Design and comparison study of different types of Pressure vessel heads.

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Abstract :

This research is all about selecting the right pressure vessel head for the job. In the industry, there's always a lot of confusion about choosing among performance, material costs, and safety. You can't just rely on basic manual calculations and hope for the best; due to cost-cutting, going for a cheaper head can lead to serious accidents and safety gaps.

Our goal is to bridge the gap between 'design by rule' and 'design by analysis' using tools like PV Elite and SolidWorks. We're mainly looking at four types: Hemispherical, Elliptical, Conical, and Torispherical heads. Using SA 516 Gr. 70 steel, we want to find the best MAWP-to-weight ratio at a design pressure of 0.70 MPa. We're following ASME Section VIII Division 1 standards and then double-checking everything with PV Elite and SolidWorks to make sure the results are solid.

Numerical analysis identifies that the most significant geometry is hemispherical, achieving a maximum allowable working pressure of 2.23 MPa and superior structural stability with an efficiency index of 0.00507 MPa/kg. Moreover, the torispherical head is not recommended because of a significant stress peak of 206.9 MPa and a forming strain of 9.652%. This strain should be within the 5% threshold as per UCS-79; it requires a post-forming heat treatment to ensure structural safety and prevent failure. Conversely, the hemispherical head has maximum safety and uniform stress distribution; however, its crown-and-petal construction results in high manufacturing and labour costs. The 2:1 Ellipsoidal Head stands out as the most efficient and well-balanced design across all parameters. It provides a 33% safety reserve and maintains a forming strain of 2.616%, which falls well within safe limits. This research establishes a comprehensive techno-economic framework for selecting the optimal head for industrial applications.

Keywords: ASME BPVC Section VIII Division 1, PV Elite, SolidWorks Simulation, SA-516 Grade 70, Pressure Vessel Heads, Stress Concentration, Maximum Allowable Working Pressure (MAWP), Structural Efficiency Index, Forming Strain (UCS-79), Finite Element Analysis (FEA).

1. Introduction :

A pressure vessel definition is a closed container designed to hold gases or liquids at a pressure significantly different from the ambient air pressure. Pressure vessels are a crucial part of modern industry. They are found almost everywhere, ranging from LPG cylinders and cryogenic storage to oil and gas refineries.

The head is a vital component of a pressure vessel, acting as the closure that transitions internal membrane stress from the shell to a sealed boundary. Designs have evolved from simple flat heads, which often failed due to bending, to sophisticated shapes like torispherical, conical, hemispherical, and elliptical heads.

While these shapes are widely used and guided by standards like the ASME BPVC, selecting and optimising the right head remains a significant challenge. Current industry practices often lack comprehensive techno-economic and structural comparisons. This leads to inefficient designs, higher manufacturing costs, and increased risks of failure due to stress concentration or fabrication limitations. Furthermore, the industry lacks sufficient experimental validation and performance data to ensure a truly reliable design approach. Therefore, a study was performed based on mechanical performance, manufacturability, cost analysis, and experimental validation to identify the most efficient pressure vessel head configuration.

2. Literature Survey:

Geometric Analysis. The geometry of heads plays a crucial role in pressure vessel performance. Hemispherical heads are the most significant in terms of stress distribution, whereas ellipsoidal is an optimal choice for cost, performance, and manufacturability in industry. Conical and torispherical heads are not as preferable because of localized stresses at the knuckles and junctions.

Code Compliance: Industries follow the ASME Boiler and Pressure Vessel Code (BPVC) Section VIII, Division 1 for pressure vessel design. To determine the required thickness of the shell and heads, calculations are based on the internal design pressure. This approach identifies the most effective wall thickness to ensure safety and stability in stress distribution.

Stress Distribution: Research indicates that stress distribution is not uniform due to joint discontinuities and junctions, such as nozzles. Additionally, finite element studies confirm that heads typically experience lower stress levels. These studies also highlight that geometric transitions play a critical role in stress concentration and overall structural behavior.

Computational methods: integrating Analytical and Finite Element Analysis (FEA) tools like P V Elite software and SolidWorks, provide precise information about stress distribution, deformation, and failure under actual operating conditions.

Regarding design approaches: ASME Section VIII, Division 1 follows a Design-by-Rule approach, while ASME Division 2 follows a Design-by-Analysis approach using FEA and advanced failure theories like the von Mises criterion.

Material selection: SA-516 Grade 70 carbon steel is commonly used for pressure vessel construction due to its high strength, good weldability, and resistance to fracture. Proper material selection plays a crucial role in the long-term performance of the system.

Research Objective: This research uses an Efficiency Index (MPa/kg) to evaluate all heads under an identical medium pressure of 7.0 MPa (70 bar) to determine a more material-effective geometry.

3. Methodology

Research Framework

Objective of the Study

The primary objective of this research is to identify the optimal geometric head design for pressure vessels by evaluating structural integrity and operational efficiency under a pressure of 10 MPa.

Types of Pressure Vessel Heads Evaluated:

- Hemispherical Head
- 2:1 Ellipsoidal Head
- Torispherical Head
- Conical Head

Overall Approach

This study utilises a multidisciplinary framework to determine the most effective pressure vessel head design.

I. Analytical Verification

Perform calculations for thickness, MAWP (Maximum Allowable Working Pressure), weight, and MAP (Maximum Allowable External Working Pressure) using the ASME Section VIII, Division 1 design code. These results will then be validated and analysed using PV Elite software.

II. Numerical Analysis

Using SolidWorks Simulation software, numerical analysis will be performed to evaluate:

Von Mises Stress Distribution: To identify stress concentrations across different geometries.

Resultant Displacement: To measure the total deformation under pressure.

III. Manufacturing Assessment

Evaluate the manufacturing process for each head type based on fabrication feasibility. This includes calculating forming strain and ensuring compliance with UCS-79 and UCS-56 standards.

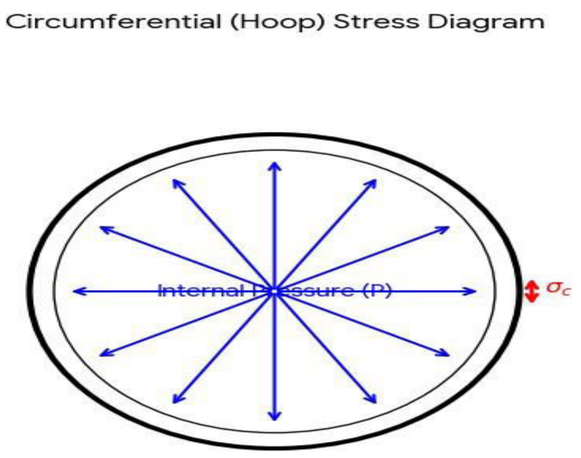
IV. Economic Evaluation

Conduct a cost-benefit analysis based on the weight-to-strength ratio, material costs, and the overall suitability of each head for the project

Table 1: Design Input Data and Material Properties

Parameter	Value (SI Units)
Material Specification	SA-516 Grade 70 Carbon Steel
Internal Design Pressure (P)	0.70 MPa (7.1379 kg/cm^2)
Design Temperature (T)	363.15 K 90^C
Inside Diameter (D)	1676.4 mm
Allowable Material Stress (S)	115.15 MPa(1174.15 kg/cm^2)
Joint Efficiency (E)	0.85 (Spot Radiographed)
Corrosion Allowance (CA)	2.0 mm(10 – year equipment life span)
Material Density (ρ)	7850 kg/m^3
Regulatory Code	ASME BPVC Section VIII, Div. 1 (2023)
Design Methodology	Design-by-Rule (DBR)

I. Analytical Verification using ASME (BPVC) CODES:



Cylindrical Shell (ASME UG-27) CIRCUMFERENTIAL stress calculations:

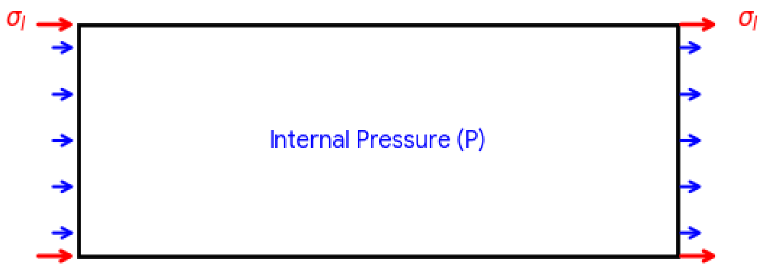
Cylinder shell thickness:

$$T = \frac{PR}{SE - 0.6P} + CA$$

$$= \frac{0.70 * 838.2}{115.15 * 0.85 - 0.6 * 0.70} + 2.0$$

$$= 8.02 \text{ mm}$$

Longitudinal Stress Diagram



Cylindrical Shell Longitudinal stress calculations(ASME UG-27):

Cylinder shell thickness

$$T = \frac{PR}{2SE + 0.4P} + CA$$

$$= \frac{0.70 * 838.2}{2 * 115.15 * 0.85 - 0.4 * 0.70} + 2.0$$

$$= 5.001 \text{ mm}$$

So, T nominal will be taken as 10 mm

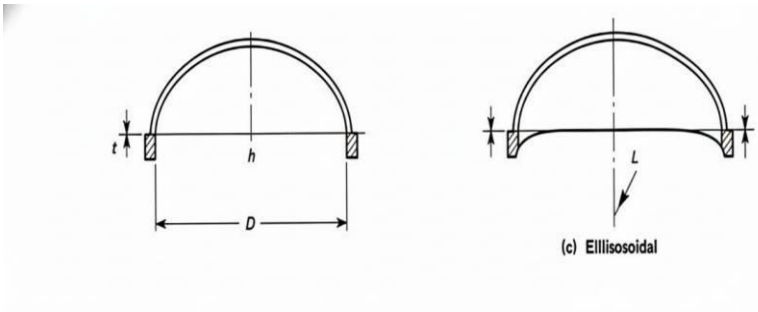
Calculating maximum allowable working pressure(mawp):($\boldsymbol{t = t_{nom} - CA}$)

$$P = \frac{SEt}{R + 0.6 * t}$$

$$= \frac{115.15 * 0.85 * 8}{838.2 + 0.6 * 8}$$

$$= 0.928 \text{ MPa}$$

Thickness calculation for ellipsoidal head(ASME UG-32d):



Thickness Calculation formula:

$$T = \frac{PD}{2SE - 0.2P} + CA$$

Where:

- P = Internal design pressure (MPa)
- R = Internal radius (mm)
- S = Allowable stress (MPa)
- E = Joint efficiency
- CA = Corrosion allowance (mm)

Substitute values

$$= \frac{0.70 * 1676.4}{2 * 115.15 * 0.85 - 0.2 * 0.70} + 2.0$$

Final answer

= 7.998 mm

So, T nominal will be taken as 10mm

Weight of ellipsoidal head:

$$AREA = 1.084 * D\hat{}2$$

$$= 1.084 * (1.6764)\hat{}2$$

$$= 3.046 m\hat{}2$$

$$Volume = A * T = 3.0463 * 0.01 = 0.0304\ m\hat{}3$$

$$Weight = V * \rho$$

$$= 0.0304 * 7850 = 238.64 \text{ Kg}$$

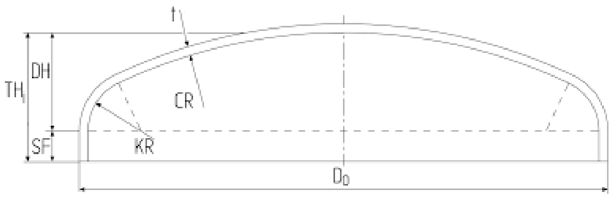
MAWP:$(\boldsymbol{t} = \boldsymbol{t_{nom}} - \boldsymbol{CA})$

$$p\ (mawp) = \frac{2SEt}{D\ + 0.2t}$$

$$= \frac{2 * 115.15 * 0.85 * 8}{1676.4 + 0.2 * 8}$$

$$= 0.933 \text{ MPa}$$

Calculating thickness for Torispherical head (ASME UG-32e):



$$T = \frac{0.885PL}{SE - 0.1P} + Ca$$

Where:

- P = Internal design pressure
- L = Crown radius (≈ D)
- S = Allowable stress
- E = Joint efficiency
- CA = Corrosion allowance

Substitution

$$= \frac{0.885 * 0.70 * 1676.4}{115.15 * 0.85 - 0.1 * 0.70} + 2.0$$

Final answer

$$= 12.618 \text{ mm}$$

So, T nominal to be taken as 14 MM

Weight calculations:

$$Area = 1.08 * \frac{3.14 * D^2}{4}$$

$$= 1.08 * \frac{3.14*1.676^2}{4} = 2.381\ m^2$$

$$Volume = A * t$$

$$= 2.381 * 0.014 = 0.0333m^3$$

$$Weight = V * \rho$$

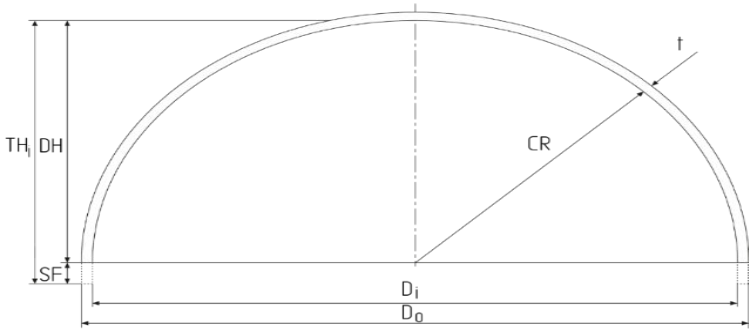
$$= 0.0300 * 7850$$
$$= 261.40\ kg$$

MAWP: $(\boldsymbol{t = t_{nom} - CA})$

$$p(mawp) = \frac{SEt}{0.885L + 0.1t}$$

$$= \frac{115.15 * 0.85 * 12}{0.885 * 1676.4 + 0.1 * 12} = 0.791\ MPa$$

Thickness calculations for hemispherical head (ASME UG-32f):



Thickness Calculation

$$T = \frac{PL}{2SE - 0.2P} + Ca$$

Where:

- P= Internal design pressure (MPa)
- R = Internal radius (mm)
- S = Allowable stress (MPa)
- E = Joint efficiency
- CA = Corrosion allowance (mm)

Substitution:

$$= \frac{0.70 * 838.2}{2 * 115.15 * 0.85 - 0.2 * 0.70} + 2.0$$

Final answer

= 4.99 mm

So, T nominal to be taken as 8 mm

Weight of Hemispherical Head:

$$Area = 2 * 3.14 * R^2$$

$$= 2 * 3.14 * 0.8382^2$$

=4.412m^2

$$Volume = A * T$$

$$= 4.412 * 0.008$$

$$= 0.0352\ m\text{^}3$$

Weight=V*ρ=$0.0352 * 7850$=$276.32\ kg$

MAWP: $(\boldsymbol{t = t_{nom} - CA})$

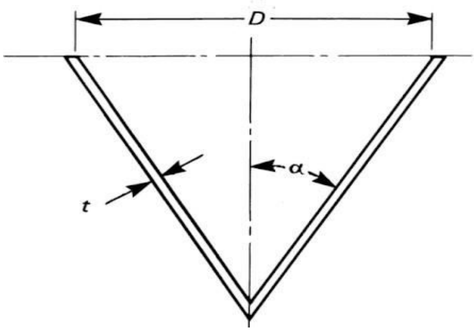
$$p(mawp) = \frac{2 * SEt}{L - 0.2t}$$

$$= \frac{2 * 115.15 * 0.85 * 6}{838.2 - 0.2 * 6}$$

=1.40 MPa

Thickness calculation for conical head(ASME UG-32):

A 30° half-apex angle is selected because it is the maximum limit allowed by **ASME Section VIII, Div. 1, Mandatory Appendix 1-5** to avoid the high fabrication cost of a transition knuckle. This specific angle effectively balances material economy with the suppression of destructive discontinuity stresses at the shell-to-cone junction.



Thickness Calculation

$$T = \frac{PD}{2\,\mathrm{SE.}\cos\propto -0.6P} + Ca$$

Where:

- P - Internal Design Pressure
- D - Internal Diameter of Vessel
- S - Allowable Stress of Material
- E - Joint Efficiency
- α - Half Apex Angle of Cone
- CA - Corrosion Allowance

Substitution:

$$= \frac{0.70 * 1676.4}{2cos(30) * 115.15 * 0.85 - 0.6 * 0.70} + 2.0$$

Final answer

= 8.93 mm

So, T nominal to be taken as 12 mm

Weight calculations:

$$h = \frac{R}{\tan\alpha} = \frac{0.8382}{\tan 30} = 1.451\mathrm{m}$$

slant height:

$$l = \sqrt{(R^2 + h^2)} = \sqrt{(0.8382^2 + 1.451^2)} = 1.6764m$$

$$\text{Area} = \pi\mathrm{RL} = 3.14 * 0.8382 * 1.676 = 4.41\mathrm{m}^2$$

$$\text{volume} = \mathrm{A} * \mathrm{T} = 4.41 * 0.012 = 0.0529\ \mathrm{m}^3$$

$$\text{weight} = \mathrm{V} * \rho = 0.0529 * 7850 = 415.27\mathrm{kg}$$

MAWP: $(\boldsymbol{t} = \boldsymbol{t_{nom}} - \boldsymbol{CA})$

$$P(mawp) = \frac{2SEtcos\propto}{D + 1.2tcos\propto}$$

$$= \frac{2 * 115.15 * 0.85 * 10 * cos(30)}{1676.4 + 1.2 * 10 * cos(30)} = 1.005MPa$$

The step-by-step calculation for the Maximum Allowable External Working Pressure (MAWP) for each head type:

Hemispherical Head Calculation (ASME UG-33(c))

Hemispherical heads are calculated as spherical shells.

- Outside Radius (Ro): 1696.4 mm/2=848.2 mm
- Nominal Thickness (t): 8 mm
- Factor A:

$$A = \frac{0.125}{R0/T} = \frac{0.125}{848.2/8} = 0.00117$$

- Factor B: From the ASME Section II-D material chart for SA-516 Gr. 70
- Factor B≈82.7 MPa (12,000 psi)
- External MAWP (Pa):

$$P_a = \frac{B}{R0/T} = \frac{82.7}{106.0} = 0.78MPa = 7.95Kg/cm^2$$

2:1 Ellipsoidal Head Calculation (ASME UG-33(d)):

Ellipsoidal heads are treated as spherical shells with an equivalent radius (Re).

- Equivalent Radius (Re): 0.90×Do=0.90×1696.4=1526.76 mm
- Nominal Thickness (t): 10 mm
- Factor **A:**

$$A = \frac{0.125}{R0/T} = \frac{0.125}{1526.76/10} = 0.000818$$

- Factor B: From material charts, Factor B≈62.1 MPa (9,000 psi).
- External MAWP (Pa):

$$P_a = \frac{B}{R0/T} = \frac{62.1}{1526.7/10} = 0.406\ MPa = 4.14Kg/cm\text{^}2$$

Tori spherical Head Calculation (ASME UG-33(e)):

Torispherical heads use an equivalent sphere radius equal to the outside crown radius (Lo).

- Outside Crown Radius (Lo): 1676.4 mm (Assuming L=D) + 14 mm=1690.4 mm
- Nominal Thickness (t): 14 mm
- Factor A:

$$A = \frac{0.125}{R0/T} = \frac{0.125}{1690.4/14} = 0.00103$$

- Factor B: Factor B≈75.8 MPa (11,000 psi).
- External MAWP (Pa):

$$P_a = \frac{B}{R0/T} = \frac{75.8}{1690.4/14} = 0.627MPa = 6.39Kg/cm^2$$

Conical Head Calculation (ASME UG-33(f)):

- Effective Thickness (te): t·cos(α)=12 mm cos(30∘)=10.39 mm.
- Effective Length (Le): Based on your paper's geometry, the axial length (h) is 1451 mm. The effective length for buckling is Le=(h/2)·(1+Dso/Dlo).

Step 1: Calculate L/Do and Do/t ratios

- L/Do≈1451/1696.4=0.855.
- Do/t=1696.4/10.39=163.27

Step 2: Determine Factor A: Using the ASME Section II, Part D geometric chart:

For the calculated ratios, Factor A:

$$A = \frac{0.125}{R0/T} = \frac{0.125}{850.2/10.392} = 0.00076$$

Step 3: Determine Factor B From the material property chart for SA-516 Gr. 70 at 90∘C:

- Using Factor A, Factor B≈51.7 MPa (7,500 psi).

Step 4: Calculate Allowable External Pressure (Pa)

$$P_a = \frac{4B}{3(D_0/T_e)} = \frac{4 * 51.7}{3 * 163.27} = 0.422MPa = 4.30\ kg/cm^2$$

II. Numerical analysis:

P V Elite Analysis: Comparative Study of Head Types Based on Industrial Standards

1. Geometric Input: Each head type, Hemispherical, Elliptical, Torispherical, and Conical, is modelled with an outside diameter (OD) of 1676.4 mm.

2. Material Selection: SA-516 Gr. 70 carbon steel is assigned to all components.

3. Environmental Factors: A corrosion allowance of 2 mm is included in the design.

4. Load Application: An internal design pressure of 0.7 MPa is applied, with a joint efficiency of 0.85.

P V Elite analysis pictures

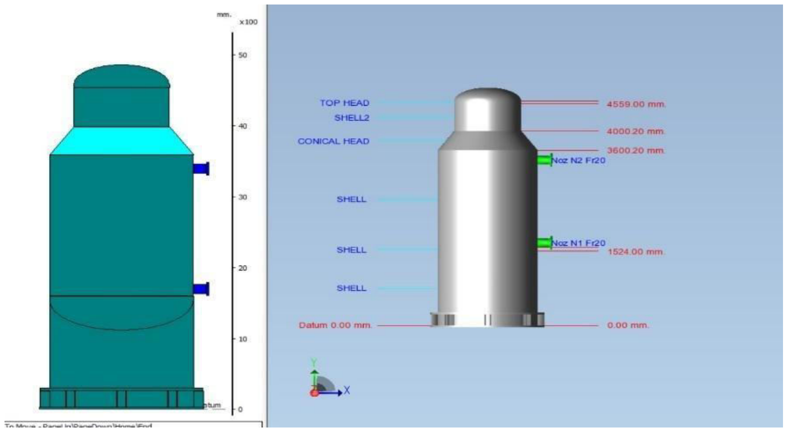


Fig 1. Conical head design in PV Elite

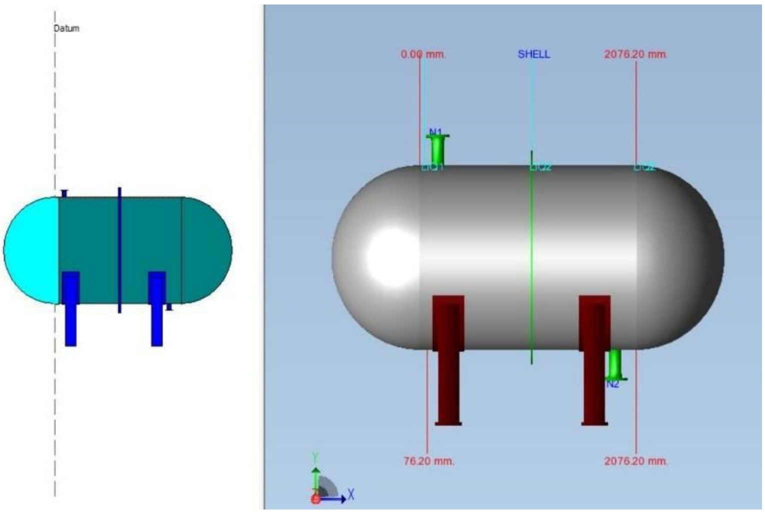


Fig 2. Hemispherical head design in PV Elite

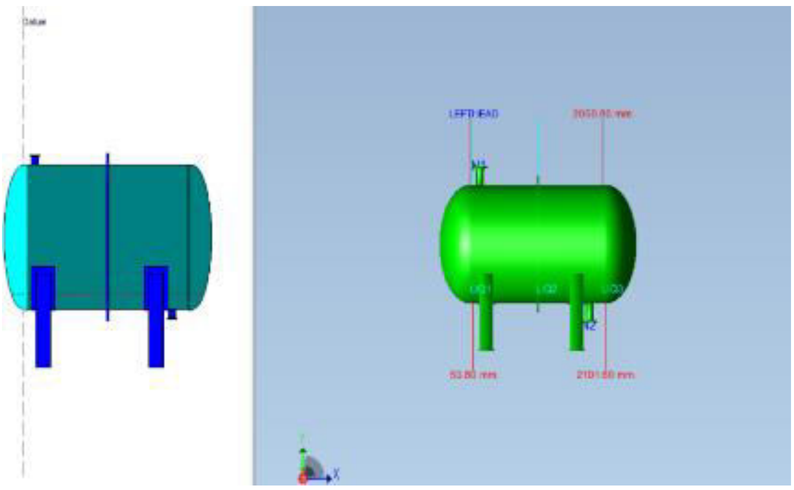


Fig 3. Torispherical head design in PV Elite

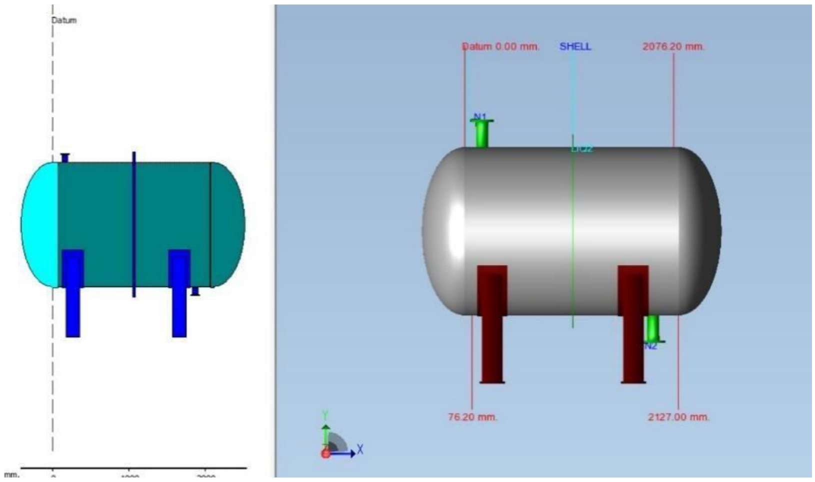


Fig 3. Ellipsoidal head design in PV Elite

Code-Bond Evaluation

ASME Compliance:

The required thickness for each head is determined according to ASME Section VIII, Division1.

Buckling Analysis: The software analyses the head based on an external pressure of 0.103 MPa to ensure protection against vacuum-induced collapse.

Hydrostatic Simulation: A virtual hydrotest is performed on each head (typically at 1.3 to 1.5 times the design pressure) to evaluate the design for pressure-induced stresses and strains.

$$P_{test} = 1.3 \times P_{design} \times \frac{S_t}{S_d}$$

P_{test} = Hydrostatic test pressure

P_{design}= Design pressure (MAWP)

S_t = Allowable stress at test temperature

S_d= Allowable stress at design temperature

Forming Check: The percentage of elongation for the head is measured per UCS-79 to ensure the material remains within safe limits after cold forming.

Output values

MAWP (Maximum Allowable Working Pressure): This is the highest internal pressure at which a vessel is designed to operate safely at a specific temperature. It is calculated based on the vessel's weakest component.

MAEWP (Maximum Allowable External Working Pressure): This is the highest external pressure at which a vessel is designed to operate safely at a specific temperature, considering its weakest point or minimum thickness.

Required vs. Nominal Thickness: This comparison determines whether the specified thickness of the material is sufficient to handle the design pressure, indicating if the vessel is safe to operate.

Analysis Using SolidWorks Simulation

This analysis investigates stress distributions and deformations across various head geometries under realistic loading conditions. The process involved the following steps:

3D Modelling: A 3D model was generated for each head type.

Material Assignment: SA-516 Gr. 70 steel was assigned to the models, featuring a yield strength of 239 MPa.

Physical Meshing and Boundary Conditions:

Fixtures: Fixed geometry constraints were applied to secure the models for load analysis.

Load Application: An internal pressure of 0.7 MPa was applied normal to the internal faces of each head.

Discretisation: The models were discretised using a fine finite element mesh to ensure high numerical accuracy in the results. Analysis Using SolidWorks Simulation

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Stress and Deformation Analysis

Von Mises Stress: The simulation identifies the peak stress region of each head using color mapping.

Displacement Mapping: The displacement of the head was calculated to analyze the stiffness of the head.

Factor of Safety (FoS): The FoS of each head is calculated by comparing the induced stress to the yield limit, confirming that all designs are safe.

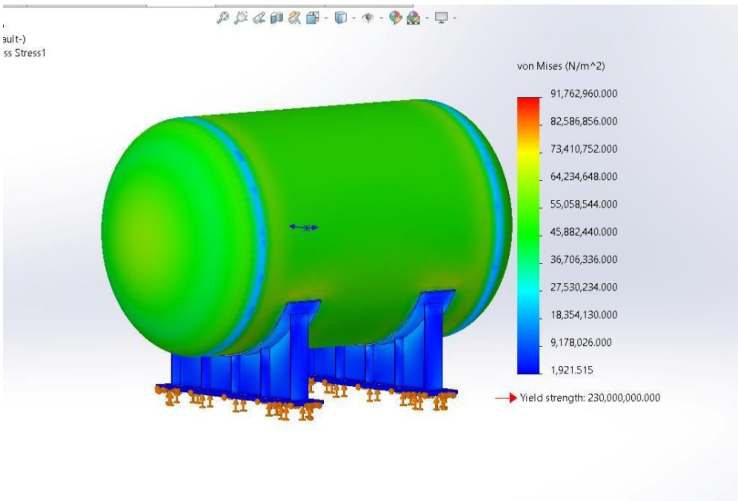


Fig 5. Elliptical head stress distribution analysis in SolidWorks

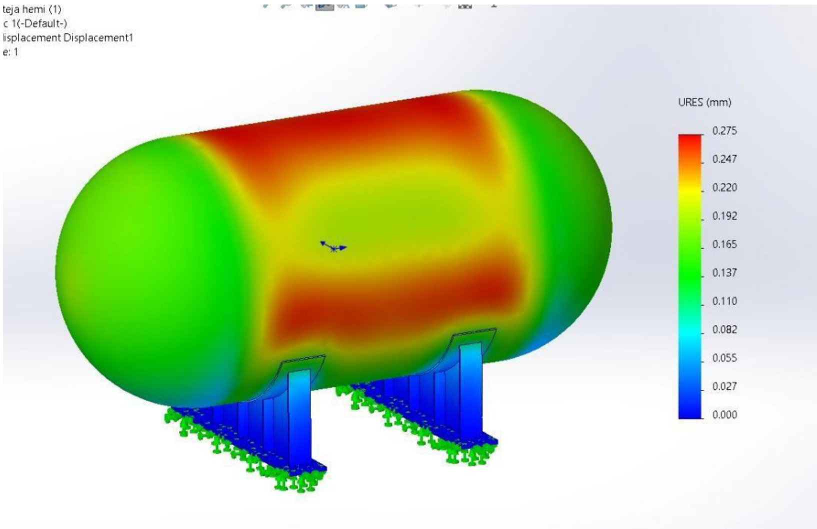


Fig 6. Hemispherical head stress distribution analysis in SolidWorks

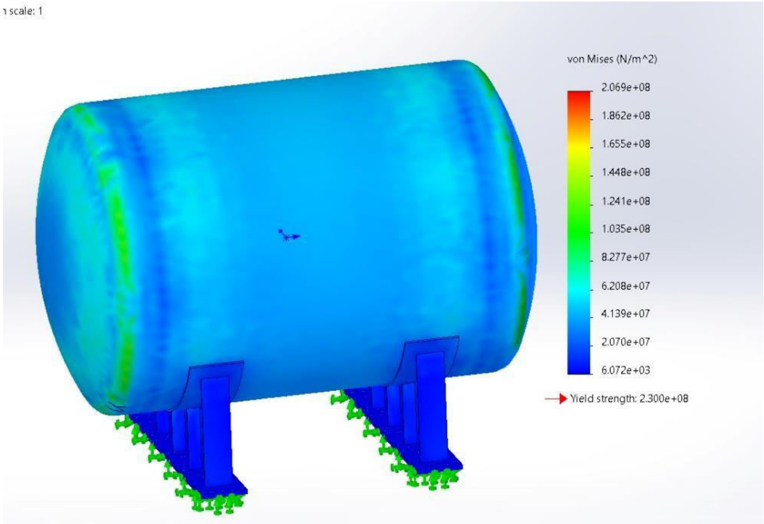


Fig 7. Torispherical head stress distribution analysis in SolidWorks

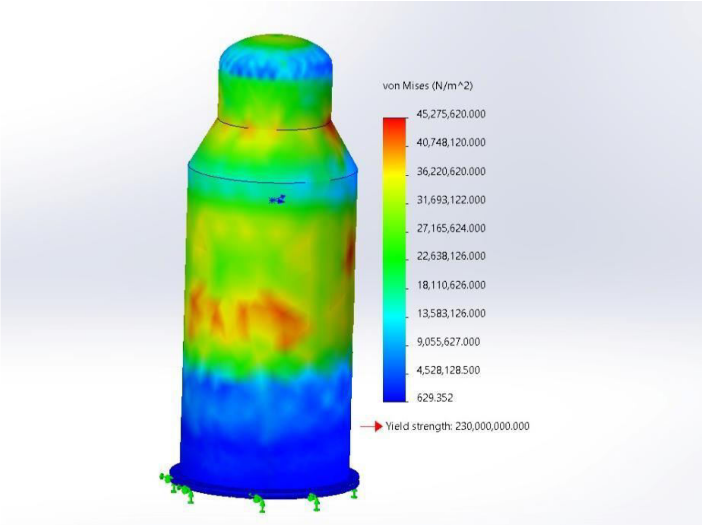


Fig 7. Conical head stress distribution analysis in SolidWorks

III. Manufacturing Process :

Evaluations for Different Modes of Fabrication

Hemispherical Head: Segmented welding (petal construction) is used for larger diameters. Several "petal-like" segments are welded to a central circular crown.

2:1 Ellipsoidal Head: Dishing or deep drawing is employed for fabrication.

Torispherical Head: Dishing and flanging processes are used to form the specific crown and knuckle radii.

Conical Head: These are produced by rolling plates followed by longitudinal welding.

Manufacturing Difficult:

Fabrication Complexity

High Difficulty: Hemispherical and conical heads are difficult to fabricate due to the continually changing rolling radius and Segmented welding.

Moderate Difficulty: Ellipsoidal and torispherical heads present moderate difficulty, as they are often standardised to simplify the manufacturing process.

Equipment Requirements: A high-capacity hydraulic press is required for deep drawing and flanging operations.

Inspection Challenges: Radiographic inspection of welds in these complex geometries is difficult due to varying material thicknesses and the intricate setup required to capture accurate views.

IV. Cost Estimation Breakdown

1. Material Cost

Torispherical and Conical Heads: These incur higher costs because they require a greater nominal thickness (14 mm and 12 mm).

Ellipsoidal and Hemispherical Heads: These have lower material costs due to a reduced required nominal thickness (10 mm and 8 mm).

2. Fabrication and Labour Costs

Conical Heads: Higher costs due to the complexity of rolling and welding.

Hemispherical Heads: These require "petal" construction and highly skilled labour, leading to increased man-hours.

3. Heat Treatment

Torispherical Heads: These require heat treatment to restore ductility and gain necessary strength after the forming process.

4. Results and Discussion

After a significant analysis of Hemispherical, 2:1 Ellipsoidal, Torispherical, and Conical heads based on structural behaviour, manufacturing complexity, and economic efficiency under a 0.7 MPa design pressure, the results are as follows:

Structural integrity and stress analyses were performed using SolidWorks Simulation under a 0.7 MPa load to evaluate stress concentration.

Hemispherical Head: Proved to be the most reliable and stable, exhibiting the lowest displacement at 0.275 mm.

2:1 Ellipsoidal Head: Offered the most balanced results, with a maximum stress of 9.76 MPa and a displacement of 0.78 mm.

Conical Head: Recorded the lowest peak stress at 45.28 MPa and a high factor of safety of 5.08.

Torispherical Head: Identified as the most vulnerable option, with the lowest factor of safety at 1.1.

Analytical Pressure Rating (MAWP):

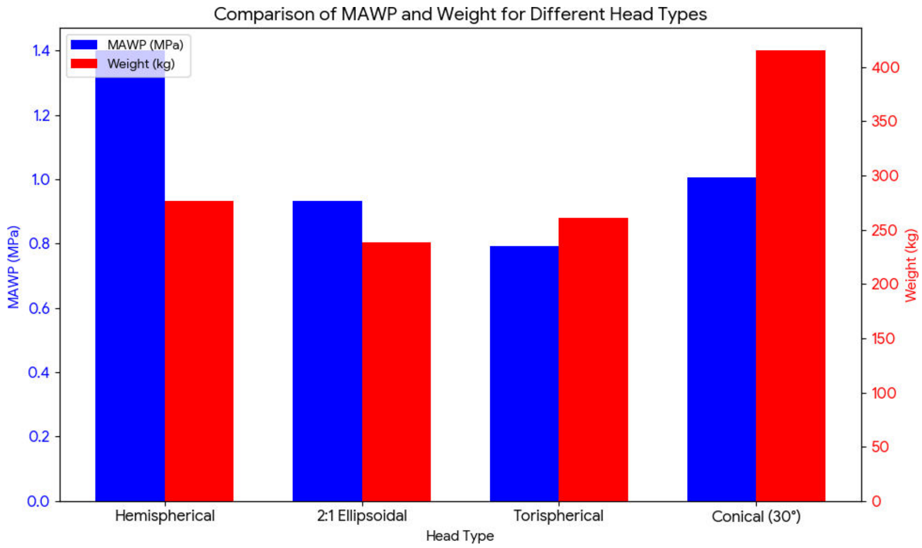
Using a bond curing joint efficiency of 0.85 and a 2 mm corrosion allowance, analyses were performed using PV Elite software.

Hemispherical Head: Proved to be the most suitable and efficient, achieving a MAWP of 2.23 MPa. This provides a 100% safety reserve over the design pressure.

Elliptical Head: Achieved a MAWP of 0.93 MPa, offering a 33% safety reserve, making it the standard choice for medium pressure.

Torispherical Head: Provided the lowest MAWP of all types at 0.71 MPa.

Hydrotest Results: The test performed on the Conical Head resulted in the highest stress at 74.27% of its allowable limit, whereas the Ellipsoidal Head reached only 48.5%.

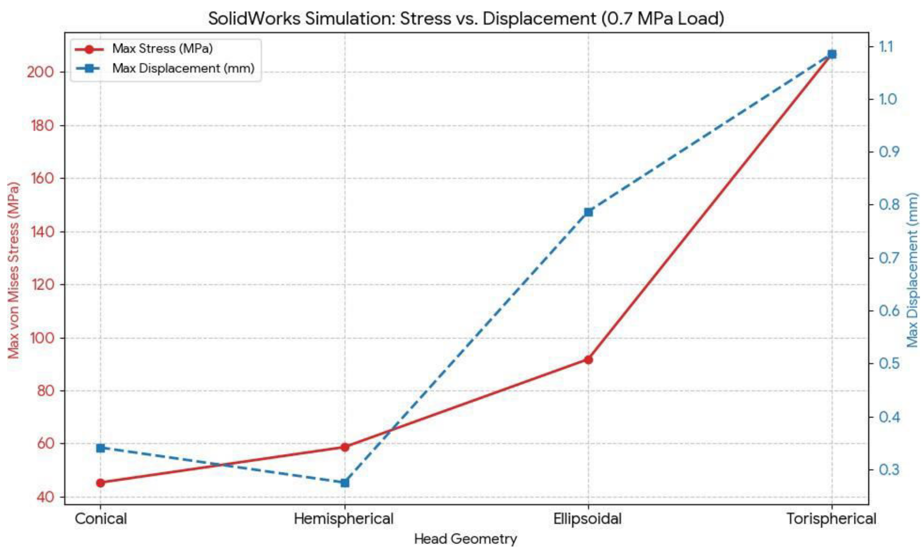


3. Manufacturing and Economic Efficiency

Torispherical Head Strain: Under UCS-79, the forming strain for torispherical heads is 9.652%, which exceeds the 5% threshold. In contrast, the strain for ellipsoidal heads remains within safe limits at 2.616%.

Structural Efficiency Index: The ratio of maximum allowable working pressure (MAWP) to weight is defined as the Structural Efficiency Index. The hemispherical head is the most efficient, with an index of 0.00507 MPa/kg, followed by the ellipsoidal head at 0.00391 MPa/kg.

Optimal Balance: The 2:1 ellipsoidal head represents the most balanced and reliable choice for general pressure vessel applications.



5. Conclusion

In conclusion, based on the analytical calculations, numerical analysis, manufacturing feasibility, and cost performance of various heads, the hemispherical head is the most trusted and reliable. It exhibits the lowest deformation, a high safety margin, and uniform stress distribution; however, it is not cost-effective due to manufacturing difficulties. In contrast, the elliptical head is a well-balanced option in terms of stress distribution, manufacturability, and moderate thickness, making it the best choice for industrial purposes. While the torispherical head is economical regarding fabrication and safety, it suffers from peak stress concentrations at the knuckle.

The performance of a conical head is adequate, but its performance is considered moderate because it undergoes high utilisation during hydrotesting; numerical analysis shows that localised stress effects result from this design. Finally, from research, we can conclude that for higher-pressure applications, a hemispherical head is best. For standard industrial conditions, an ellipsoidal head is the best choice in terms of safety, cost, and manufacturability.

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