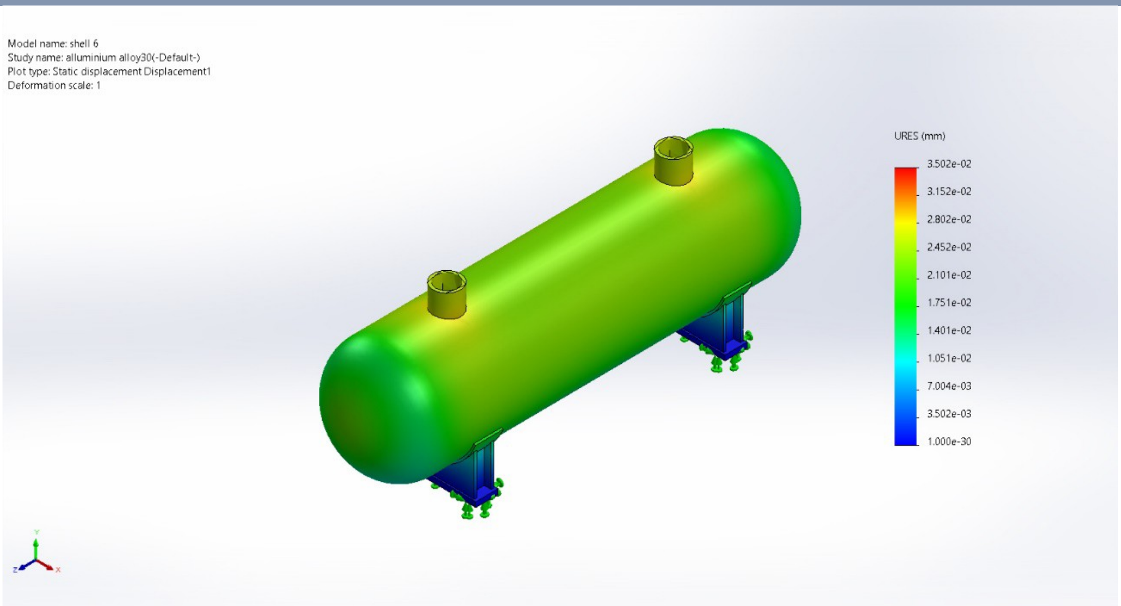
Name	Type	Min	Max
Stress1	VON: von Mises Stress	7.954e+04N/m^2 Node: 995	1.853e+07N/m^2 Node: 1532



shell 6-alluminium alloy30-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	3.502e-02mm Node: 3179

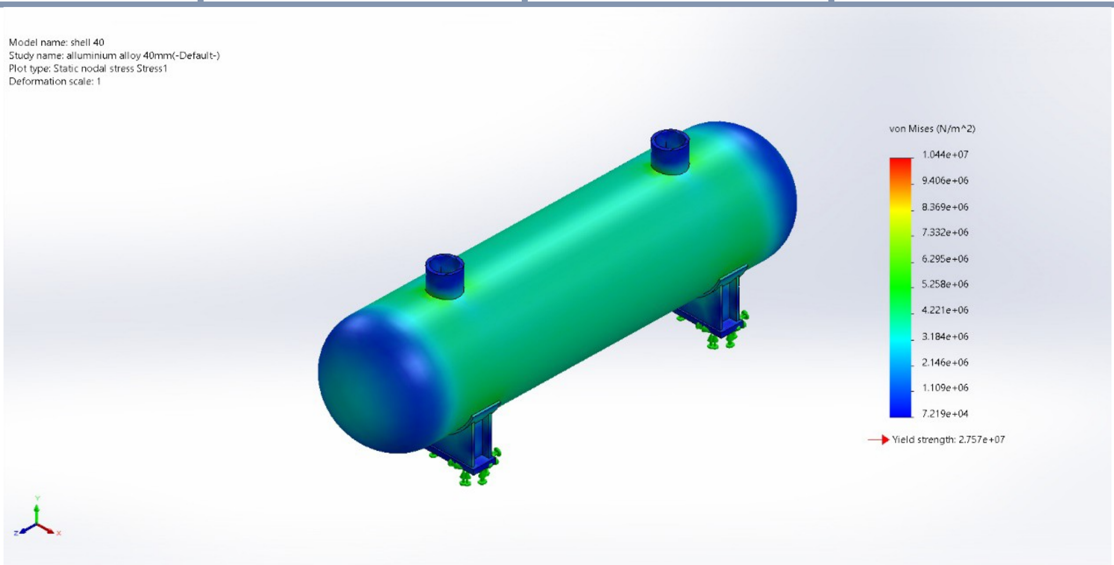


shell 6-alluminium alloy30-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 35mm:

STRESS ANALYSIS:

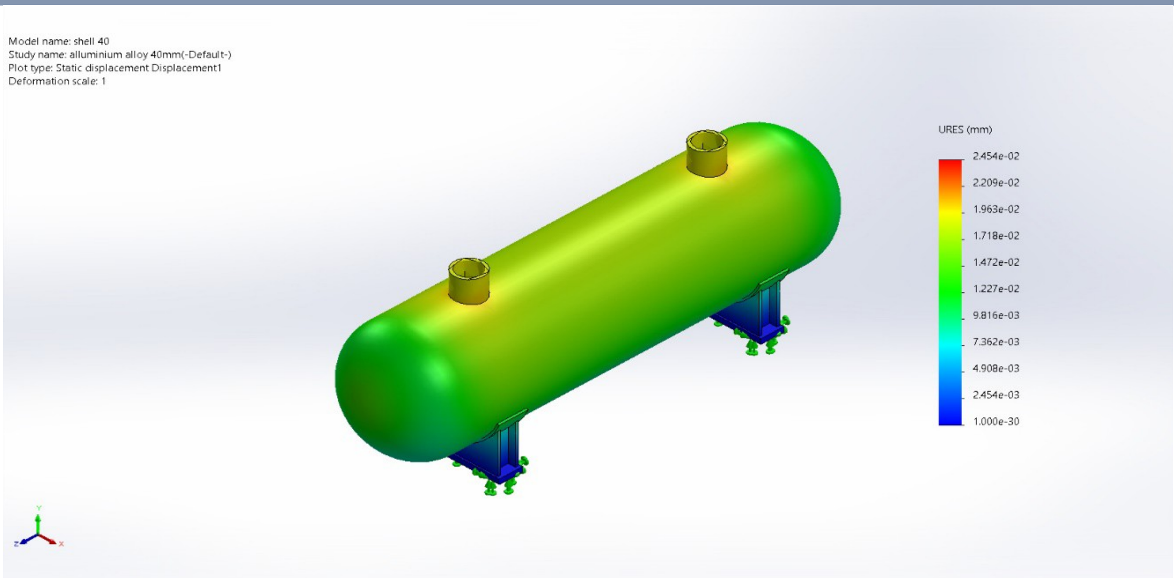
Name	Type	Min	Max
Stress1	VON: von Mises Stress	7.219e+04N/m^2 Node: 8209	1.044e+07N/m^2 Node: 1532



shell 40-alluminium alloy 40mm-Stress-Stress1

DISPLACEMENTS:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	2.454e-02mm Node: 2429



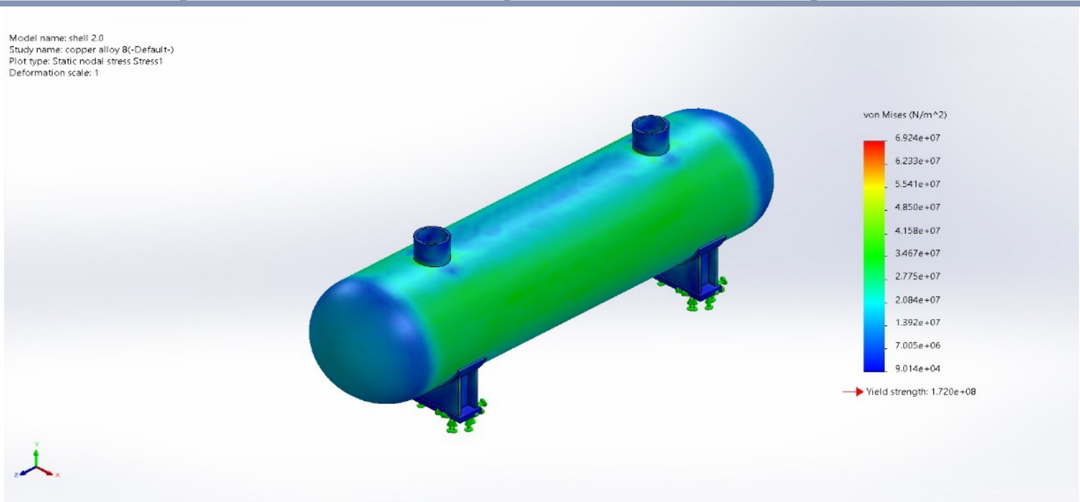
shell 40-alluminium alloy 40mm-Displacement-Displacement1

- **COPPER ALLOY:**

PRESSURE VESSEL OF THICKNESS 8mm:

STRESS ANALYSIS:

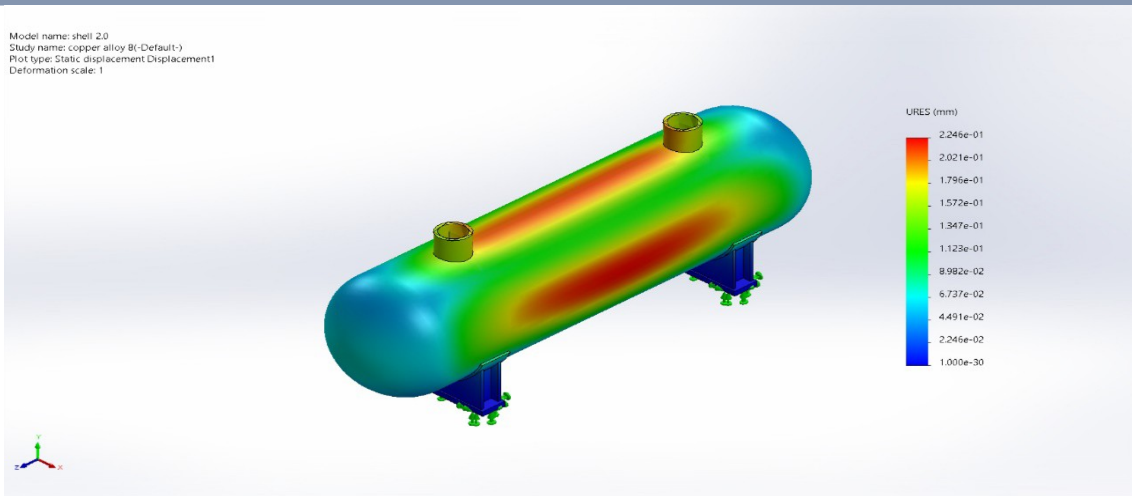
Name	Type	Min	Max
Stress1	VON: von Mises Stress	9.014e+04N/m^2 Node: 3501	6.924e+07N/m^2 Node: 10253



shell 2.0-copper alloy 8-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2676	2.246e-01mm Node: 2512

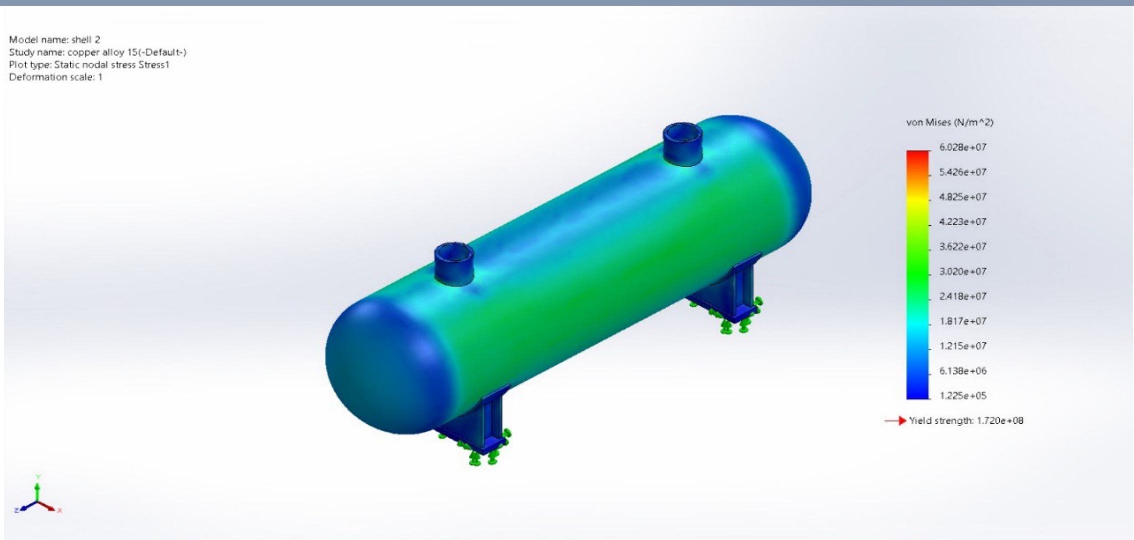


shell 2.0-copper alloy 8-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 10mm

STRESS ANALYSIS

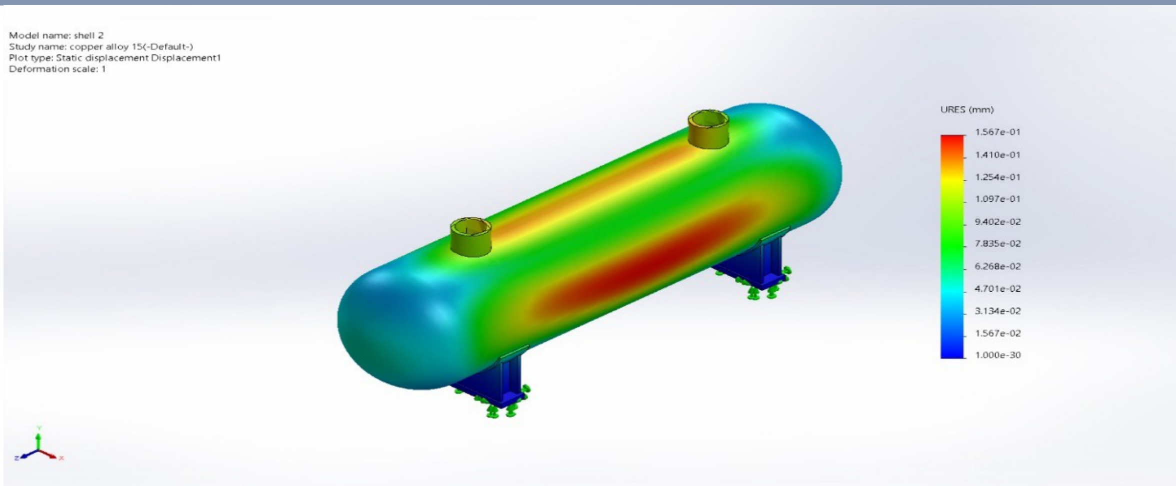
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.225e+05N/m^2 Node: 3421	6.028e+07N/m^2 Node: 10255



shell 2-copper alloy 15-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2679	1.567e-01mm Node: 12135

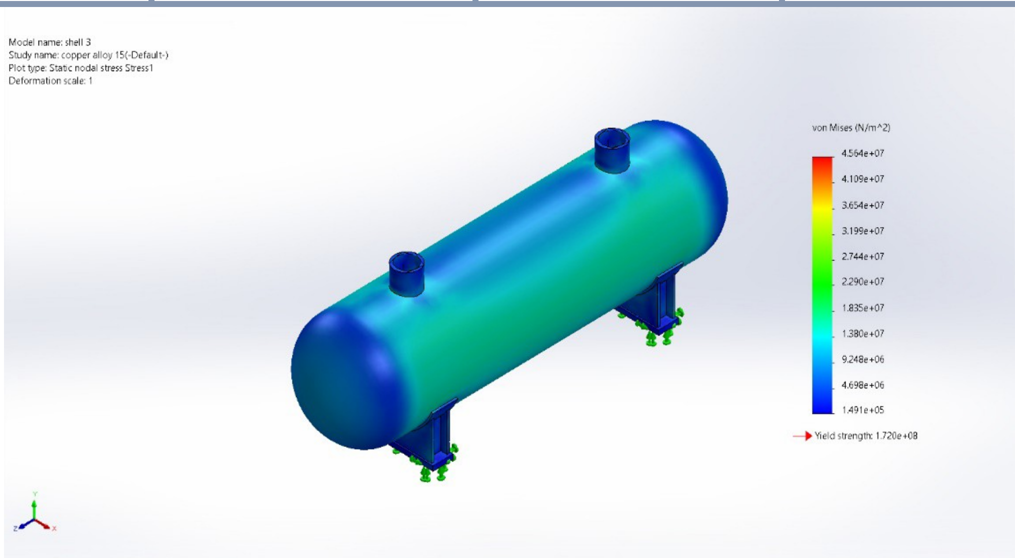


shell 2-copper alloy 15-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 15mm:

STRESS ANALYSIS:

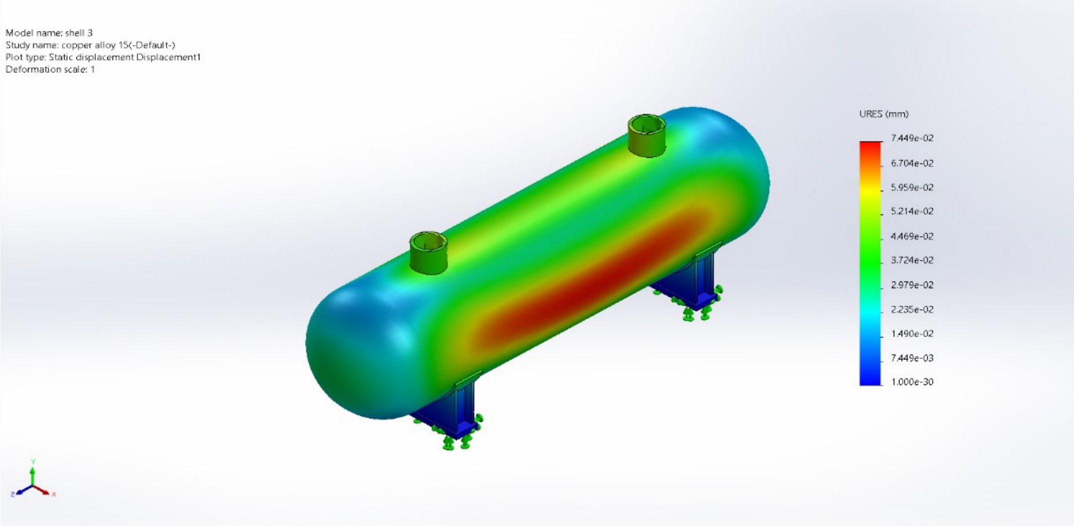
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.491e+05N/m^2 Node: 3350	4.564e+07N/m^2 Node: 9888



shell 3-copper alloy 15-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2587	7.449e-02mm Node: 475

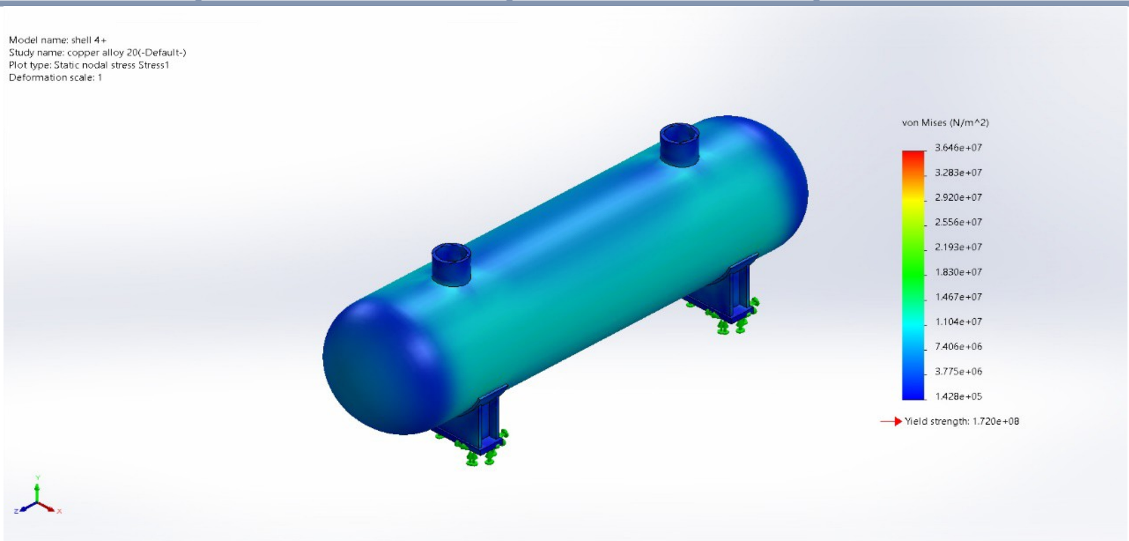


shell 3-copper alloy 15-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 20mm:

STRESS ANALYSIS:

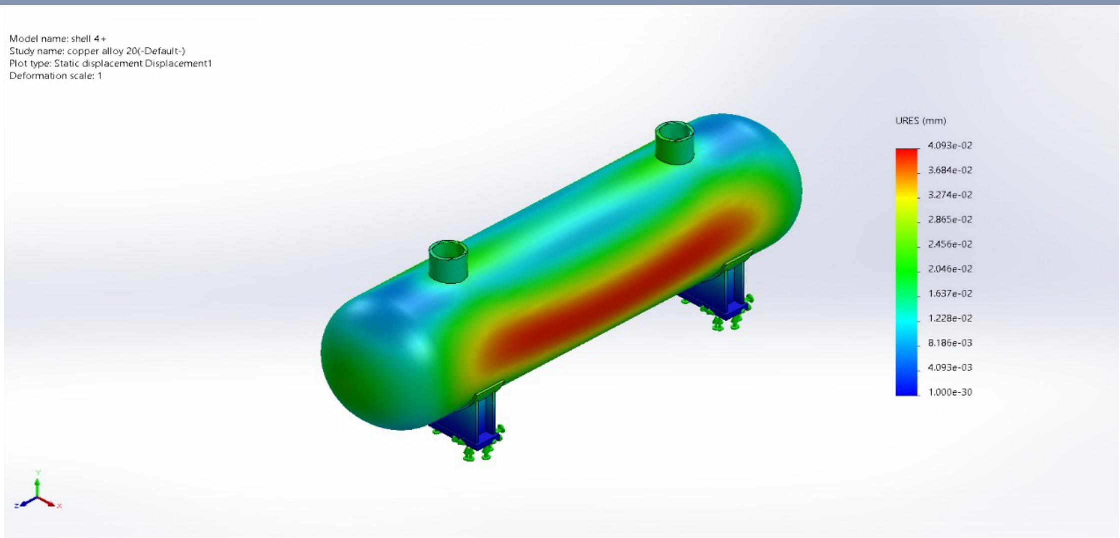
Name		Type	Min	Max
Stress1		VON: von Mises Stress	1.428e+05N/m^2 Node: 3105	3.646e+07N/m^2 Node: 9786



shell 4+-copper alloy 20-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2566	4.093e-02mm Node: 13745

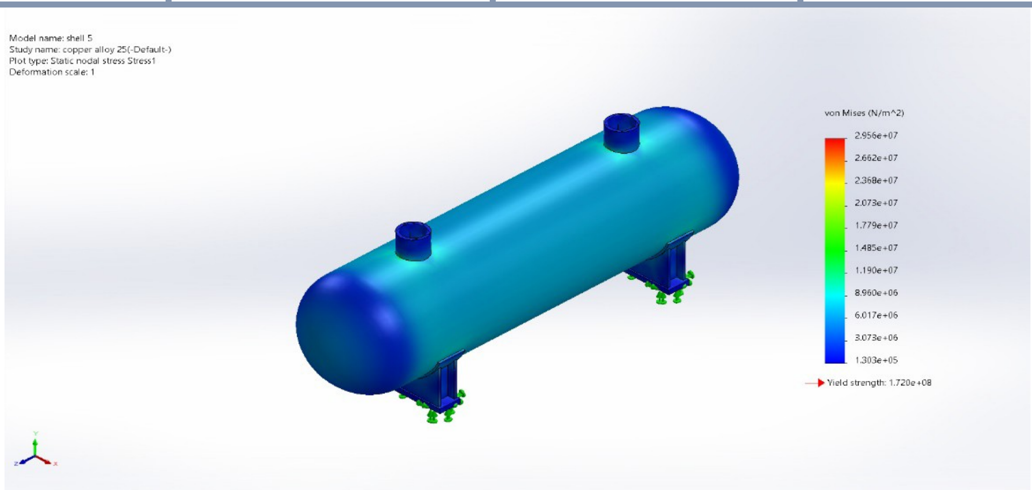


shell 4+-copper alloy 20-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 25mm:

STRESS ANALYSIS:

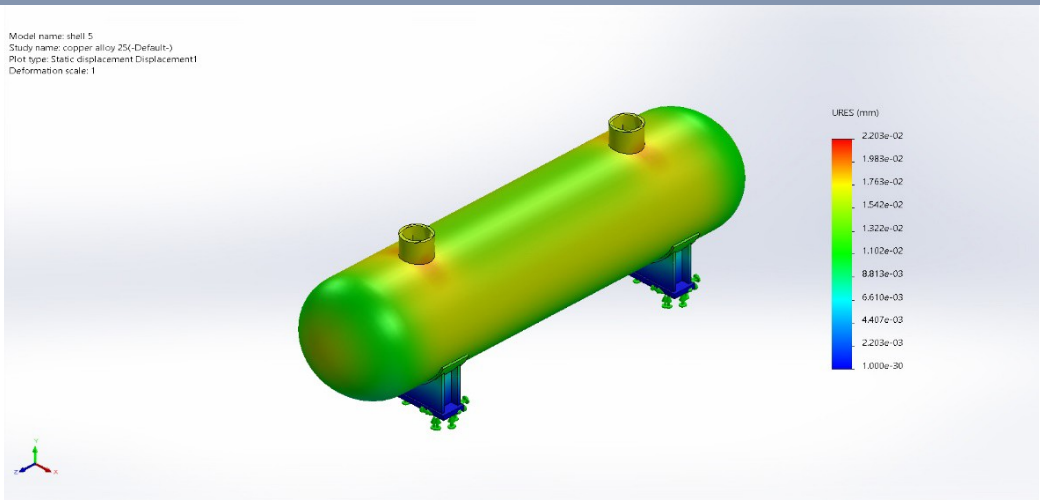
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.303e+05N/m^2 Node: 6376	2.956e+07N/m^2 Node: 1532



shell 5-copper alloy 25-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	2.203e-02mm Node: 3184

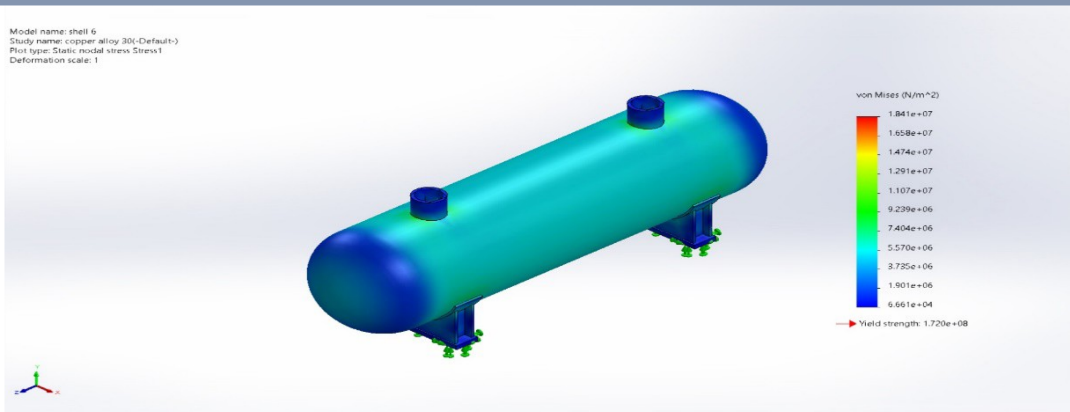


shell 5-copper alloy 25-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 30mm:

STRESS ANALYSIS:

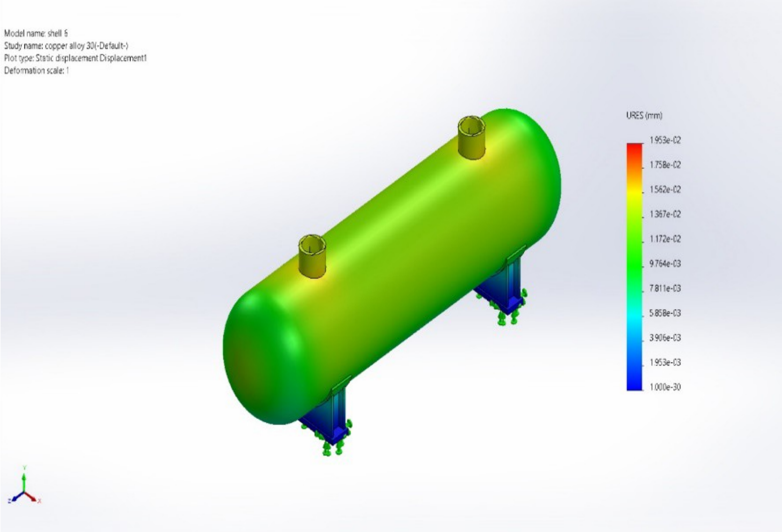
Name	Type	Min	Max
Stress1	VON: von Mises Stress	6.661e+04N/m^2 Node: 995	1.841e+07N/m^2 Node: 1532



shell 6-copper alloy 30-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.953e-02mm Node: 6789

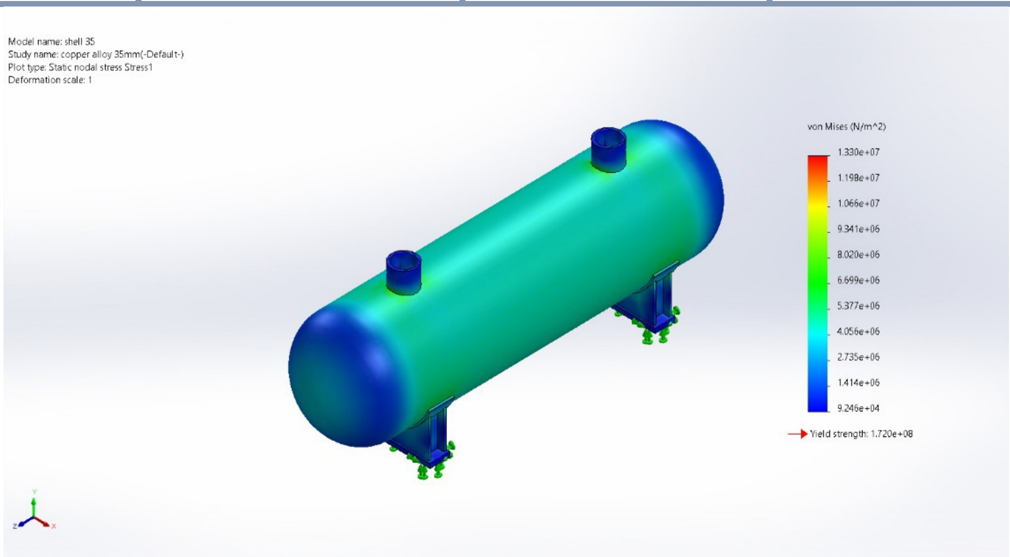


shell 6-copper alloy 30-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 35mm:

STRESS ANALYSIS:

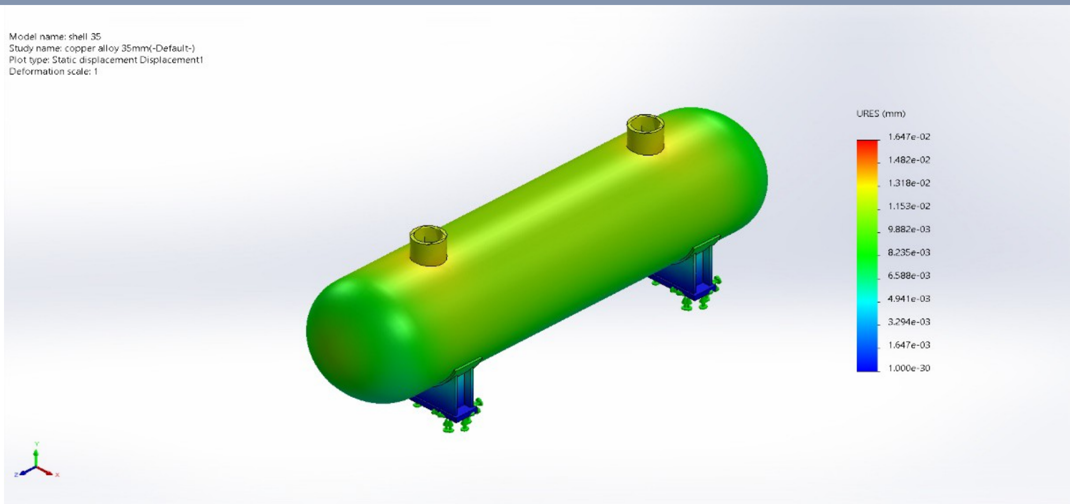
Name	Type	Min	Max
Stress1	VON: von Mises Stress	9.246e+04N/m^2 Node: 6426	1.330e+07N/m^2 Node: 1527



shell 35-copper alloy 35mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.647e-02mm Node: 2441

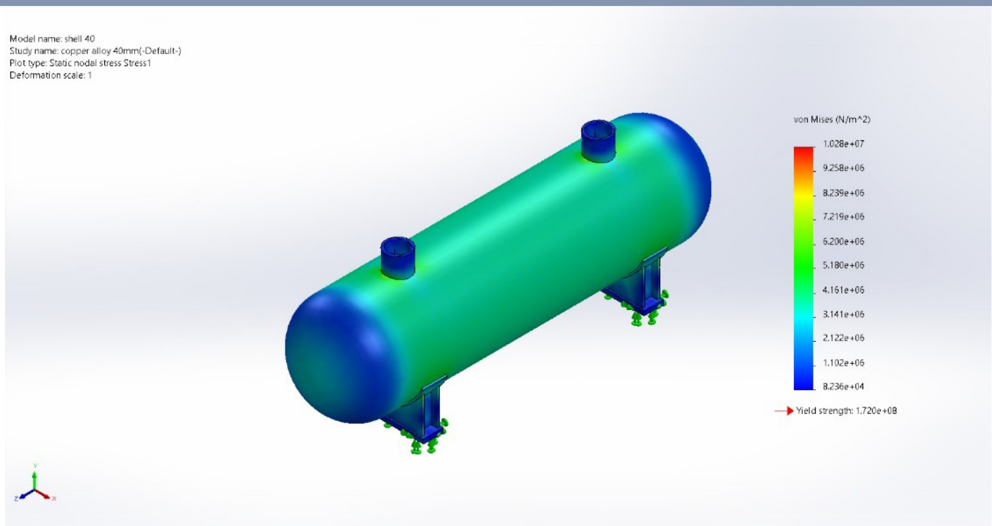


shell 35-copper alloy 35mm-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 40mm:

STRESS ANALYSIS:

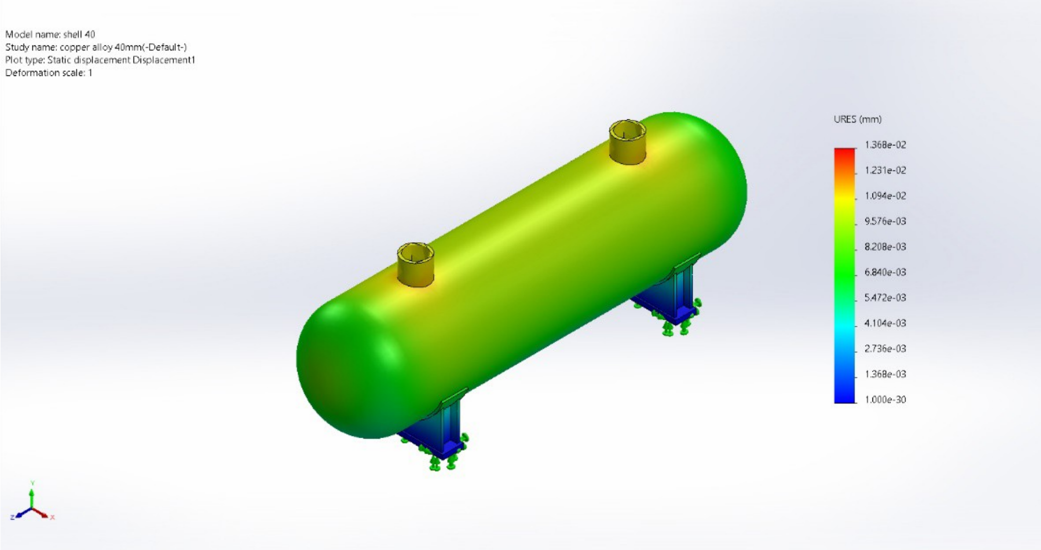
Name	Type	Min	Max
Stress1	VON: von Mises Stress	8.236e+04N/m^2 Node: 8209	1.028e+07N/m^2 Node: 1532



shell 40-copper alloy 40mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.368e-02mm Node: 2429



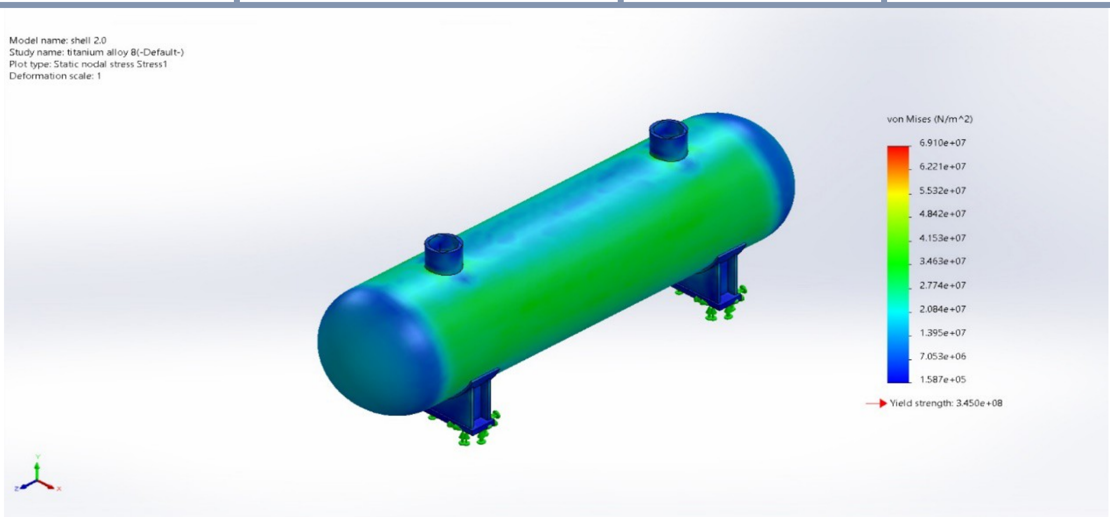
shell 40-copper alloy 40mm-Displacement-Displacement1

➢ **TITANIUM ALLOY:**

PRESSURE ANALSIS OF THICKNESS 8mm:

STERESS ANALYSIS:

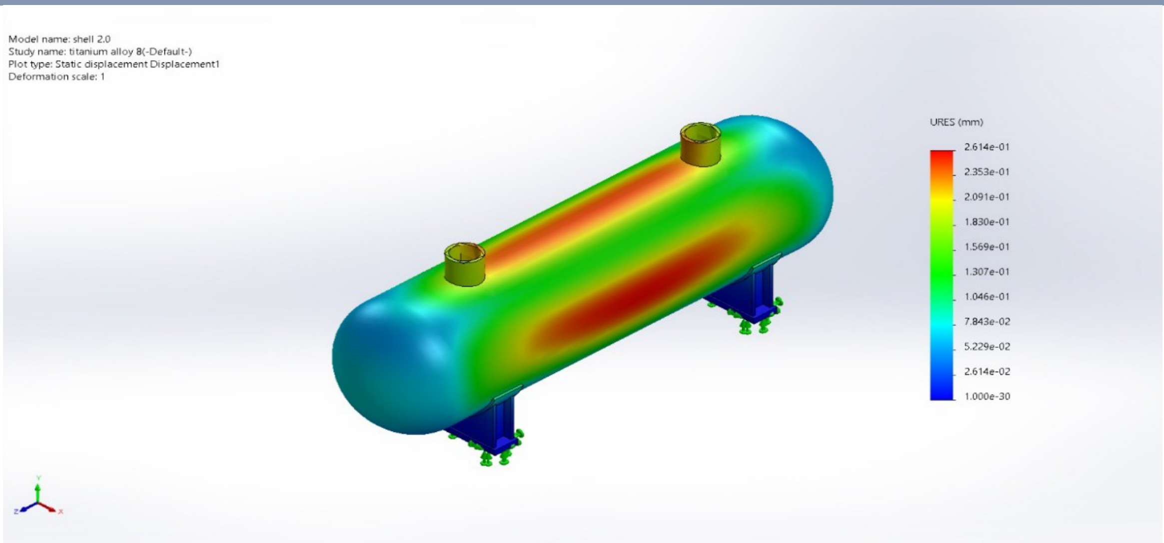
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.587e+05N/m^2 Node: 3404	6.910e+07N/m^2 Node: 10253



shell 2.0-titanium alloy 8-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2676	2.614e-01mm Node: 2512

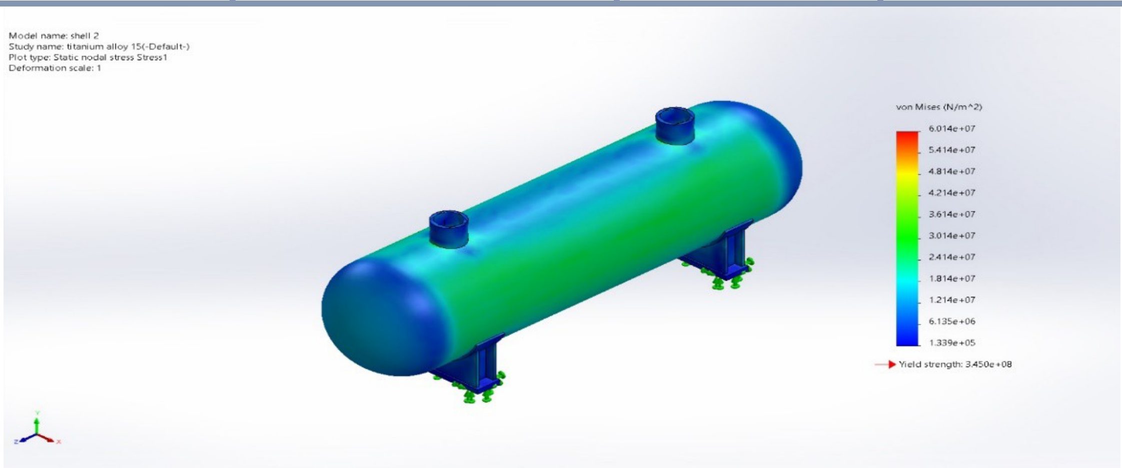


shell 2.0-titanium alloy 8-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 8mm:

STRESS ANALYSIS:

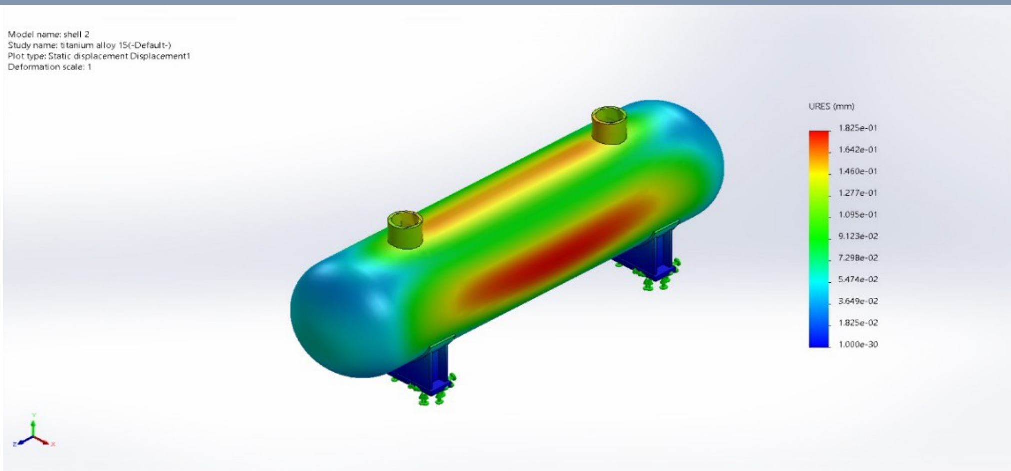
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.339e+05N/m^2 Node: 3235	6.014e+07N/m^2 Node: 10255



shell 2-titanium alloy 15-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2679	1.825e-01mm Node: 12135

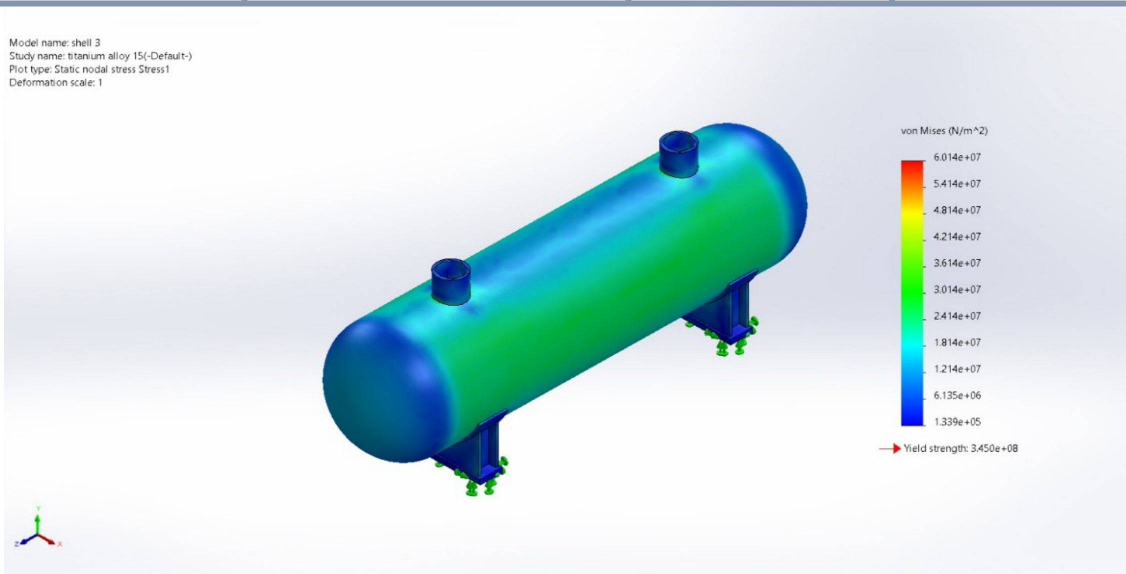


shell 2-titanium alloy 15-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 15mm:

STRESS ANALYSIS:

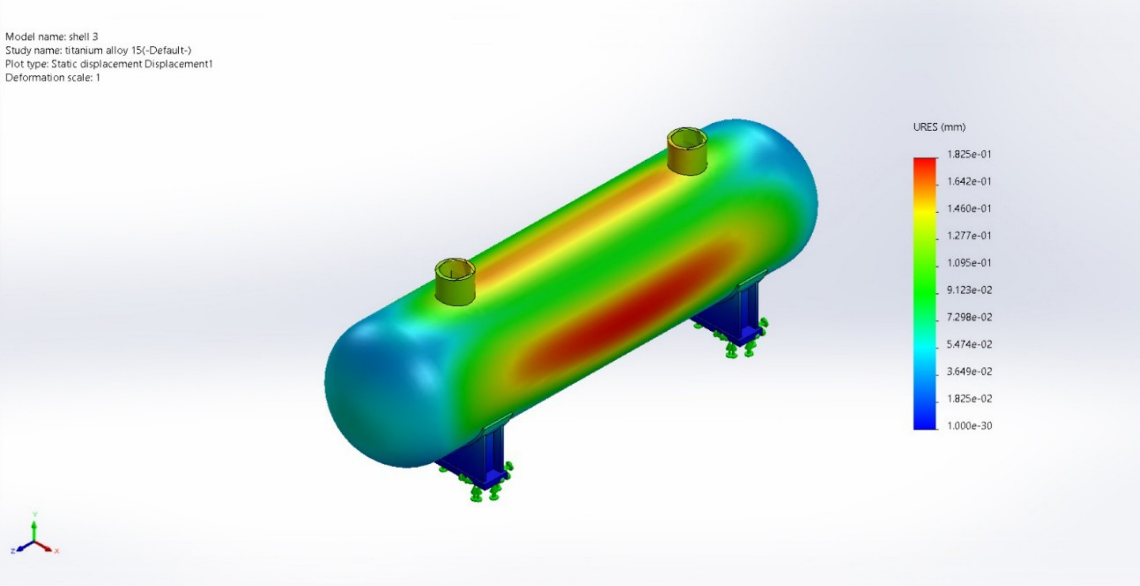
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.339e+05N/m^2 Node: 3235	6.014e+07N/m^2 Node: 10255



shell 3-titanium alloy 15-Stress-Stress1

DISPLACMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2679	1.825e-01mm Node: 12135

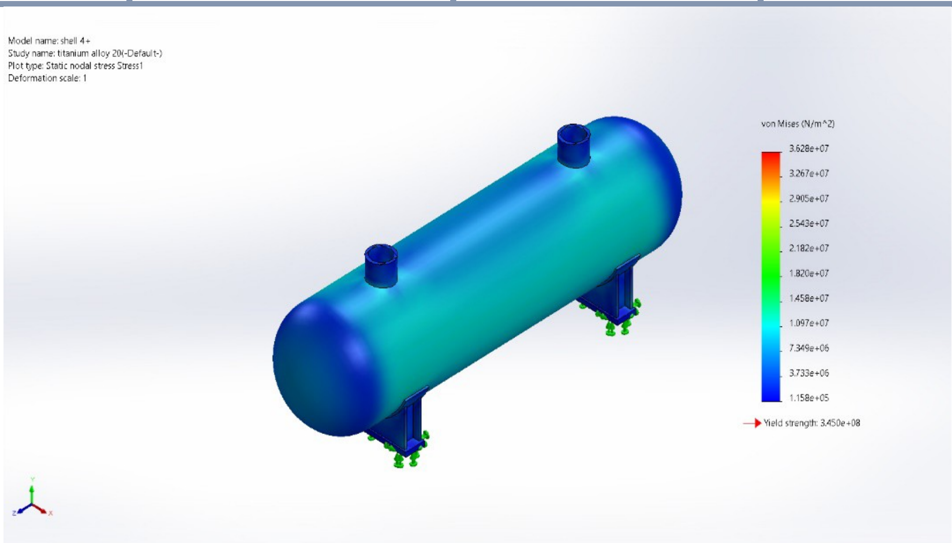


shell 3-titanium alloy 15-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 20mm:

STRESS ANALYSIS:

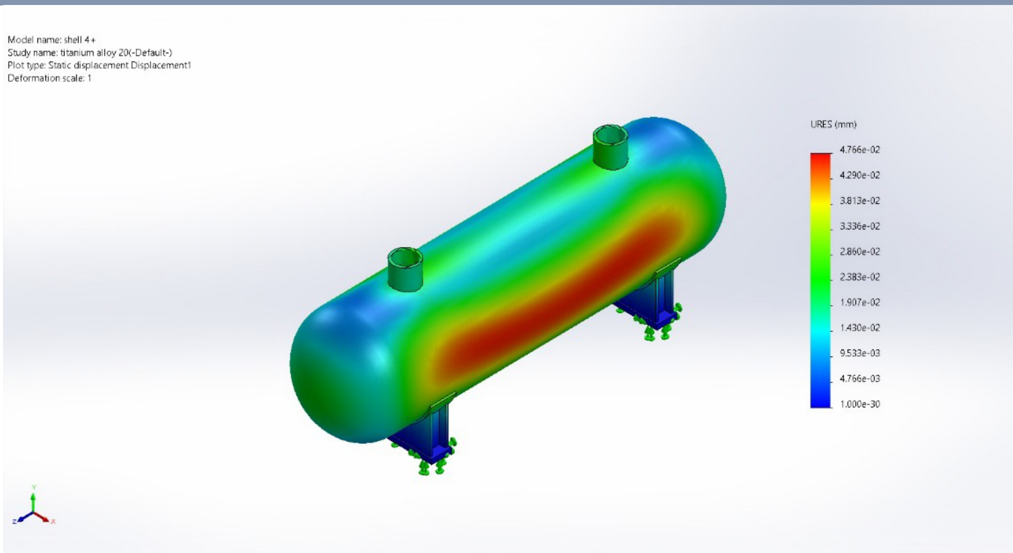
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.158e+05N/m^2 Node: 2645	3.628e+07N/m^2 Node: 9786



shell 4+-titanium alloy 20-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 2566	4.766e-02mm Node: 13745

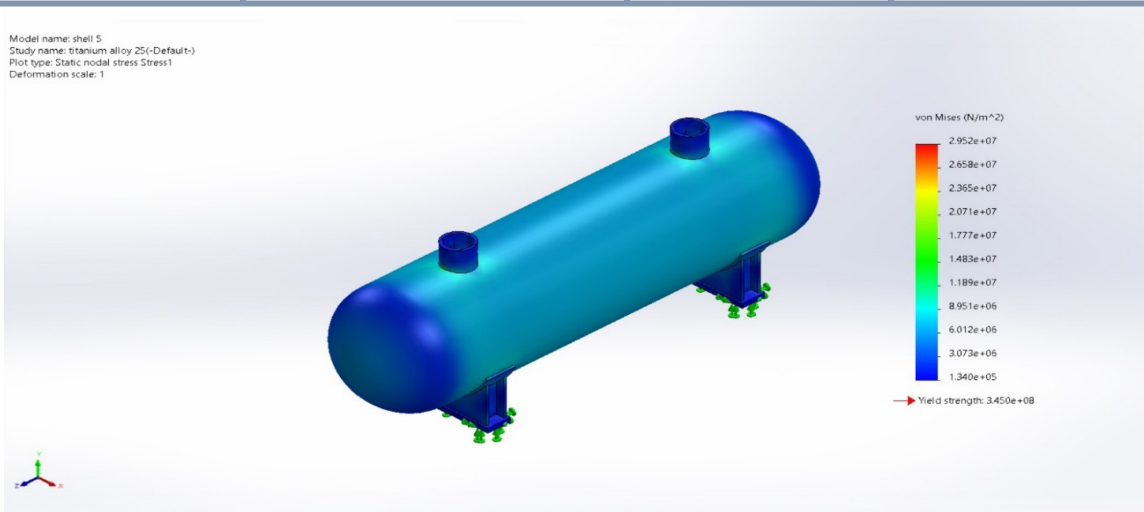


shell 4+-titanium alloy 20-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 25mm:

STRESS ANALYSIS:

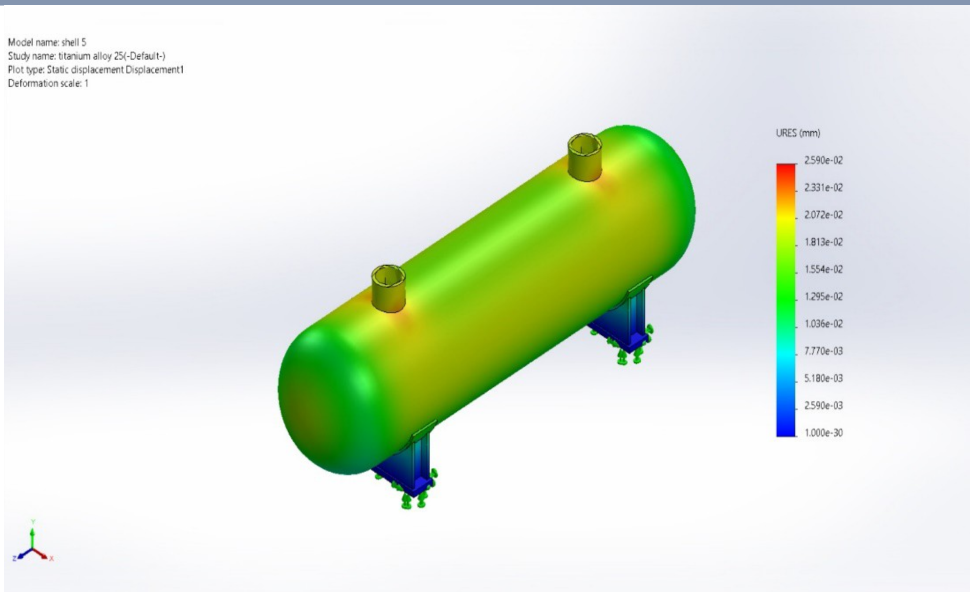
Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.340e+05N/m^2 Node: 6517	2.952e+07N/m^2 Node: 1532



shell 5-titanium alloy 25-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	2.590e-02mm Node: 3184

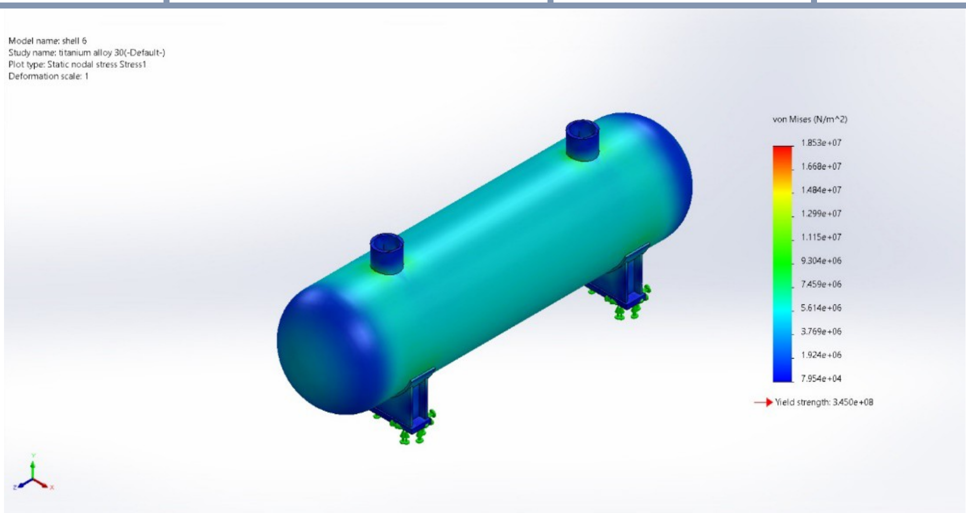


shell 5-titanium alloy 25-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 30mm:

STRESS ANALSIS:

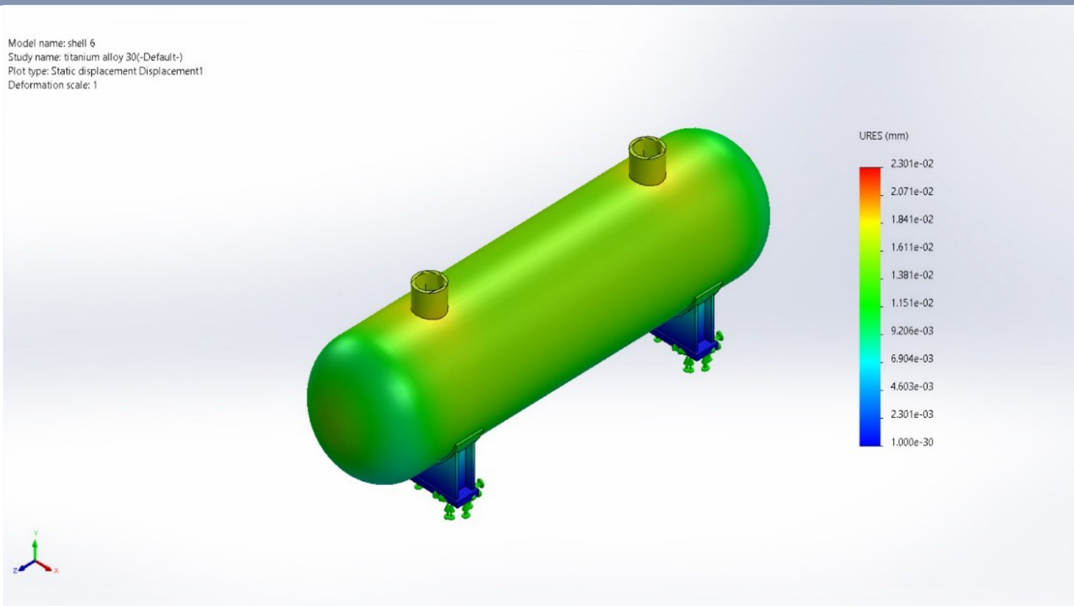
Name	Type	Min	Max
Stress1	VON: von Mises Stress	7.954e+04N/m^2 Node: 995	1.853e+07N/m^2 Node: 1532



shell 6-titanium alloy 30-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	2.301e-02mm Node: 3179

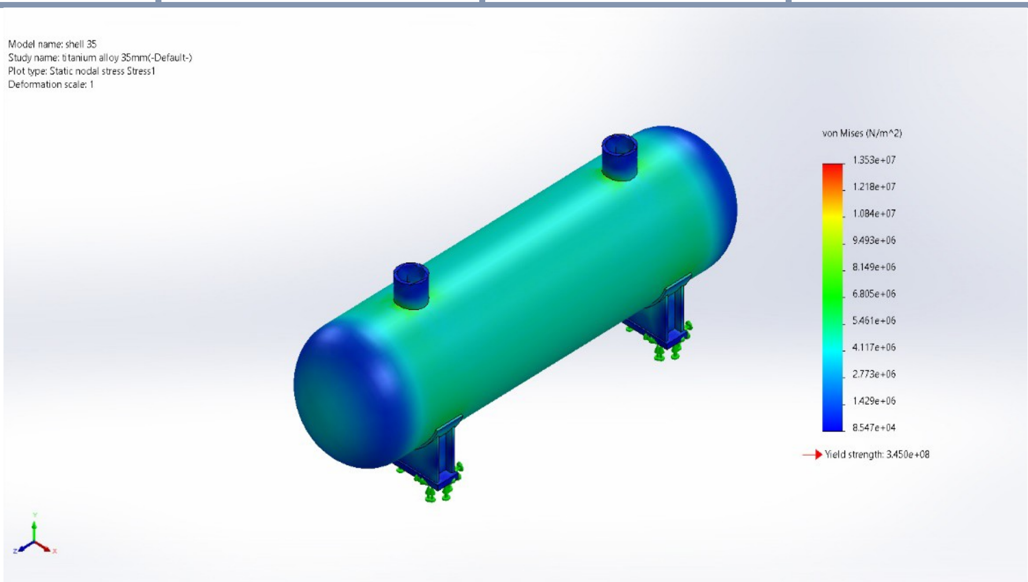


shell 6-titanium alloy 30-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 35mm

STRESS ANALYSIS:

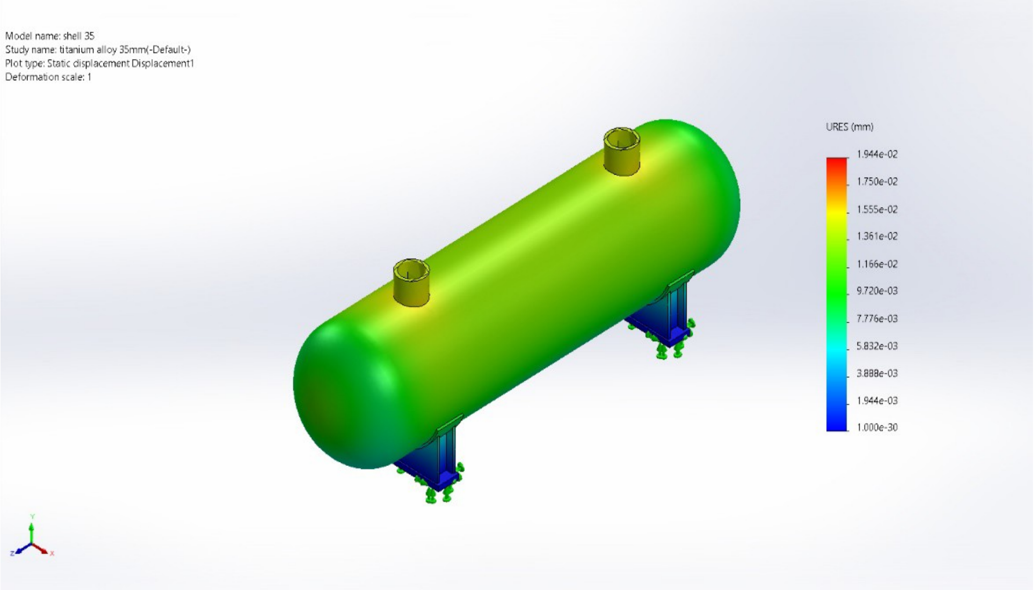
Name	Type	Min	Max
Stress1	VON: von Mises Stress	8.547e+04N/m^2 Node: 6426	1.353e+07N/m^2 Node: 1527



shell 35-titanium alloy 35mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.944e-02mm Node: 2441

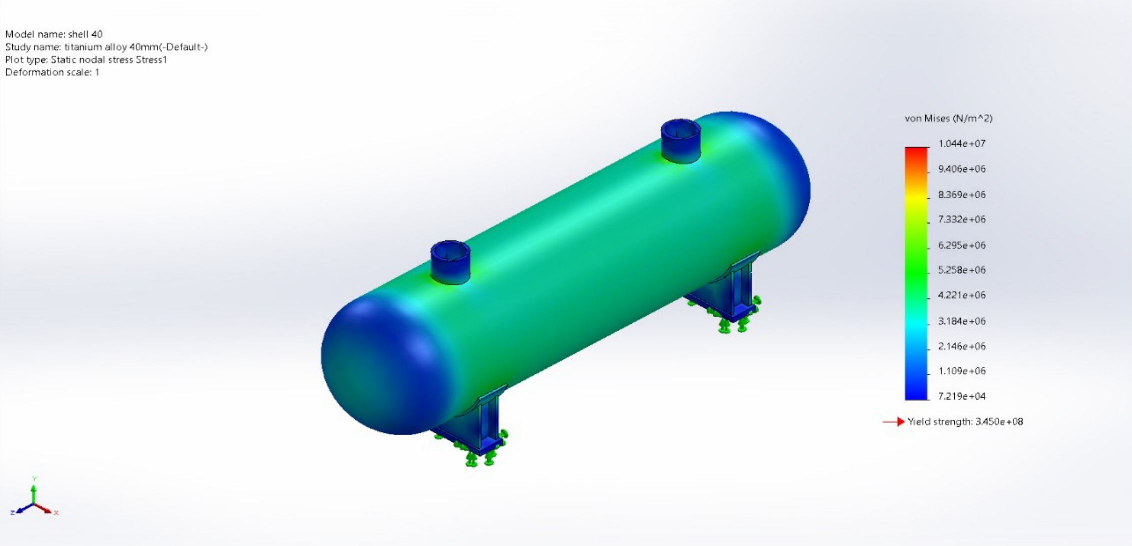


shell 35-titanium alloy 35mm-Displacement-Displacement1

PRESSURE VESSEL OF THICKNESS 40mm:

STRESS ANALYSIS:

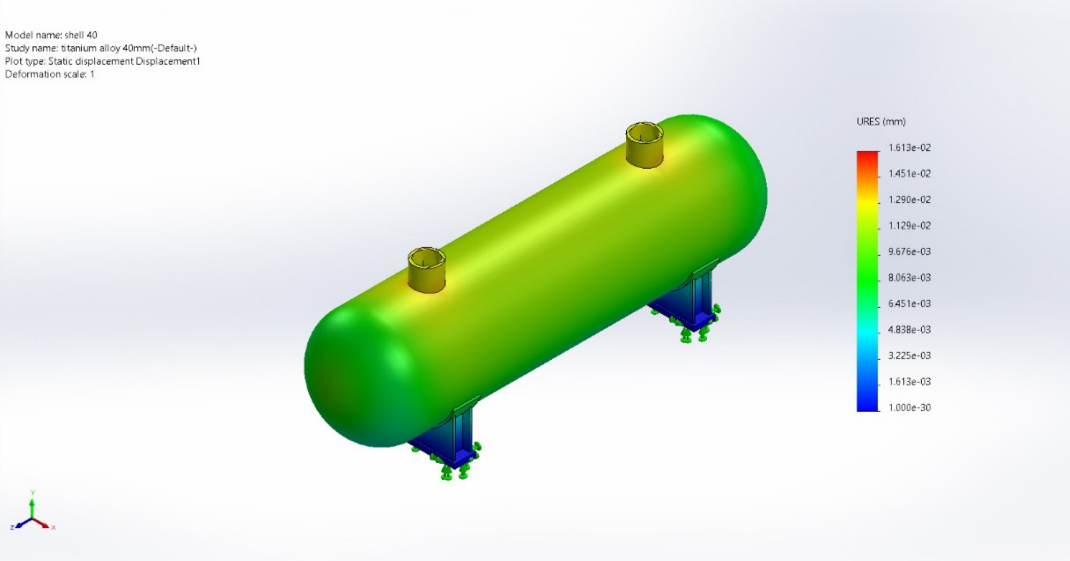
Name	Type	Min	Max
Stress1	VON: von Mises Stress	7.219e+04N/m^2 Node: 8209	1.044e+07N/m^2 Node: 1532



shell 40-titanium alloy 40mm-Stress-Stress1

DISPLACEMENT:

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0.000e+00mm Node: 1533	1.613e-02mm Node: 2429



shell 40-titanium alloy 40mm-Displacement-Displacement1

CHAPTER-7

RESULTS AND COMPARISIONS

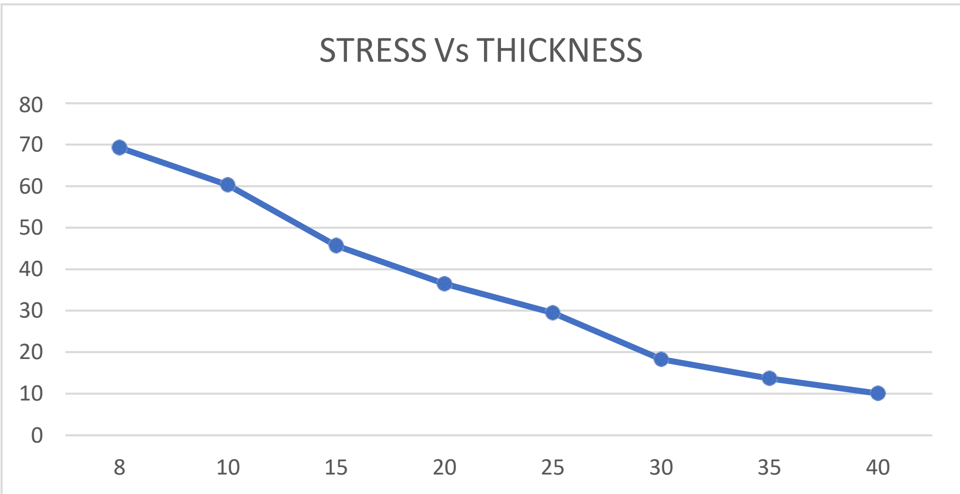
ALLOY STEEL:

Elastic module: 2.1 x 10^{11} N/m^2

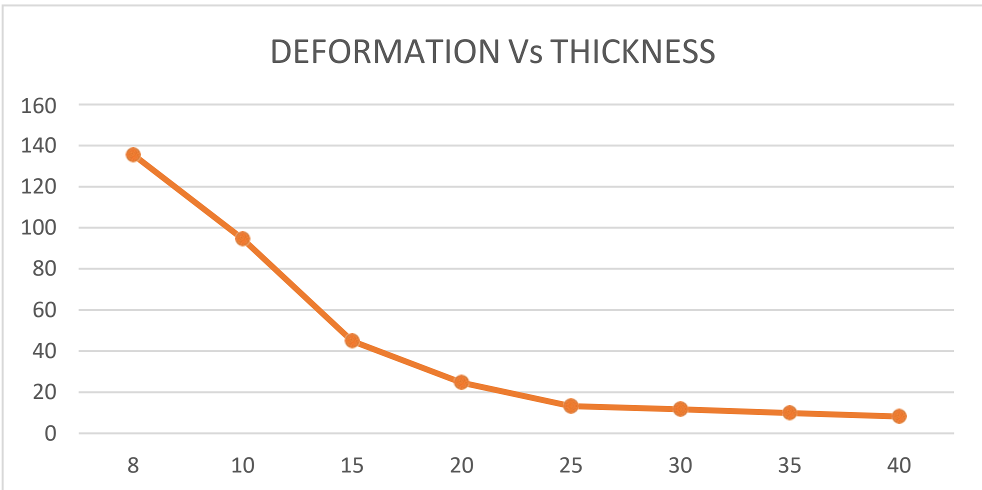
Poisson's ratio: 0.28

Yield strength: 620422000 N/m^2

S.no	Thickness (mm)	Stress (N/m^2)	Displacement (mm)
1	8	6.933 X 10^7	1.355 X 10^{-1}
2	10	6.036 X 10^7	9.458 X 10^{-2}
3	15	4.573 X 10^7	4.495 X 10^{-2}
4	20	3.657 X 10^7	2.471 X 10^{-2}
5	25	2.985 X 10^7	1.322 X 10^{-2}
6	30	1.832 X 10^7	1.170 X 10^{-2}
7	35	1.3147 X 10^7	9.863 X 10^{-3}
8	40	1.017 X 10^7	8.198 X 10^{-3}



- **THICKNESS ON X-AXIS (mm)**
- **STRESS ON Y-AXIS (MPa)**



- **THICKNESS ON Y-AXIS (mm)**
- **DEFORMATION ON X-AXIS (micrometer)**

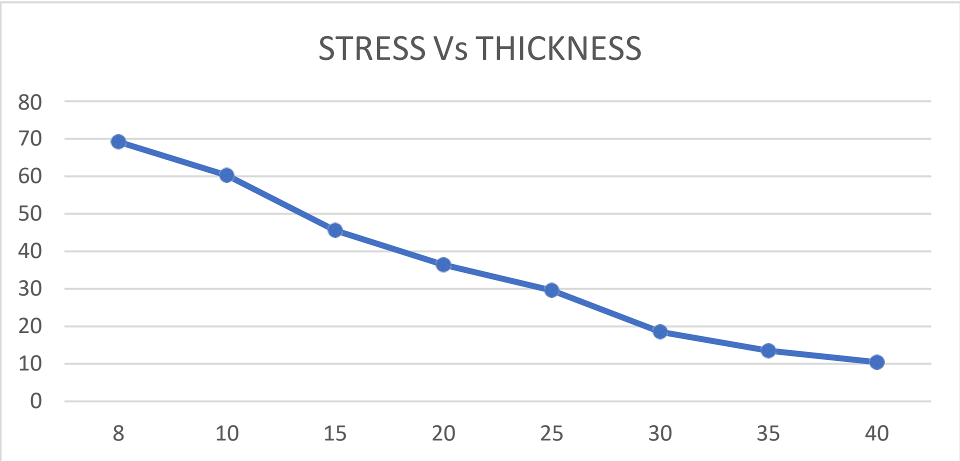
CAST CARBON STEEL:

Elastic module: 2 x 10^{11} N/m^2

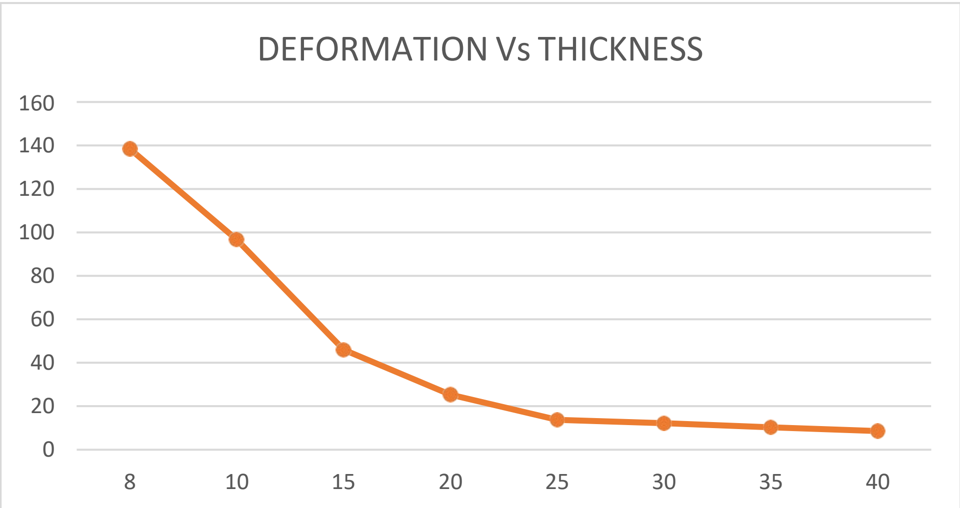
Poisson's ratio: 0.32

Yield strength: 248168000 N/m^2

S. No	THICKNESS (mm)	STRESS (N/m^2)	DISPLACEMENT (mm)
1	8	6.915 X 10^7	1.383 X 10^{-1}
2	10	6.019 X 10^7	9.652 X 10^{-2}
3	15	4.555 X 10^7	4.589 X 10^{-2}
4	20	3.634 X 10^7	2.521 X 10^{-2}
5	25	2.954 X 10^7	1.366 X 10^{-2}
6	30	1.849 X 10^7	1.212 X 10^{-2}
7	35	1.345 X 10^7	1.024 X 10^{-3}
8	40	1.039 X 10^7	80494 X 10^{-3}



- THICKNESS ON X-AXIS (mm)
- STRESS ON Y-AXIS (MPa)



- THICKNESS ON Y-AXIS (mm)
- DEFORMATION ON X-AXIS (micrometer)

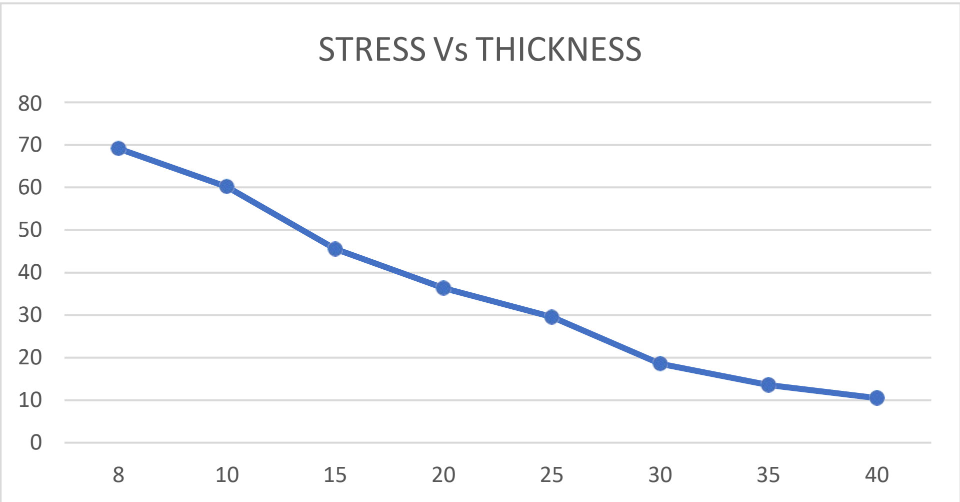
ALLUMINIUM ALLOY:

Elastic module: 6.9 x 10^{10} N/m^2

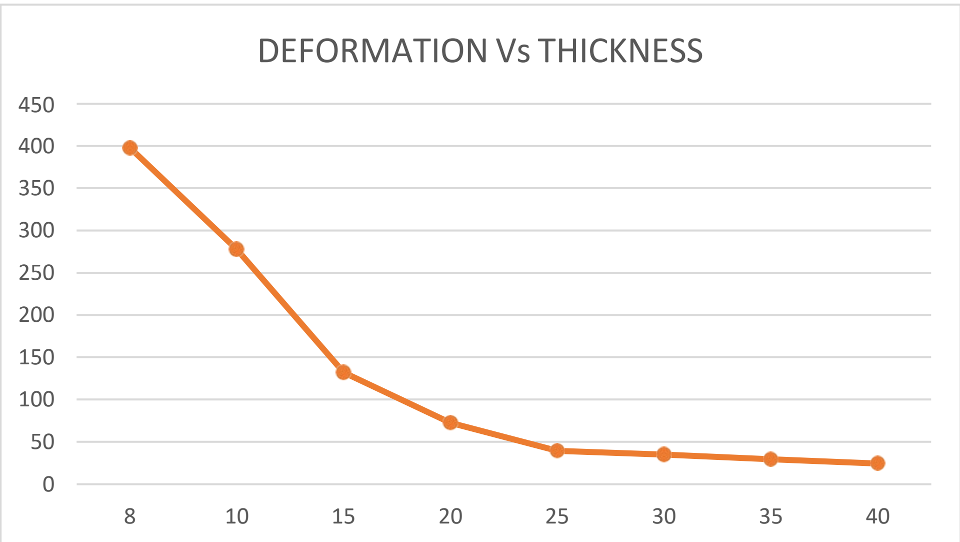
Poisson's ratio: 0.33

Yield strength: 27574200 N/m^2

S. No	THICKNESS (mm)	STRESS (N/m^2)	DISPLACEMENT (mm)
1	8	6.910 X 10^7	3.978 X 10^{-1}
2	10	6.014 X 10^7	2.777 X 10^{-1}
3	15	4.551 X 10^7	1.320 X 10^{-1}
4	20	3.628 X 10^7	7.253 X 10^{-2}
5	25	2.951 X 10^7	3.941 X 10^{-2}
6	30	1.853 X 10^7	3.502 X 10^{-2}
7	35	1.353 X 10^7	2.958 X 10^{-2}
8	40	1.044 X 10^7	2.454 X 10^{-2}



- **THICKNESS ON X-AXIS (mm)**
- **STRESS ON Y-AXIS (MPa)**



- **THICKNESS ON Y-AXIS (mm)**
- **DEFORMATION ON X-AXIS (micrometer)**

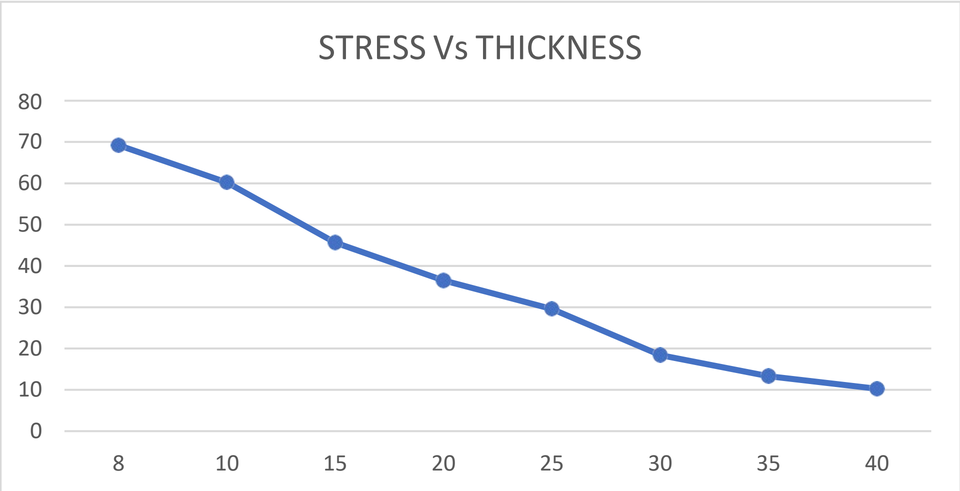
<u>COPPER ALLOY:</u> (Beryllium copper, UNS C173000)

Elastic module: 1.25×10^{11} N/m^2

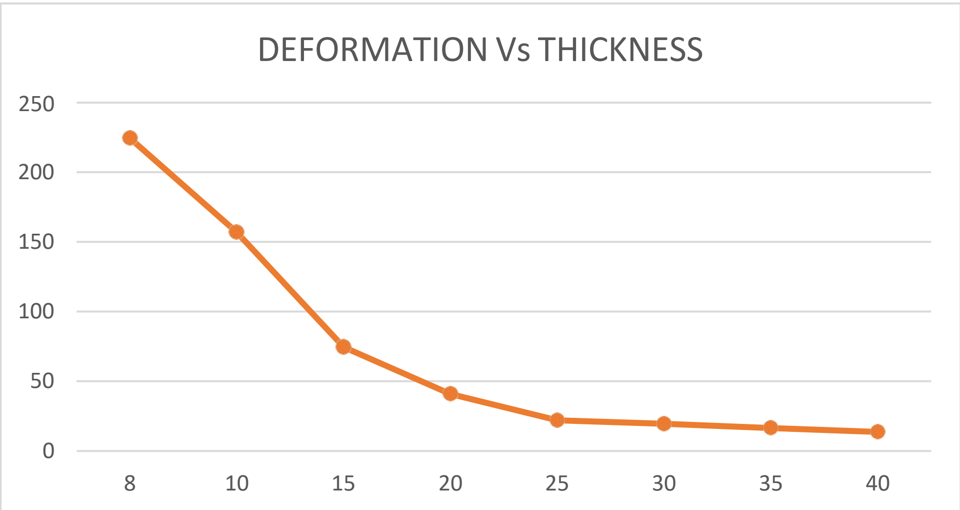
Poisson's ratio: 0.3

Yield strength: 172000000 N/m^2

S. No	THICKNESS (mm)	STRESS (N/m^2)	DISPLACEMENT (mm)
1	8	6.924×10^7	2.244×10^{-1}
2	10	6.023×10^7	1.569×10^{-1}
3	15	4.564×10^7	7.449×10^{-2}
4	20	3.646×10^7	4.093×10^{-2}
5	25	2.656×10^7	2.203×10^{-2}
6	30	1.840×10^7	1.952×10^{-2}
7	35	1.330×10^7	1.647×10^{-2}
8	40	1.028×10^7	1.368×10^{-2}



- **THICKNESS ON X-AXIS (mm)**
- **STRESS ON Y-AXIS (MPa)**



- **THICKNESS ON Y-AXIS (mm)**
- **DEFORMATION ON X-AXIS (micrometer)**

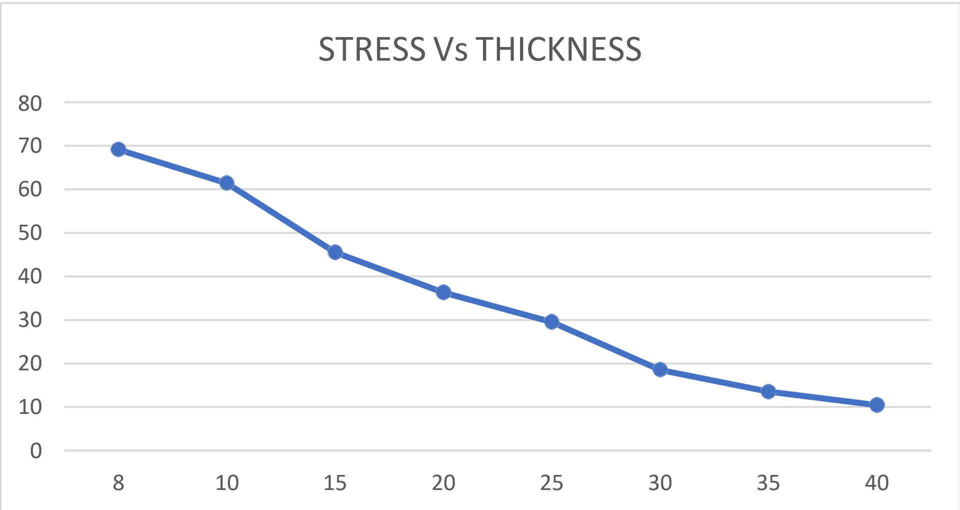
TITANIUM ALLOY:

Elastic module: 1.05×10^{11} N/m^2

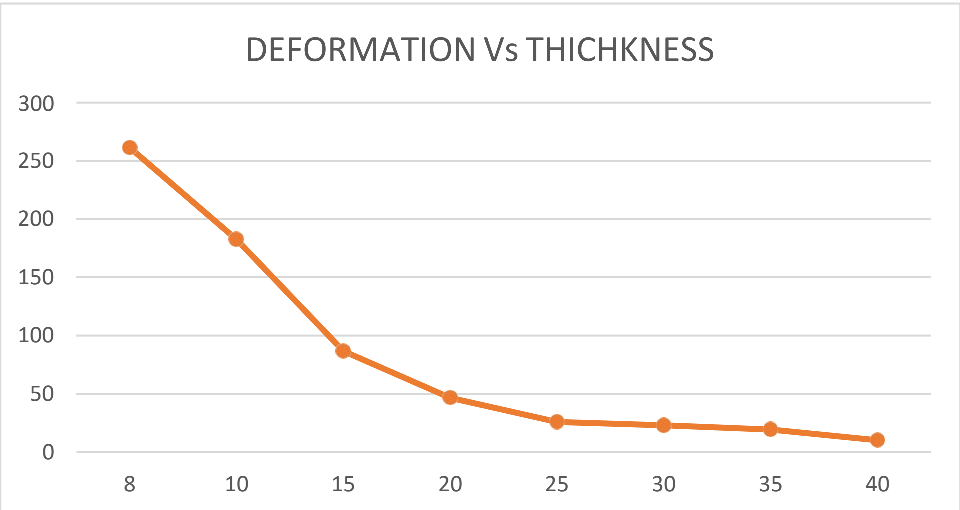
Poisson's ratio: 0.33

Yield strength: 345000000 N/m^2

S. No	THICKNESS (mm)	STRESS (N/m^2)	DISPLACEMENT (mm)
1	8	6.910×10^7	2.613×10^{-1}
2	10	6.140×10^7	1.825×10^{-1}
3	15	4.551×10^7	8.675×10^{-2}
4	20	3.628×10^7	4.767×10^{-2}
5	25	2.952×10^7	2.589×10^{-2}
6	30	1.852×10^7	2.301×10^{-2}
7	35	1.353×10^7	1.944×10^{-2}
8	40	1.044×10^7	1.930×10^{-2}



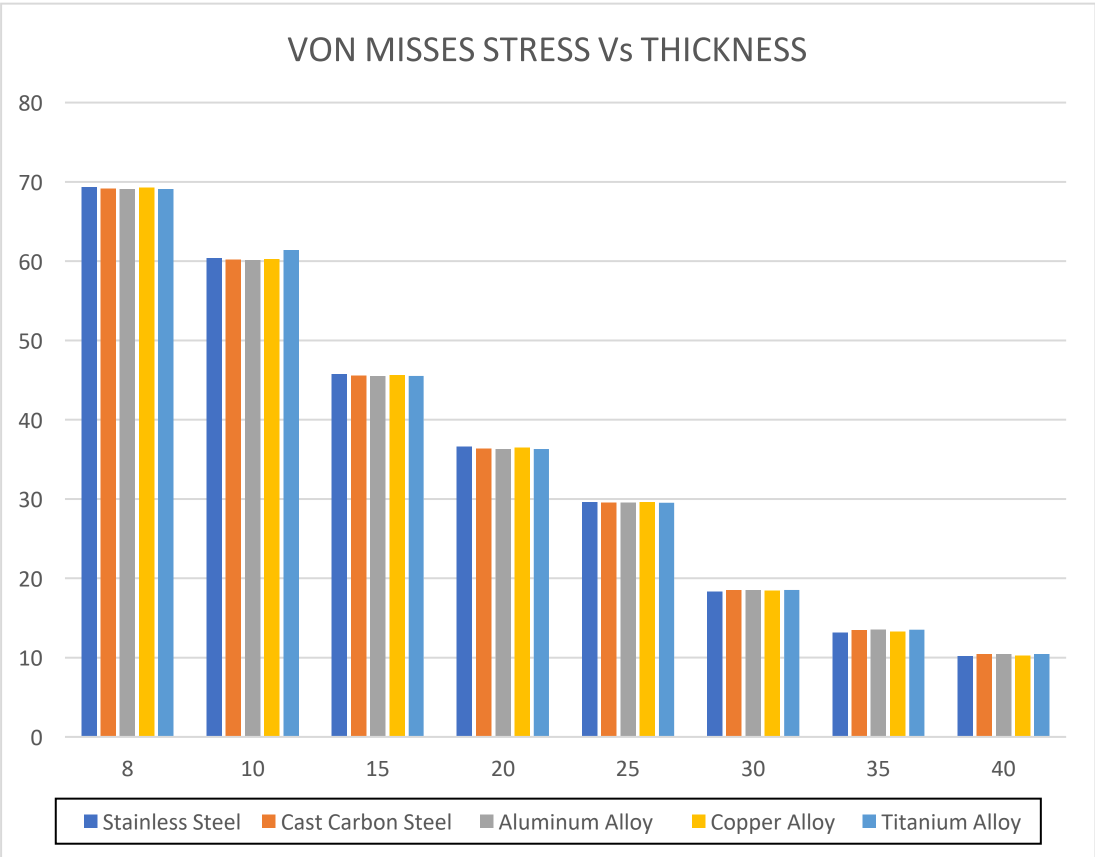
- **THICKNESS ON X-AXIS (mm)**
- **STRESS ON Y-AXIS (MPa)**



- **THICKNESS ON Y-AXIS (mm)**
- **DEFORMATION ON X-AXIS (micrometer)**

COMPARISION OF FIVE MATERIALS (VON MISSES'S STRESSES):

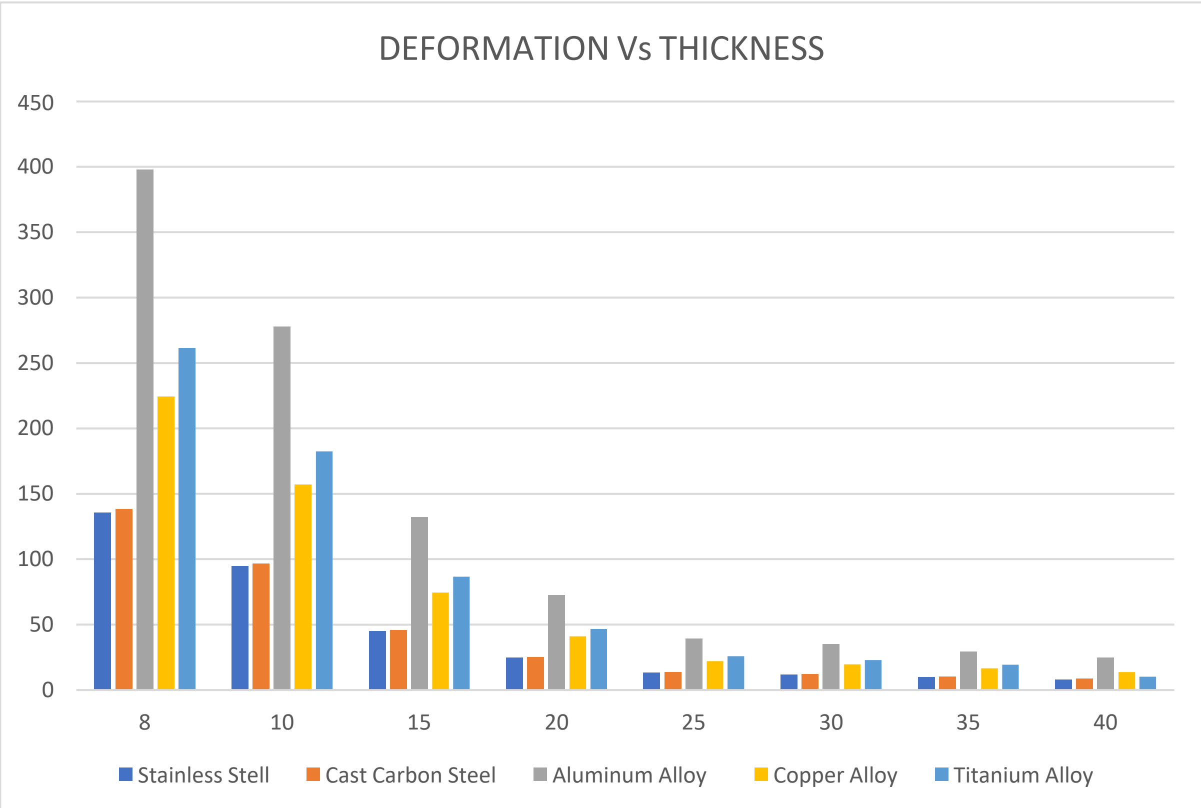
THICKNESS→ (mm)	8	10	15	20	25	30	35	40
MATERIALS	VON MISES STRESS(MPa)							
Alloy Steel	69.3	60.36	45.73	36.57	29.58	18.32	13.17	10.17
Cast Carbon Steel	69.15	60.19	45.55	36.34	29.54	18.49	13.45	10.39
Aluminum Alloy	69.1	60.14	45.51	36.28	29.52	18.53	13.53	10.44
Copper Alloy	69.24	60.23	45.64	36.46	29.56	18.4	13.3	10.28
Titanium Alloy	69.1	61.4	45.51	36.28	29.52	18.52	13.53	10.44



- **Thickness on x-axis (mm)**
- **Stress on Y-axis (MPa)**

COMPARISION OF FIVE MATERIALS (DEFORMATION)

THICKNESS→ (mm)	8	10	15	20	25	30	35	40
MATERIALS	VON MISES STRESS(MPa)							
Alloy Steel	135.5	94.58	44.95	24.71	13.22	11.7	9.863	8.198
Cast Carbon Steel	138.3	96.52	45.89	25.21	13.66	12.12	10.24	8.494
Aluminum Alloy	397.8	277.7	132	72.53	39.41	35.02	29.58	24.54
Copper Alloy	224.4	156.9	74.49	40.93	22.03	19.52	16.47	13.68
Titanium Alloy	261.3	182.5	86.75	46.7	25.89	23.01	19.44	10.3



- **Thickness on x-axis (mm)**
- **Deformation on Y-axis (micrometer)**

- ✓ We have observed the results of various materials at various thickness from the tables and graphs shown above
- ✓ From the tables and graphs we observed that the reduction in stress level is notable up to 30mm (>10%) from 30mm the reduction in stress level gets less notable (approx. 10%). This determines the optimum thickness for the given pressure vessel.
- ✓ We can also see that this principle follows for all the different materials in same way showing 30mm as the optimum thickness.
- ✓ By checking the deformation graphs and tables of different material we can see that the deformation rate of Alloy Steel and Cast carbon steel are much better than other materials and the deformation of aluminum and copper are high, this makes Alloy Steel the better material to be used.
- ✓ The deformation rate in ascending order:
  Alloy Steel < Cast Carbon Steel < Titanium Alloy < Copper Alloy < Aluminum Alloy.

CHAPTER-8

CONCLUSION

8.1 OBJECTIVE:

To determine the optimum thickness of a pressure vessel with suitable material required for prescribed working conditions. Comparative study for stress analysis has been made for cylindrical pressure vessel having the same volume by varying the thickness of the pressure vessel. Comparative study for stress analysis has been made for cylindrical pressure vessel having the same dimensions by varying the material used for the pressure vessel.

8.2 PROBLEM IDENTIFICATION:

The main intention behind this project is to determine stress level and deformation range on the walls of pressure vessel. If the stress values are large enough & cross the limitation of allowable stress values of material of vessel, we then check for the appropriate thickness of shell wall. Solving the model by FEM with Solid works simulation platform after every change in thickness of shell wall, we calculate the longitudinal, hoop stresses or Von-Mises stress over the shell & verify whether the stress values or deformation minimize with the increase in thickness of shell. Also, we study the effect of stresses on vessel walls by changing material used for cylindrical pressure vessel. By analyzing all the results from the study, we provide the optimum thickness of the pressure vessel with suitable material for the prescribed working conditions.

8.3 JUSTIFICATION OF RESEARCH WORK:

There are many reasons behind failure of pressure vessels. But the most prominent cause of failure is improper selection of materials of shells & door systems, inadequate thickness of shell & door mechanism, wrong estimation of pressure level & temperature range for safe working & ultimately incomplete conclusions about the stresses generate at different locations of vessel, faulty design of shape of vessel, welding problems, unsafe modifications or alteration. In this research work we have included these considerations & tried to solve these problems by standard methods of design prescribed by A.S.M.E., We have also used the D.B.A (Design by Analysis) method to justify our research work.

8.4 BENEFIT OCCURS FROM THIS PROJECT WORK:

The main benefit from this research work is that we can observe the behavior of pressure vessels under pressure constraints for different thickness of pressure vessel with different materials. We can also identify the prominent failure areas of the vessel & determine the stress and deformation on the walls of the pressure vessel. Thus, we can easily conclude the optimum thickness of the pressure vessel with suitable material. To avoid these defects and failure, we can make suitable modifications on vessel & thus optimize the design data. We have followed both the design procedure of ASME as well as the Design by Analysis method which increased the accuracy of design.

8.5 LIMITATIONS OF PROJECT WORK:

The main drawbacks in different FEA based research works are that we always have to compare the results from simulation with practical exposures & analytical results. The main reason behind this is that, the results of simulations & their accuracy totally depend on the right application of simulation tools & exact knowledge of the different parameters used to define & simulate practical conditions of the job. The result of simulation may change with wrong estimation and application and lack of knowledge, for the same observation.

8.6 CONCLUSION:

In this project work, Finite Element Analysis of a pressure vessel under pressure loading is investigated using simulation-based methods with Solid works software package. Here in stress plots, the Von-mises yield criterion has been used to determine the stress for different thickness. Here we observed that pressure loading & stress generated due to pressure loading have a significant role in the deformation of the pressure vessel. The stressed areas are also different for pressure loading depending on the material type used for the pressure vessel. When we compare the stress effect and deformation on a circular cross-section by varying the thickness and materials used in a pressure vessel, there is a significant difference in the behavior of stress and deformation in the pressure vessel. By comparing pressure loading, it is clearly visible that stress is decreasing with the increase of thickness of circular section pressure vessel. The percentage decrease in stress level is prominent to certain thickness beyond it the change in stress level is less prominent (approx. 10%), the breaking point is considered as optimum thickness of pressure vessel, and it is compared with different materials.

From the present project work, pressure vessel is designed as per A.S.M.E standards and according to its analysis is done.

• It was found that from the design calculation, the minimum thickness required for the shell and dished end are about 5 mm excluding the corrosion allowance of 3 mm.

• It was found that the pattern of the graphs for the different materials, that the stresses are different, and similarly for the thickness conditions. By comparing all the results for solid works, we conclude the optimum thickness of pressure vessels with suitable material.

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