DESIGN AND ANALYSIS OF SADDLE SUPPORTED PRESSURE VESSELS

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ABSTRACT

This study focuses on the design and analysis of saddle-supported horizontal pressure vessels used in industrial applications. Saddle supports are essential for maintaining stability and transferring loads safely to the foundation. The design is carried out in accordance with ASME Boiler and Pressure Vessel Code Section VIII Division 1. Key parameters such as base plate thickness, web plate thickness, and stiffening ribs are selected to ensure structural safety and cost efficiency. Stress analysis is performed using PV Elite software, and analytical evaluation is conducted using Zick's method. The results confirm that the designed vessel satisfies safety requirements under both operating and hydrotest conditions.

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KEYWORDS — Pressure Vessel; Saddle Support; Zick Analysis; ASME Section VIII; Structural Integrity; PV Elite.

I. INTRODUCTION

petrochemical plants and refineries. Unlike vertical vessels, they require saddle supports to handle the combined load of the vessel, internal fluid, and external attachments. Proper design of these supports is necessary to avoid excessive stresses and deformation.

Zick's method is widely used for evaluating stresses in saddle-supported vessels. It helps determine longitudinal, shear, and circumferential stresses developed due to loading conditions. In this work, the vessel is designed based on ASME Section VIII Division 1 guidelines, and additional loads such as wind and seismic forces are considered using relevant Indian Standards. The objective is to ensure safe and reliable operation of the pressure vessel.

TYPES OF SUPPORTS FOR PRESSURE VESSELS

Pressure vessels use different types of supports depending on their orientation and size.

SADDLE SUPPORTS : Used for horizontal vessels to distribute load evenly.

SKIRT SUPPORTS : Suitable for vertical vessels to resist external forces.

LEG SUPPORTS: Used for smaller vessels due to simplicity and low cost.

LUG SUPPORTS: Used when vessels are mounted on structural frames.

II. LITERATURE REVIEW

1. R. K. Rajput (2022) This study analyzed the structural behavior of horizontal pressure vessels using analytical and numerical methods based on ASME standards.
The author emphasized the role of saddle spacing and support width in reducing stress concentration. Finite Element Analysis (FEA) results were compared with classical Zick's method. Proper optimization of saddle location significantly reduces longitudinal stress and improves vessel life.

2. M. A. Khan and S. Ahmed (2023) This research focused on stress distribution in saddle-supported vessels under internal pressure and thermal loading.ANSYS simulation was used to evaluate deformation and stress variations at saddle regions. The study highlighted the effect of temperature gradients on support reactions.Combined thermal and pressure loads increase stress near saddles, requiring careful design consideration.

3. P. S. Rao et al. (2021) The paper investigated the comparison between Zick's method and modern FEA tools for vessel analysis. Different loading conditions including hydrotest and operating loads were evaluated. Results showed minor deviations between analytical and numerical approaches. Zick's method remains reliable for preliminary design, while FEA ensures higher accuracy.

4. J. Lee and H. Kim (2024) This study examined optimization of saddle supports using parametric modeling techniques. Various geometric parameters like saddle angle, thickness, and stiffeners were optimized. The analysis was conducted using advanced simulation tools. Optimized saddle geometry can reduce material usage while maintaining structural safety.

5. A. Gupta and V. Sharma (2022) The authors studied the effect of seismic and wind loads on horizontal pressure vessels as per Indian Standards. Dynamic loading conditions were applied to evaluate stress variation. Support reactions and displacement were analyzed under extreme conditions. Seismic loads significantly influence saddle design and must be included for safety compliance.

6. D. Zhang et al. (2023) This research focused on fatigue analysis of saddle-supported pressure vessels. Cyclic loading and stress concentration zones were identified using FEA. The study evaluated crack initiation near saddle regions. Fatigue failure is most likely near saddle supports, requiring reinforcement and inspection.

7. S. K. Singh (2024) The study presented cost optimization in pressure vessel support design Material selection and thickness optimization were performed while maintaining ASME compliance. Economic analysis was included along with structural validation. Efficient design can reduce fabrication cost without compromising safety.

III OBJECTIVES OF THE PROJECT

The primary objectives of this study are:

1. To design a horizontal pressure vessel in accordance with **ASME Section VIII Division 1** standards.

2. To evaluate the structural integrity of the vessel using **Zick's Analytical Method** for saddle-induced stresses.

3. To perform a comparative stress analysis between operating and hydrotest conditions to ensure safety under maximum loading.

4. To verify the saddle support components (web, ribs, and base plate) using **AISC Unity Checks** and bearing pressure limits.

5. To optimize the thickness of the shell and support members for cost-efficiency without compromising safety.

MECHANICAL DESIGN CONSIDERATION

Designing a saddle-supported vessel requires addressing several critical mechanical factors:

VESSEL GEOMETRY: The ratio of the distance from the head to the saddle (A) versus the total length (L). If A > 0.25L, the vessel may behave differently than a simple beam.

SADDLE CONTACT ANGLE (θ): Usually ranges between 120° and 150°. A larger angle reduces the peak circumferential stress at the saddle horn.

INTERNAL PRESSURE (P): Determines the minimum required thickness (tr) of the shell and heads to prevent rupture.

STRESS CATEGORIES:

Longitudinal Bending: The vessel acts as a beam supported at two points.

Tangential Shear: Occurs at the plane of the saddle.

Circumferential Horn Stress: Highly localized stress at the edge of the saddle cradle.

Thermal Expansion: In high-temperature applications, one saddle is usually "fixed" while the other is "sliding" (slotted holes) to allow for longitudinal expansion.

IV. DESIGN SPECIFICATIONS

A. PRESSURE VESSEL PARAMETERS

The pressure vessel and its saddle support system are designed based on the following input parameters. The vessel

is a horizontal cylindrical shell fabricated from SA-516 Grade 70 carbon steel and supported on two symmetrically placed saddle assemblies. Key design parameters are summarized in Table I.

TABLE I

DESIGN PARAMETERS OF PRESSURE VESSEL AND SADDLE SUPPORT

PARAMETER	VALUE
Inside Diameter	1117.6 mm
T-T Length (L)	2000 mm
Design Pressure	6.8946 bar
Shell Material	SA-516 Gr.70
Corrosion Allowance	1.5 mm
Shell Thickness	10.0 mm
Joint Efficiency	1.0
Saddle Contact Angle	120°
Saddle Width	200 mm
Wear Plate Thickness	10 mm

Base Plate Thickness	20 mm
No. of Ribs	4 (2 out + 2 in)

B. MATERIAL DATA

The shell, heads, and saddle wear plate are fabricated from SA-516 Grade 70 steel. The saddle structural members are fabricated from ASTM A36 / IS 2062 E250 structural steel. Allowable stress limits and design methodology are per Bednar [4] and Brownell & Young [5]. Key material properties are: Allowable Tensile Stress S = 137.90 N/mm², Yield Strength Sy = 240.69 N/mm², Allowable Bending Stress Sba = 174.67 N/mm², and Bolt Allowable Stress = 172.38 N/mm².

Pressure Vessel Calculations

A. REQUIRED SHELL THICKNESS

Per ASME UG-27(c)(1), the required thickness of a cylindrical shell under internal pressure is given by Equation (1):

$t_r = P \times R / (S \times E - 0.6 \times P)$ … (1)

Substituting P = 0.6895 N/mm², R = 558.8 mm, S = 137.90 N/mm², E = 1.0: tr = 2.803 mm. Adding corrosion allowance of 1.5 mm gives 4.303 mm. The selected nominal thickness of 10.0 mm is adequate.

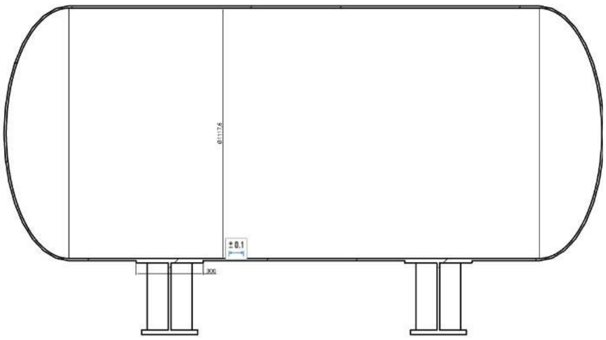

Fig:1 Dimensions of Pressure Vessel

B. REQUIRED HEAD THICKNESS

Per ASME UG-32(d) for semi-ellipsoidal 2:1 heads:

$t_h = P \times D / (2 \times S \times E - 0.2 \times P) = 2.795$ mm … (2)

With corrosion allowance: 4.295 mm. Nominal thickness of 10.0 mm is adequate.

C. VESSEL WEIGHT AND SADDLE REACTIONS

The total saddle reaction force Q is computed from the vessel self-weight, contents, and any external loads. For the operating case Q = 643.62 Kgf per saddle, and for the hydrotest case Q = 1,749.47 Kgf per saddle.

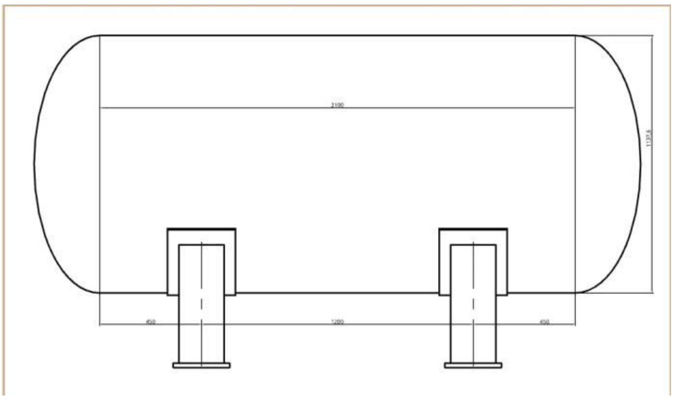

Fig:1.1 Dimensions of Pressure Vessel

V. Saddle Support Calculations

A. CROSS-SECTION PROPERTIES

The moment of inertia of the saddle cross-section is computed by summing contributions from the shell, wear plate, web plate, and base plate using the parallel-axis theorem, following the methodology of Singh & Soler [8]. Results: Centroid C_1 = 135 mm, Moment of Inertia I = 24,238 cm⁴, Cross-sectional area As = 111 cm².

B. HORIZONTAL FORCE ON SADDLE

The horizontal force component is determined using the Zick lateral-load coefficient K_1:

$Fh = K_1 \times Q$ … (3)

Hydrotest case: Fh = 0.2035 × 1749.47 = 356.056 Kgf.

Saddle Analysis for Longitudinal Stresses

A. ZICK BENDING COEFFICIENTS

The Zick analysis uses geometry-dependent coefficients K_1 and K determined from the ratio A/L = 400/2100 = 0.1905 and saddle angle θ = 120°: K_1 = 1.3807, K = 9.3799, X = 0.1932.

B. LONGITUDINAL BENDING STRESS AT SADDLE

The longitudinal bending stress at the saddle location is computed as:

$S_1 = (0.25 \times Q \times L \times K_1) / (\pi \times R^2 \times t_e)$ … (4)

Operating: S_1 = 23.13 N/mm² < 117.22 N/mm² (Safe). Hydrotest: S_1 = 59.33 N/mm² < 117.22 N/mm² (Safe).

VI. Transverse Direction Stresses

A. TANGENTIAL SHEAR STRESS

The tangential shear stress in the shell near the saddle is given by Zick's formula. Operating: S_2 = 0.67 N/mm² < 110.32 N/mm² (Safe). Hydrotest: S_2 = 1.81 N/mm² < 110.32 N/mm² (Safe).

B. CIRCUMFERENTIAL BENDING STRESS AT SADDLE HORN

The circumferential stress at the horn of the saddle is evaluated as:

$S_4 = -Q/[4 \times t_s \times (b + 1.56\sqrt{(R \times t_e)})] - 12 \times Q \times R \times K_7/(L \times t_s^2)$ … (5)

Hydrotest: $S_4 = -24.53$ N/mm² > −206.85 N/mm² allowable (Safe). Circumferential compressive stress in shell: −2.35 N/mm² > −120.35 N/mm² (Safe).

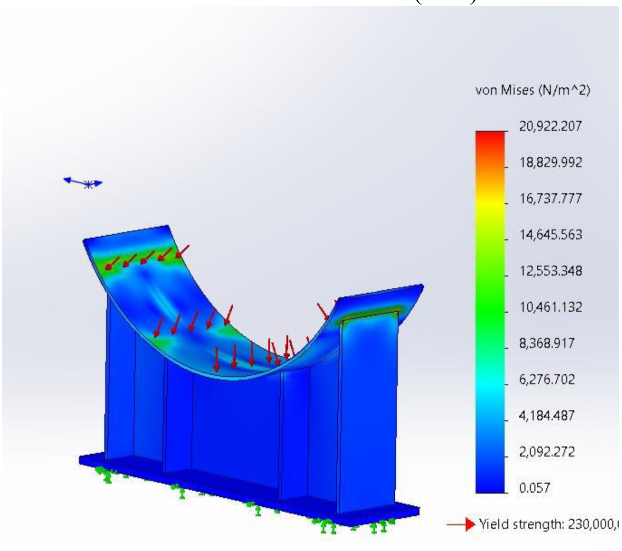

Fig:2 Static Analysis of Saddle Support

VII. Bending Moment and Stress Calculations

A. At Saddle Locations

The bending moment in the saddle cross-section due to the lateral force is M = Fh × d, where d = 396.94 mm (distance to neutral axis). Hydrotest: M = 141.334 Kg·m. Bending stress:

$S_b = M \times C_1 / I = 0.7728$ N/mm² < 174.67 N/mm² (Safe) … (6)

B. At Mid-Span

The mid-span section experiences maximum longitudinal bending. Stress results are summarized in Table II.

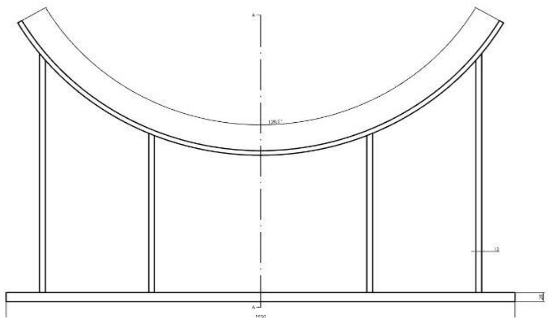

Fig:3 Dimensions of Saddle Support

Table II

Stress Results Summary — Hydrotest Case, Left Saddle

Stress Type	Actual (N/mm²)	Allow. (N/mm²)	OK
Long. at saddle top	59.33	117.22	✓
Long. at saddle btm	56.36	117.22	✓
Long. at mid-span top	57.65	117.22	✓
Long. at mid-span btm	58.03	117.22	✓
Tang. shear in shell	1.81	110.32	✓
Circ. at saddle horn	−24.53	−206.85	✓
Circ. at wear plate tip	−21.06	−206.85	✓
Circ. comp. in shell	−2.35	−120.35	✓

VIII. Web Tension Check

The web plate carries the horizontal component of the vessel load in tension. Tensile stress is computed as:

$S_t = Fh / As = 356.056 / 111 = 0.3135$ N/mm² < 157.21 N/mm² (Safe) … (7)

The safety factor for web tension under the most severe hydrotest condition exceeds 500, confirming that the web is adequately sized with significant reserve capacity.

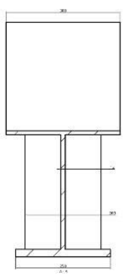

Fig:3.1 Dimensions of Saddle Support

IX. Lower Section of Saddle

A. Axial Compressive Stress in Ribs

Axial load P = Ap × Bp = 426.9 × 0.67 = 284.5 Kgf. Compressive stress Sc = 284.5 / 46.004 = 0.6065 N/mm².

B. AISC UNITY CHECKS

The AISC E2-1 unity check combines axial compression and bending:

$$UC = Sc/Sca + (Sb/Z)/Sba \leq 1.0 \quad \dots (8)$$

Results: Outside Ribs (Hydrotest) UC = 0.01 ✓; Inside Ribs (Hydrotest) UC = 0.01 ✓. All unity checks are well within the threshold of 1.0, confirming rigid structural behavior under all load combinations. **CODE COMPLIANCE:** Modern engineering relies on the **ASME Boiler and Pressure Vessel Code (BPVC) Section VIII Division 1**. Specifically, Non-mandatory Appendix G provides the framework for saddle design, ensuring that localized stresses at the saddle horns do not exceed the yield strength of the material.

COMPUTATIONAL METHODS: Recent studies by Singh, Soler, and Moss have integrated Zick's formulas into Finite Element Analysis (FEA) and specialized software like **PV Elite**. These tools allow for the consideration of secondary loads such as seismic (IS 1893) and wind (IS 875) forces, which are critical for outdoor process plant installations.

MATERIAL SCIENCE: Research into SA-516 Grade 70 carbon steel highlights its efficacy for pressure vessel fabrication due to its high notch toughness and excellent weldability, making it the preferred choice for saddle-supported systems.

X. BEARING PRESSURE

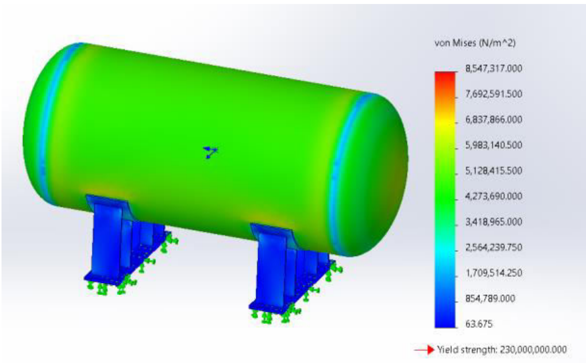

Fig:4 Buckling Analysis of Pressure Vessel with Saddle Support

The bearing pressure on the foundation beneath the saddle base plate is:

$$q = Q_{total} / (Bplen \times Bpwidth) \quad \dots (9)$$

Hydrotest bearing pressure = (1912.58 × 9.81) / 315,000 = 0.0595 N/mm². This is well below the allowable bearing pressure for reinforced concrete foundations (≈ 0.3–0.5 N/mm²).

Minimum base plate thickness (Moss method [3]): t_bp = 4.379 mm. Deployed thickness of 20.0 mm provides a safety factor of 4.5 over the minimum requirement.

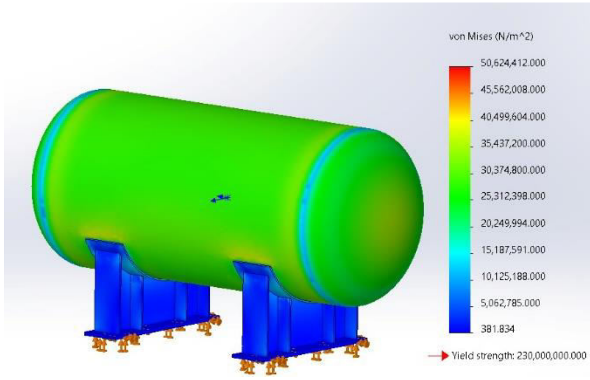

Fig:5 Static Analysis of Pressure Vessel with Saddle Support

XI CONCLUSIONS

The Design and analysis of a saddle-supported pressure vessel have been successfully carried out using ASME standards and Zick's method. The results show that all stresses developed in the vessel and supports are within allowable limits under both operating and hydrotest conditions.

The saddle structure, including web plates, ribs, and base plate, has sufficient strength to withstand applied loads. Bearing pressure on the foundation is also within safe limits. Overall, the design is safe, reliable, and suitable for industrial applications.

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