DESIGN OF CYLINDRICAL VACUUM CHAMBERS FOR FULL VACUUM CONDITION USING STIFFENING RINGS AS PER ASME RULES

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ABSTRACT

This study presents a comprehensive and systematic design methodology for cylindrical vacuum chambers operating under full vacuum conditions, with primary emphasis on ensuring structural stability against external atmospheric pressure in accordance with ASME Section VIII, Division 1. Unlike internally pressurized vessels that are governed by tensile stresses, vacuum chambers are subjected to compressive stresses where elastic buckling becomes the dominant mode of failure, particularly in thin shells characterized by high diameter-to-thickness ratios and extended unsupported lengths. To address these challenges, the shell thickness and geometric parameters were determined using UG-28 provisions for external pressure design, while the adequacy of stiffening rings was verified based on UG-29 requirements to ensure sufficient moment of inertia and reduction in effective unsupported length.

In this work, a comparative analysis was carried out using different stiffening configurations, namely no stiffening ring, one stiffening ring, two stiffening rings, and three stiffening rings, to evaluate their influence on structural stability, critical buckling pressure, and allowable external pressure limits. The results clearly indicate that the absence of stiffening rings leads to reduced load-carrying capacity and higher susceptibility to buckling, whereas the introduction of stiffening rings progressively enhances stability by decreasing the effective unsupported length and increasing resistance to external pressure.

A dual-validation approach was employed by integrating manual calculations derived from ASME external pressure charts with PV Elite simulations, ensuring consistency and compliance with design codes. Furthermore, the study incorporates optimization of ring spacing, appropriate material selection, and sensitivity analysis of geometric variations to establish a robust and reproducible design workflow adaptable to various chamber configurations. The results demonstrate that the strategic incorporation of stiffening rings enables the safe use of thinner shells without compromising structural integrity, thereby providing a cost-effective and reliable design solution. This work effectively bridges the gap between empirical ASME methodologies and computational tools such as PV Elite, offering a structured, code-compliant approach that enhances predictive accuracy, improves safety in vessel construction, and contributes a validated framework for modern pressure vessel engineering applications.

Keywords: Vacuum Chamber, External Pressure Vessel, ASME Section VIII, Stiffening Rings, M.A.W.P., Elastic Buckling, End Closure, Finite Element Analysis.

INTRODUCTION:

Pressure vessels subjected to external pressure commonly referred to as vacuum chambers—are critical components in a wide range of industrial processes such as plasma nitriding, heat treatment, electron beam welding, chemical vapor deposition, and nuclear power applications. These chambers are specifically designed to maintain a controlled low-pressure environment inside while being exposed to atmospheric pressure externally. Their structural integrity is therefore of utmost importance, as failure can lead to sudden collapse and severe operational hazards.

Unlike internally pressurized vessels, which are primarily governed by tensile stresses and tend to fail through plastic deformation or yielding, externally pressurized vessels experience compressive stresses acting on their shell. Under such conditions, the dominant mode of failure is elastic buckling, which is highly sensitive to geometric parameters, material properties, and imperfections. Elastic buckling is a sudden and catastrophic failure mode that occurs when the compressive hoop stress in the shell exceeds a critical value, causing the structure to lose stability and collapse inward without significant prior deformation or warning.

This behavior is particularly critical in thin-walled cylindrical shells with high diameter-to-thickness (D/t) ratios and long unsupported lengths, where even small imperfections or residual stresses can significantly reduce the critical buckling pressure. Therefore, the design of vacuum chambers must carefully consider these factors and incorporate appropriate measures such as stiffening rings and optimized geometrical configurations. By reducing the effective unsupported length and increasing the structural rigidity, these design enhancements help improve resistance to buckling and ensure safe operation under external pressure conditions.

WORKING PRINCIPLE

A cylindrical vacuum chamber operates on the principle of maintaining a very low pressure or vacuum inside the vessel while being subjected to atmospheric pressure on the outside. This pressure difference generates compressive stresses on the shell, which tend to cause inward deformation and potential collapse of the structure. Unlike internal pressure vessels that expand outward, vacuum chambers are prone to instability due to external loading conditions.

The structural stability of the chamber depends on its ability to resist buckling under these compressive forces. To improve resistance, stiffening rings are introduced along the length of the cylinder. These rings reduce the effective unsupported length of the shell and increase its moment of inertia, thereby enhancing its load-carrying capacity. The design is carried out using ASME code procedures, where external pressure resistance is evaluated through UG-28 charts and stiffener adequacy is verified using UG-29 rules. The performance of the chamber is further analyzed and validated using simulation tools, ensuring reliable and safe operation under vacuum conditions.

LITERATURE REVIEW

1. N. Fazlalipour et al. (2025) presented a review on the buckling capacity of steel cylindrical shells under external pressure, focusing on the influence of reinforcements and imperfections. The study highlights that reinforcements such as stiffening rings and composite materials can significantly enhance the structural stability of shells. However, the presence of geometric imperfections reduces the critical buckling load and affects the failure behavior. The authors conclude that geometric parameters have a dominant influence on the buckling response of cylindrical shells.

2. Y. Xiang et al. (2025) investigated the buckling strength and sealing performance of cylindrical structures under external pressure. The study emphasizes that external pressure significantly reduces structural stability and affects sealing efficiency. It was observed that optimized geometric configurations improve both buckling resistance and structural integrity. The authors conclude that careful geometric design is essential for enhancing performance under pressure conditions.

3. J. Guo et al. (2025) conducted a buckling analysis and parametric optimization of metal-composite cylindrical shells. The study focuses on the influence of material anisotropy and design parameters such as fiber orientation and shell thickness. Results indicate that these parameters strongly affect the critical buckling load and overall structural response. The authors conclude that optimized composite design significantly enhances buckling performance.

4. K. Foroutan et al. (2025) performed a dynamic post-buckling analysis of stiffened cylindrical shells. The study highlights the effectiveness of spiral stiffeners in improving post-buckling strength and vibration resistance. It was observed that dynamic behavior plays a crucial role in shell stability under loading conditions. The authors conclude that stiffener configuration significantly influences post-buckling performance.

5. F. Tabish and I. Mamaghani (2025) studied the buckling behavior and design optimization of stiffened aluminum cylindrical shells. The research emphasizes the importance of stiffener spacing in improving structural performance. Results show that proper spacing significantly increases critical pressure and enhances stability. The authors conclude that optimized stiffener design is essential for preventing buckling failure.

6. M. Opstrup Andersen et al. investigated the self-buckling behavior of pressurized cylindrical tubes. The study reveals that internal pressure can stabilize cylindrical shells, whereas external pressure leads to instability and buckling. The interaction between internal and external pressures plays a critical role in structural response. The authors conclude that pressure conditions must be carefully evaluated in shell design.

7. U. Ramakrishna conducted a numerical study on the buckling behavior of perfect and imperfect cylindrical shells under external pressure. The results indicate that geometric imperfections and parameters such as radius-to-thickness ratio (R/t) and length-to-radius ratio (L/R) significantly reduce buckling strength. The study highlights the sensitivity of shells to imperfections. The author concludes that accurate modeling of imperfections is essential for reliable design.

8. M. Tănase et al. performed an analytical study of cylindrical shells under distributed external pressure. The study focuses on improving the prediction accuracy of buckling loads using advanced analytical models. Results show better agreement with realistic structural behavior. The authors conclude that analytical approaches are effective for preliminary design and validation.

9. Zhang L. et al. investigated the buckling behavior of composite cylindrical shells under hydrostatic pressure. The study demonstrates that composite materials provide a higher strength-to-weight ratio and improved resistance to buckling. The results highlight the advantages of composites over traditional materials. The authors conclude that composites are highly suitable for underwater and pressure applications.

10. Yuan K. et al. studied the buckling failure of composite cylindrical hulls under hydrostatic pressure. The research emphasizes the role of material layup and thickness variation in failure behavior. Results indicate that improper design can lead to premature buckling. The authors conclude that optimized layering is critical for structural safety.

11. Wei R. et al. examined the effect of delamination on composite cylindrical shells under pressure. The study shows that delamination significantly reduces buckling strength and structural integrity. The presence of defects accelerates failure mechanisms. The authors conclude that defect control is crucial in composite shell applications.

12. Shen K.C. et al. analyzed the buckling behavior of perforated cylindrical shells under hydrostatic pressure. The study highlights that openings reduce structural strength and increase stress concentration. However, the use of stiffeners can compensate for this reduction. The authors conclude that reinforcement is necessary for perforated shell designs.

13. Zuo X. et al. investigated the collapse behavior of stiffened cylindrical shells under external pressure. The study shows that helical and ring stiffeners significantly enhance collapse pressure. Different stiffening configurations were compared for performance evaluation. The authors conclude that stiffener design plays a vital role in improving buckling resistance.

14. Zhu Y. et al. studied the buckling performance of stiffened polymer composite cylindrical shells. The research demonstrates that stiffeners improve both elastic and plastic buckling behavior. The results indicate enhanced load-carrying capacity. The authors conclude that stiffening techniques are effective for polymer composite structures.

15. Guan W. et al. conducted experimental and numerical buckling analysis of CFRP cylindrical shells. The study shows good agreement between experimental results and finite element analysis. Composite shells exhibit superior buckling resistance compared to conventional materials. The authors conclude that CFRP is highly efficient for high-performance structural applications.

3 PROJECT OVERVIEW

3.1 PROJECT OVERVIEW

This project presents a detailed design methodology for cylindrical vacuum chambers operating under full vacuum conditions, with a focus on stability against external atmospheric pressure as per ASME Section VIII, Division 1. Unlike internal pressure vessels that experience tensile stresses, vacuum chambers are subjected to compressive stresses, making them highly susceptible to elastic buckling, especially in thin shells with high diameter-to-thickness ratios and long unsupported lengths.

To address this, external pressure design calculations were performed using UG-28 provisions, and stiffening ring design was carried out using UG-29 requirements. A comparative study was conducted for different configurations no stiffening ring, one stiffening ring, two stiffening rings, and three stiffening rings.

Based on the results shown in the analysis, the Maximum Allowable Working Pressure (M.A.W.P.) increases with the number of stiffening rings. The integration of stiffening rings significantly improves structural stability by reducing unsupported length and enhancing resistance to buckling. Validation was performed using both manual ASME calculations and PV Elite simulations, ensuring code compliance and accuracy.

3.2 OBJECTIVES OF THE PROJECT

1. To design a cylindrical vacuum chamber as per ASME Section VIII, Division 1.
2. To analyze the effect of external pressure on thin cylindrical shells.
3. To evaluate buckling behavior under full vacuum conditions.

4. To perform external pressure calculations using UG-28 provisions.
5. To design and validate stiffening rings using UG-29 requirements.
6. To compare performance for the following cases:
    - No stiffening ring
    - One stiffening ring
    - Two stiffening rings
    - Three stiffening rings
7. To determine Maximum Allowable Working Pressure (M.A.W.P.) for each case.
8. To validate analytical results using PV Elite software.
9. To optimize design for safety and cost-effectiveness.

3.3 PROBLEM IDENTIFICATION

Vacuum chambers designed without proper stiffening are highly prone to buckling failure due to external atmospheric pressure. The major challenges identified are:

- Thin cylindrical shells fail under compressive stresses rather than tensile stresses.
- High diameter-to-thickness ratio increases instability.
- Long unsupported lengths lead to early buckling.
- Conventional designs without stiffening rings result in low M.A.W.P. values.

From the analysis (image results):

- No stiffening ring configuration shows the lowest stability and minimum allowable pressure.
- One stiffening ring improves strength but is still limited.
- Two and three stiffening rings significantly increase M.A.W.P. and structural safety.

Hence, the key problem is to enhance buckling resistance while minimizing material usage and cost, ensuring compliance with ASME standards.

4 INTRODUCTION OF MODELING

This widely used engineering software for the design, analysis, and evaluation of pressure vessels and heat exchangers based on international design codes such as ASME Section VIII Division 1. It is developed to simplify complex pressure vessel calculations by integrating design, modeling, and analysis into a single user-friendly platform. The software is extensively used in industries such as oil and gas, petrochemical, power plants, and chemical processing, where safe and efficient pressure vessel design is critical.

PV Elite allows engineers to create detailed models of pressure vessels by defining components such as cylindrical shells, heads (elliptical, hemispherical, or torispherical), nozzles, and supports. It provides both input-based modeling and graphical visualization, including 2D sectional views and 3D representations. The software automatically performs essential calculations such as required thickness, stress analysis, nozzle reinforcement checks, and Maximum Allowable Working Pressure (MAWP), ensuring compliance with design codes.

One of the key advantages of PV Elite is its ability to handle both internal and external pressure conditions, including vacuum design where buckling is a critical factor. It also enables parametric studies, allowing users to optimize designs by modifying parameters like thickness, material, and stiffening rings. With its accurate calculations, detailed reports, and easy-to-use interface, PV Elite plays a vital role in improving design efficiency, safety, and reliability in pressure vessel engineering.

4.1 DESIGN PROCEDURE :

The design procedure of a pressure vessel with elliptical heads using PV Elite software begins with defining the fundamental input parameters and proceeds through modeling, analysis, and validation within a single integrated environment. Initially, essential vessel data such as inside diameter (500 mm), shell length (2000 mm), material (SS304 as per SA-240), design temperature (500°C), and external design pressure (1.013 bar) are entered based on operating conditions like plasma nitriding under full vacuum. The vessel is then modeled as a cylindrical shell with elliptical head end closures, which are commonly used due to their ability to provide uniform stress distribution and improved strength under pressure conditions. Nozzle details, such as internal diameter (100 mm), are also specified. applies ASME Section VIII Division 1 (UG-28) rules to evaluate shell thickness under external pressure, comparing required and actual thickness values across different shell spans to ensure safety against buckling, while also calculating nozzle thickness and verifying code compliance.

Once the input data is defined, the modeling process is carried out using PV Elite’s graphical interface, where the cylindrical shell, elliptical heads, and nozzles are created using dedicated modeling tools. The software generates both 2D sectional views and 3D models for clear visualization. Input tools are used to define loading conditions and material properties, while modeling tools such as shell, head (elliptical), nozzle, stiffening ring, and support tools (saddle or skirt) are used to construct the complete vessel geometry. **MAEP**

(Maximum Allowable External Pressure) refers to the maximum external pressure that a cylindrical shell or pressure vessel can safely withstand without undergoing buckling failure. In pressure vessel design, particularly as per ASME Section VIII Division 1 (UG-28), MAEP is a critical parameter used to evaluate the structural stability of vessels operating under vacuum or external pressure conditions. Unlike internal pressure cases where failure is governed by material yielding, external pressure failure occurs due to instability and collapse of the shell. The MAEP value depends on several factors such as shell diameter, thickness, length, material properties, and the presence of stiffening elements. For a safe design, the MAEP must be greater than or equal to the design external pressure, ensuring that the structure remains stable and does not buckle under operating conditions. This parameter is widely used in engineering analysis and software tools like PV Elite to verify the safety and reliability of pressure vessels.

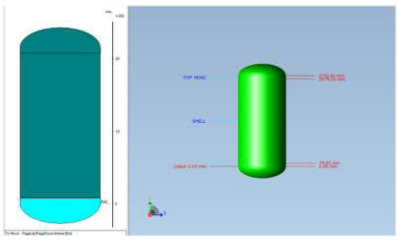

Figure 1 No stiffening ring pressure vessel

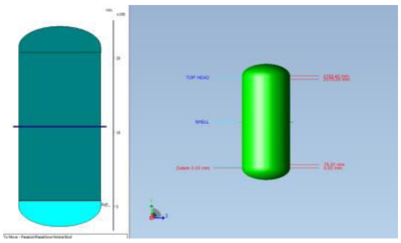

Figure 2 One stiffening ring pressure vessel

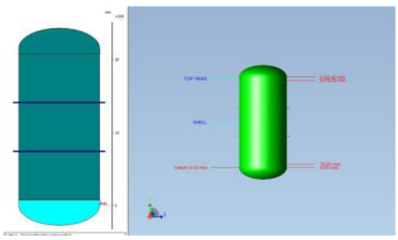

Figure 3 Two stiffening ring pressure vessel

4.2 DESIGN SPECIFICATIONS

A. PRESSURE VESSEL PARAMETERS

The pressure vessel is designed for a plasma nitriding process operating at temperatures up to 500°C with full internal vacuum. The critical design condition is uniform external pressure equal to atmospheric: P_design = 1.013 bar (101.325 kPa). The vessel is a cylindrical shell with the design parameters summarized in Table I.

TABLE I
DESIGN PARAMETERS OF PRESSURE VESSEL

Parameter	Value
Inside Diameter	500 mm
External Diameter	512 mm
Total Shell Length (L)	2000 mm
Design Pressure (External)	1.013 bar
Design Temperature	500 °C
Shell Material	SS304 (SA-240)
Corrosion Allowance	0 mm (SS304)
Shell Thickness (Nodes 20–30)	8.0 mm
Shell Thickness (End Bays)	6.0 mm
Number of Nozzles	12 (4 rows × 3)
Nozzle Internal Diameter	100 mm
Saddle Contact Angle	N/A (vertical rings)

B. MATERIAL DATA

The shell, nozzles, and stiffening rings are fabricated from stainless steel SS304 (ASME SA-240 Type 304), an austenitic 18/8 Chromium-Nickel alloy offering superior corrosion resistance and reliable mechanical properties at elevated temperatures. Key allowable stresses from ASME Section II Part D at 500°C are: Allowable Tensile Stress S = 86.18 N/mm², Elastic Modulus E = 193 GPa, Yield Strength S_y = 205 N/mm², and Density ρ = 7900 kg/m³.

4.3 PRESSURE VESSEL CALCULATIONS

A. REQUIRED SHELL THICKNESS

Per ASME UG-28, the required thickness of a cylindrical shell under external pressure is determined by an iterative Factor A–Factor B procedure. Assume trial thickness t = 6 mm. Compute L/D_o and D_o/t:

$$L/Do = 1000/512 = 1.953, \quad Do/t = 512/6 = 85.3 \qquad \ldots (1)$$

Reading Factor A from ASME Section II Part D, Fig. G [7] and Factor B from Fig. HA-1 [8] for SS304 at 500°C:

$$Factor\ A = 0.000408, \quad Factor\ B = 3830.67\ psi \qquad \ldots (2)$$

The maximum allowable external working pressure for the assumed thickness is:

$$Pa = 4B / (3 \times Do/t) = 4(3830.67)/[3(85.3)] = 59.85\ psi = 4.126\ bar \qquad \ldots (3)$$

Since Pa = 4.126 bar >> P_design = 1.013 bar, the assumed shell thickness t = 6 mm satisfies the ASME UG-28 requirement. The standard pipe schedule adopts t = 8 mm for the main shell span (Nodes 20–30) and t = 6 mm for the end bays, providing additional safety margin. The required thicknesses confirmed by software are listed in Table II.

TABLE II
SHELL SECTION THICKNESSES (UNSTIFFENED)

Shell Span	Actual t (mm)	Req. t (mm)	OK
Node 10→20	6.000	4.395	✓
Node 20→30	8.000	6.336	✓
Node 30→40	6.000	4.395	✓

B. REQUIRED NOZZLE THICKNESS

Nozzle internal diameter = 100 mm (D_o = 112 mm). Per ASME UG-28, for L = 70 mm bay length and t = 6 mm assumed: Factor A = 0.01579, Factor B = 7188, Pa = 513.42 psi (35.4 bar). The adopted standard schedule thickness t_n = 8.56 mm is fully adequate, requiring only 8 mm minimum.

4.4 PARAMETRIC STUDY: EFFECT OF STIFFENING RINGS ON M.A.W.P.

A systematic parametric study is conducted using ASME-compliant external pressure calculation software. The number of stiffening rings is varied from 0 to 3. For each configuration, M.A.W.P. is computed for every bay between adjacent supports. The minimum M.A.W.P. governs the vessel rating. All bays use SS304, design pressure = 1.013 bar, design temperature = 500°C.

A. NO STIFFENING RING

In the unstiffened configuration, the shell spans its full length between head flanges (2000 mm). The ASME external pressure software results are:

TABLE III
EXTERNAL PRESSURE RESULTS – NO STIFFENING RING

From→To	Act. t (mm)	Req. t (mm)	Des. P (bar)	M.A.W.P. (bar)
10 → 20	6.000	4.395	1.013	2.448
20 → 30	8.000	6.336	1.013	2.126
30 → 40	6.000	4.395	1.013	2.448
Minimum				2.126

The governing span is Node 20→30 (the main 2000 mm shell). With actual t = 8 mm against required 6.336 mm, M.A.W.P. = 2.126 bar = 2.10 × design pressure. This is structurally adequate but represents the lowest performance in the series.

B. ONE STIFFENING RING

A single ring is inserted at mid-span of the main shell, dividing it into two 1000 mm bays, each analyzed with t = 6 mm. Results:

TABLE IV
EXTERNAL PRESSURE RESULTS – ONE STIFFENING RING

From→To	Act. t (mm)	Req. t (mm)	Des. P (bar)	M.A.W.P. (bar)
10 → 20	6.000	4.395	1.013	2.448
20 → Ring	6.000	5.142	1.013	1.724
Ring → 30	6.000	5.142	1.013	1.724
30 → 40	6.000	4.395	1.013	2.448
Minimum				1.724

A critical counterintuitive result emerges: adding one ring reduces minimum M.A.W.P. from 2.126 bar to 1.724 bar (−18.9%). The ring bisects the thick (8 mm) central bay into two thinner (6 mm) bays; the required thickness of 5.142 mm is now closer to the actual 6 mm, reducing the M.A.W.P. margin. This demonstrates that naive addition of rings without careful thickness coordination is detrimental.

C. TWO STIFFENING RINGS

Two rings divide the main shell into three equal bays (approximately 667 mm each). Results:

TABLE V
EXTERNAL PRESSURE RESULTS – TWO STIFFENING RINGS

From→To	Act. t (mm)	Req. t (mm)	Des. P (bar)	M.A.W.P. (bar)
10 → 20	6.000	4.395	1.013	2.448
20 → Ring1	6.000	4.675	1.013	2.438
Ring1 → Ring2	6.000	4.393	1.013	3.088
Ring2 → 30	6.000	4.674	1.013	2.441
30 → 40	6.000	4.395	1.013	2.448
Minimum				2.438

With two rings, minimum M.A.W.P. recovers strongly to 2.438 bar—14.7% above the unstiffened baseline and 41.5% above the one-ring case. The central bay achieves 3.088 bar owing to the shortest effective length. The outer stiffened bays produce nearly symmetric M.A.W.P. values of 2.438–2.441 bar, confirming the symmetric geometry.

D. THREE STIFFENING RINGS

Three rings create four bays within the main shell span. Results:

TABLE VI
EXTERNAL PRESSURE RESULTS – THREE STIFFENING RINGS

From→To	Act. t (mm)	Req. t (mm)	Des. P (bar)	M.A.W.P. (bar)
10 → 20	6.000	4.395	1.013	2.448
20 → Ring1	6.000	4.397	1.013	3.077
Ring1 → Ring2	6.000	4.067	1.013	4.126
Ring2 → Ring3	6.000	4.067	1.013	4.126
Ring3 → 30	6.000	4.397	1.013	3.077
30 → 40	6.000	4.395	1.013	2.448
Minimum				2.448

Three rings achieve the highest minimum M.A.W.P. of 2.448 bar (+15.2% over unstiffened), now governed by the fixed end bays (Nodes 10–20 and 30–40) rather than the stiffened main shell bays. Central bays reach 4.126 bar. Beyond this configuration, further rings cannot improve performance as the end bays become the permanent constraint.

E. COMPARATIVE SUMMARY

TABLE VII
M.A.W.P. COMPARISON – ALL STIFFENING RING CONFIGURATIONS

Config.	Rings	Min. M.A.W.P. (bar)	Governing Bay	vs. Base
No Ring	0	2.126	Node 20–30	—
One Ring	1	1.724	20→Ring, Ring→30	−18.9%
Two Rings	2	2.438	20→Ring1, Ring2→30	+14.7%
Three Rings	3	2.448	Node 10–20 & 30–40	+15.2%

Table VII reveals that two and three rings both outperform the unstiffened baseline. The marginal gain of the third ring over two rings is only 0.010 bar (0.41%), making the two-ring configuration the most economical optimum. Beyond three rings, end bay geometry limits further improvement to M.A.W.P. = 2.448 bar. These results are consistent with Tabish and Mamaghani [2], who showed that stiffener count and spacing must be optimized together.

4.5 END CLOSURE DESIGN AND COMPARISON

A. HEAD GEOMETRIES CONSIDERED

Three end-closure geometries are evaluated under full vacuum per ASME Section VIII Division 1: (i) Flat circular head; (ii) Torispherical head; (iii) Hemispherical head. For the torispherical head (L-6.2): Crown radius $R = D_i = 500$ mm; Knuckle radius $r = 50$ mm (10%D); Straight flange = 3.5t. Thickness varied at t = 6, 8, 10 mm for each type [6].

B. FEA METHODOLOGY

Nine 3D solid models are generated in Creo Parametric 4.0 and exported to ANSYS Workbench 16 using tetrahedral elements (100, 150, 200 node divisions). Heads are fixed at their periphery; uniform external pressure of 101,325 Pa is applied to outer surfaces. Total deformation and von Mises equivalent stress are extracted. FEA results are validated against the classical flat plate closed-form solution [9]:

$$\delta_{max} = 3PR^4(1-\nu^2) / (16Et^3) \quad \ldots (4)$$

Agreement between FEA and theory is within 0.5%, confirming model fidelity.

C. HEAD COMPARISON RESULTS

TABLE VIII
MAXIMUM DEFORMATION AND STRESS OF END CLOSURES (200-NODE MESH, T = 6 MM)

Head Type	Max. Deform. (mm)	Max. VM Stress (MPa)	Cost
Flat	1.6194	126.29	Low
Torispherical	0.0277	14.645	Med.
Hemispherical	0.0023	4.057	High

The hemispherical head gives minimum deformation (0.0023 mm) but requires expensive forging. The flat head deforms 704× more than torispherical at t = 6 mm, necessitating greater thickness that negates its cost advantage. The torispherical head achieves a 0.0277 mm deformation—only 12× more than hemispherical—at intermediate fabrication cost and is selected as the optimal end closure, consistent with Lawate and Deshmukh [6].

NOZZLE AND REINFORCEMENT DESIGN

A. NOZZLE CONFIGURATION

Twelve nozzles (100 mm ID, $t_n = 8.56$ mm) are arranged in four circumferential rows of three nozzles each, spaced 90° apart and distributed axially at 500 mm intervals. Each nozzle is fabricated from SS304 and designed per ASME UG-28. The ASME software confirms Pa > 1.013 bar for all nozzle bay lengths up to 120 mm.

B. REINFORCEMENT PER ASME UG-38

Each opening requires supplementary reinforcement. The required replacement area is:

$$A\_req = d \times t\_r \times F = 100 \times 6 \times 1.0 = 600 \ mm^2 \quad \dots (5)$$

where d = finished opening diameter = 100 mm, t_r = required shell thickness = 6 mm, and F = 1.0 for perpendicular nozzles. Reinforcement pad dimensions: Outside diameter D_p = 140 mm; Pad thickness t_e = 10 mm; Fillet leg = 5 mm. These satisfy UG-38 area replacement requirements.

5 RESULTS AND DISCUSSIONS

NO STIFFENING RINGS:

From	To	Actual Thickness (mm)	Required Thickness (mm)	Design Pressure (bar)	MAWP (bar)
10	20	6.00000	4.39467	1.01300	2.44837
20	30	8.00000	6.33633	1.01300	2.12627
30	40	6.00000	4.39467	1.01300	2.44837
Minimum MAWP					**2.126**

The above table presents the external pressure analysis results for the pressure vessel without stiffening rings. It shows the comparison between actual shell thickness and the required thickness for different shell spans under a design external pressure of 1.013 bar (full vacuum condition). For all sections (10–20, 20–30, and 30–40), the actual thickness is greater than the required thickness, indicating that the design is safe against buckling as per ASME Section VIII Division 1 (UG-28) criteria.

However, the Maximum Allowable Working Pressure (MAWP) varies along the shell length due to differences in thickness and unsupported span. The lowest MAWP is observed in the middle section (20–30), with a value of 2.126 bar, which governs the overall design. Since this minimum MAWP is significantly higher than the design pressure (1.013 bar), the vessel is considered structurally safe even without the use of stiffening rings. This analysis confirms that the selected shell thickness provides adequate resistance against external pressure, although further optimization using stiffeners can improve stability and efficiency.

ONE STIFFENING RINGS

From	To	Actual Thickness (mm)	Required Thickness (mm)	Design Pressure (bar)	MAWP (bar)
10	20	6.00000	4.39467	1.01300	2.44837
20	Ring	6.00000	5.14200	1.01300	1.72404
Ring	30	6.00000	5.14200	1.01300	1.72404
30	40	6.00000	4.39467	1.01300	2.44837
Minimum MAWP					**1.724**

The above table presents the external pressure analysis results for the pressure vessel with one stiffening ring. The shell is divided into multiple sections, including regions between the vessel ends and the stiffening ring, to evaluate the effect of the ring on structural performance under a design external pressure of 1.013 bar. It is observed that in all sections, the actual thickness (6 mm) is greater than the required thickness, satisfying the ASME Section VIII Division 1 (UG-28) criteria for safety against buckling.

However, the introduction of a single stiffening ring alters the distribution of stresses and unsupported lengths, particularly in the regions adjacent to the ring (20–Ring and Ring–30). In these sections, the Maximum Allowable Working Pressure (MAWP) drops to 1.724 bar, which is the minimum value governing the design. Although this value is still higher than the design pressure, it is lower compared to the case without stiffening rings. This indicates that a single stiffening ring, if not optimally positioned, may not effectively enhance the vessel's resistance to external pressure. Therefore, proper placement and the number of stiffening rings are crucial for improving stability and achieving an efficient design.

TWO STIFFENING RINGS

From	To	Actual Thickness (mm)	Required Thickness (mm)	Design Pressure (bar)	MAWP (bar)
10	20	6.00000	4.39467	1.01300	2.44837
20	Ring	6.00000	4.67514	1.01300	2.43788
Ring	Ring	6.00000	4.39281	1.01300	3.08807
Ring	30	6.00000	4.67358	1.01300	2.44092
30	40	6.00000	4.39467	1.01300	2.44837
Minimum MAWP					**2.438**

The table presents the comparison between actual thickness and required thickness for different sections, along with the corresponding design pressure and Maximum Allowable Working Pressure (MAWP). For all sections, the actual thickness is maintained at 6 mm, while the required thickness varies across different segments such as 10–20, 20–Ring, Ring–Ring, Ring–30, and 30–40. The design pressure remains constant at 1.013 bar for all cases. The calculated MAWP values differ slightly for each section due to variation in required thickness, indicating the pressure-handling capability of each segment. Based on the results, the minimum MAWP obtained across all sections is 2.438 bar, which governs the overall safe operating limit of the system. This ensures that the component can safely withstand pressures up to this value without risk of failure.

6 CONCLUSION

This study presents a comprehensive design and parametric analysis of a cylindrical vacuum chamber with stiffening rings under full vacuum conditions using ASME Section VIII Division 1. The following conclusions are drawn:

1) The ASME UG-28 procedure determines that a6mmshell thickness (adopted 8 mm for the main span, 6mmfor end bays) satisfies all external pressure requirements with a safety margin exceeding 2× the design pressure for the unstiffened configuration.

2) The unstiffened vessel achieves a minimum M.A.W.P. of 2.126 bar, governed by the main2000mmshell span, representing a safety factor of 2.10over the1.013 bar design pressure.

3) Adding one stiffening ring is counter productive for this geometry: M.A.W.P. decreases by 18.9%to1.724bar due to unfavorable bay-thickness redistribution. This critically demonstrates that stiffener placement must be analyzed rigorously.

4) Two stiffening rings yield M.A.W.P. =2.438bar(+14.7% over unstiffened) and represent the economically optimal configuration. The marginal gain from a third ring is only 0.010 bar (0.41%), making two rings the recommended design choice.

5) Three stiffening rings achieve the maximum M.A.W.P. of 2.448 bar (+15.2%). Beyond this, end bay geometry (M.A.W.P. = 2.448 bar) limits further improvement regardless of additional ring count.

6) The torispherical head is confirmed as the optimal end closure: deformation of 0.0277 mm at t =6mmunder full vacuum—704× less than flat head at the same thickness—at intermediate fabrication cost, outperforming flat heads structurally and hemispherical heads economically.

7) All auxiliary components (nozzles, reinforcement pads, flanges) satisfy their respective ASME code provisions with adequate margins, and seamless pipe construction is recommended to avoid residual-stress induced buckling pressure reductions.

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