PRESSURE-VESSEL-DESIGN

DESIGN OF A HORIZONTAL PRESSURE VESSEL WITH A CYLINDRICAL SHELL, CONICAL AND 2:1 ELLIPSOIDAL HEADS, NOZZLE REINFORCEMENT, AND SADDLE SUPPORT.

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ABSTRACT

This report presents the design and structural evaluation of a horizontal pressure vessel used as a three-phase separator in oil well extraction. Operating at a design pressure of 1500 psi, the vessel consists of a cylindrical shell, a 2:1 ellipsoidal head, a conical head, reinforced nozzles, and saddle supports. SA516 Grade 70 carbon steel was selected as the fabrication material based on its mechanical properties and suitability for high-pressure applications. The design process was governed by ASME Boiler and Pressure Vessel Code standards. Initial dimensions—including internal diameter, vessel length, and analytical wall thicknesses—were established through code-based formulas. To validate these analytical calculations and identify potential stress concentrations, a computer-aided design (CAD) model was generated in OnShape and subjected to finite element analysis (FEA) using Fusion360. The structural simulation confirmed that the pressure vessel can safely sustain the 1500 psi operating pressure without mechanical failure. Furthermore, the FEA successfully identified the center of the 2:1 ellipsoidal head as the system's primary localized weak point, providing critical insight for safety margins and future design iterations. Beyond structural integrity, the finalized design was comprehensively evaluated against real-world engineering constraints, including manufacturability, total weight estimation, fabrication costs, and non-destructive inspection requirements mandated by ASME code. Ultimately, this engineering document links theoretical code design with practical engineering verification, ensuring a safe, cost-effective, and code-compliant solution for high-pressure oil and gas separation.

Keywords: pressure vessel, ASME Section VIII, cylindrical shell, ellipsoidal head, conical head, nozzle reinforcement, saddle support, finite element analysis

NOMENCLATURE

Roman Letters

C_m	Material cost per unit mass or weight [$/lb]
D_i	Inside diameter of vessel [in.]
E	Weld joint efficiency [-]
L	Straight shell length or tangent-to-tangent length [in.]
P	Internal design pressure [psi]
R_i	Inside radius of vessel [in.]
S	Allowable stress of selected material [psi]
W	Estimated vessel weight [lb]

Lowercase Roman Letters

t_a	Corrosion allowance [in.]
t_h	Required head thickness [in.]
t_s	Required cylindrical shell thickness [in.]

Greek Letters

α	Cone half-apex angle [deg.]
ρ	Material weight density [lb/in^3]
σ	Normal stress [psi]
σ_{vm}	Von Mises stress [psi]

Subscripts

c	Corrosion allowance
h	Head
i	Inside
m	Material
s	Shell
vm	Von Mises

1. INTRODUCTION

Oil makes up roughly one-third of global energy production and accounts for nearly 90% of the energy utilized in global transportation [1]. A key step in the oil extraction process is to separate the unrefined oil from natural gas and water from the well, which can be done with a horizontal separator. The well fluid mixture is unable to go directly into fractional distillation since it contains water, dissolved salts, and gases, which can disrupt the distillation process. By utilizing a separator to extract unrefined oil prior to distillation, equipment corrosion and dangerous pressure buildup can be avoided [2]. There are a variety of different horizontal separator designs, and our focus is on the most common method of separation, the gravity separation design. Benefitting from the fact that different fluids have different densities, we can allow gravity to pull the mixture into its separate components within the horizontal vessel [3, 4]. Oil is free to move above the water layer, where it can be subsequently skimmed and removed. Post-separation, the unrefined oil can undergo further separation via fractional distillation and other processes not relevant to this paper.

This report focuses on the design of a three-phase horizontal separator operating under high pressure, 1500 psi. "Three-phase" simply means the well fluid mixture is separated into the aforementioned three fluids: natural gas, water, and unrefined oils [5]. While horizontal separators can operate at lower pressures, the vessel benefits from operating at this higher pressure to accommodate the initial high pressure of a new oil well and maintain controlled containment of the wellstream. Natural gas is highly flammable under appropriate fuel–air conditions, and fuel flash point has been shown to depend on pressure [6]. Therefore, the vessel design must account for pressure containment, leak prevention, proper relief protection, and safe separation of gas, oil, and water during operation.

1.1. Project Objectives

The main objectives of this project are:

1. Identify a practical pressure-vessel application through literature review.
2. Specify the design pressure, internal diameter, vessel length, material, corrosion allowance, and joint efficiency.
3. Calculate the required wall thickness and weight of the cylindrical shell.
4. Calculate the required wall thickness and weight of the selected pressure-vessel heads.
5. Create a CAD model including the cylindrical shell, heads, and saddle supports.
6. Perform structural analysis of the shell-and-head assembly under internal pressure.
7. Estimate the vessel manufacturing cost using material weight and fabrication cost assumptions.
8. Discuss relevant testing and inspection methods for pressure vessels.

2. APPLICATION SELECTION AND LITERATURE REVIEW

2.1. Practical Application

The selected vessel is a horizontal three-phase separator for upstream oil production. This application was selected because produced well fluids commonly contain crude oil, water, and natural gas, which must be separated before downstream refining and processing. A horizontal separator is suitable for this application because it provides a long residence path for gravity separation, allows gas to collect in the upper region of the vessel, and provides separate outlets for oil, water, and gas phases [5, 15]. The selected design therefore uses a horizontal cylindrical shell with a conical head for drainage, a 2:1 ellipsoidal head for pressure containment, multiple nozzles for inlet/outlet connections, and two saddle supports.

2.2. Relevant Literature and Design Standards

The design methodology follows pressure vessel design practices based on ASME Section VIII concepts. The literature review focuses on:

- pressure vessel shell and head design under internal pressure,
- nozzle reinforcement for shell openings,
- saddle support design for horizontal vessels,
- material selection and allowable stress,
- hydrostatic/pneumatic testing and nondestructive examination,
- finite element analysis of pressure vessel stress concentrations.

3. DESIGN BASIS AND INPUT PARAMETERS

3.1. Design Requirements

TABLE 1: Primary design inputs for the pressure vessel.

Parameter	Symbol	Value
Design pressure	P	1500 psi
Inside diameter	D_i	48 in
Inside radius	R_i	24 in
Shell length	L	129 in
Corrosion allowance	t_a	0.125 in
Allowable stress	S	17 500 psi
Joint efficiency	E	0.85
Material density	ρ	0.2833 lb in^{-3}
Nozzle diameter	d_i	6 in
Selected material	–	SA-516 Grade 70 carbon steel

The design inputs in Table 1 were selected from the project design basis, pressure-vessel material data, and reference design examples. The allowable stress was taken as 17 500 psi for SA-516 Grade 70 carbon steel [8]. The selected joint efficiency of 0.85 follows the pressure-vessel design example used as a reference for welded vessel calculations [9]. The selected corrosion allowance of 0.125 in accounts for lifetime material loss in service.

3.2. Material Selection

The baseline material selected for the vessel is ASME SA-516 Grade 70 carbon steel. This material was selected because it is commonly used for welded pressure vessels and provides an appropriate balance of strength, weldability, availability, and cost for industrial pressure-vessel applications. The Pressure Vessel Design Manual lists SA-516 Grade 70 as a common pressure-vessel plate material and gives representative mechanical properties for this material class [7]. For the present design, the allowable stress is taken as 17 500 psi, and the material density is taken as approximately 0.2833 lb in^{-3} [8, 17]. Because the vessel may be installed outdoors, corrosion protection such as paint, epoxy coating, or zinc galvanizing is considered for environmental protection [11].

4. ANALYTICAL PRESSURE VESSEL DESIGN

The analytical pressure-vessel design was completed before CAD modeling and finite element analysis. The purpose of the analytical design was to determine the required wall thicknesses of the cylindrical shell, the 2:1 ellipsoidal head, and the conical head under the specified internal design pressure. These calculations provide the baseline dimensions used for the CAD model and the structural verification.

The main design inputs used in the analytical calculations are summarized in Table 1. The vessel was designed for an internal pressure of 1500 psi, an inside diameter of 48 in, and an inside radius of 24 in. The selected material was SA-516 Grade 70 carbon steel, with an allowable stress of 17 500 psi. A weld joint efficiency of $E = 0.85$ was used to account for the reduced strength of welded joints. A corrosion allowance of 0.125 in was added to the calculated pressure thicknesses to account for expected material loss during service life.

4.1. Cylindrical Shell Thickness

For a cylindrical pressure vessel under internal pressure, the circumferential or hoop stress is typically the governing stress condition. This occurs because the hoop stress in a thin cylindrical shell is approximately twice the longitudinal stress. Therefore, the cylindrical shell thickness was calculated using the ASME-based internal-pressure relation for cylindrical shells [7]:

$$t_s = \frac{PR_i}{SE - 0.6P}. \tag{1}$$

In this equation, t_s is the required shell thickness before corrosion allowance, P is the internal design pressure, R_i is the inside radius, S is the allowable material stress, and E is the weld joint efficiency. The term $0.6P$ accounts for the pressure correction used in the ASME cylindrical-shell relation.

Substituting the selected design values,

$$t_s = \frac{(1500 \text{ psi})(24 \text{ in})}{(17500 \text{ psi})(0.85) - 0.6(1500 \text{ psi})}. \tag{2}$$

$$t_s = \frac{36000}{14875 - 900} = \frac{36000}{13975} = 2.576 \text{ in}. \tag{3}$$

The calculated pressure thickness is therefore 2.576 in. Since corrosion allowance must be added after the pressure thickness is calculated, the total shell thickness becomes

$$t_{s,\text{total}} = t_s + t_a. \tag{4}$$

$$t_{s,\text{total}} = 2.576 \text{ in} + 0.125 \text{ in} = 2.701 \text{ in}. \tag{5}$$

Thus, the cylindrical shell thickness used for the design was approximately 2.701 in.

4.2. Head Geometry Selection

Three possible pressure-vessel head geometries were considered for this design:

- conical head,
- 2:1 ellipsoidal head,
- hemispherical head.

Each head geometry has different structural and manufacturing characteristics. A hemispherical head generally provides the most favorable stress distribution because the curvature is smooth and the membrane stresses are relatively low. However, hemispherical heads are typically more expensive and more difficult to fabricate. A 2:1 ellipsoidal head is commonly used in pressure-vessel design because it provides a practical balance between structural efficiency, manufacturability, and cost. A conical head is less structurally efficient than a smooth ellipsoidal or hemispherical head because the change in geometry can introduce discontinuity stresses near the cone-to-shell junction. However, a conical head is useful for drainage or collection of liquid phases.

The selected vessel uses one conical head and one 2:1 ellipsoidal head. The conical head is used on one side of the separator to improve drainage of the separated liquid phase, while the 2:1 ellipsoidal head is used on the opposite side as a standard pressure-vessel closure. Compared with flat heads, both formed heads reduce stress concentration and required thickness. However, the conical head requires a larger thickness than the ellipsoidal head because of the geometry and the half-apex angle correction [7, 10, 16].

4.2.1. 2:1 Ellipsoidal Head Thickness For a 2:1 ellipsoidal head under internal pressure, the required head thickness was calculated using the ASME-based ellipsoidal-head pressure relation [7]:

$$t_E = \frac{PD_i}{2SE - 0.2P}. \tag{6}$$

In this equation, t_E is the required ellipsoidal head thickness before corrosion allowance, P is the internal design pressure, D_i is the inside diameter of the vessel, S is the allowable stress, and E is the weld joint efficiency. The correction term $0.2P$ is used in the ASME-based relation for ellipsoidal heads.

Substituting the design values,

$$t_E = \frac{(1500 \text{ psi})(48 \text{ in})}{2(17500 \text{ psi})(0.85) - 0.2(1500 \text{ psi})}. \tag{7}$$

$$t_E = \frac{72000}{29750 - 300} = \frac{72000}{29450} = 2.445 \text{ in}. \tag{8}$$

Including the corrosion allowance,

$$t_{E,\text{total}} = t_E + t_a. \tag{9}$$

$$t_{E,\text{total}} = 2.445 \text{ in} + 0.125 \text{ in} = 2.570 \text{ in}. \tag{10}$$

Therefore, the total 2:1 ellipsoidal head thickness used for the design was approximately 2.570 in.

4.2.2. Conical Head Thickness The conical head thickness was estimated using the ASME-based internal-pressure relation for conical heads [7]:

$$t_\text{cone} = \frac{PD_i}{2\cos\alpha\,(SE - 0.6P)}. \tag{11}$$

In this equation, t_{cone} is the required conical head thickness before corrosion allowance, P is the internal design pressure, D_i is the inside diameter at the large end of the cone, S is the allowable material stress, E is the weld joint efficiency, and α is the cone half-apex angle. A half-apex angle of 30° was selected for this design.

The cosine term accounts for the inclined geometry of the conical head. As the cone angle increases, $\cos\alpha$ decreases, which increases the required thickness. This means that steeper cone geometries generally require thicker walls to resist the same internal pressure.

Substituting the selected design values,

$$t_{\text{cone}} = \frac{(1500\text{ psi})(48\text{ in})}{2\cos(30°)\,[(17500\text{ psi})(0.85) - 0.6(1500\text{ psi})]}. \quad (12)$$

$$t_{\text{cone}} = \frac{72000}{2(0.866)(13975)} = 2.975\,\text{in}. \quad (13)$$

Including the corrosion allowance,

$$t_{\text{cone,total}} = t_{\text{cone}} + t_a. \quad (14)$$

$$t_{\text{cone,total}} = 2.975\text{ in} + 0.125\text{ in} = 3.100\,\text{in}. \quad (15)$$

Therefore, the total conical head thickness used for the design was approximately 3.100 in.

4.3. Summary of Analytical Thickness Results

The analytical thickness results are summarized in Table 2. The conical head required the largest total thickness because the conical geometry introduces an angular correction through the $\cos\alpha$ term. The cylindrical shell and 2:1 ellipsoidal head had similar required thicknesses, while the conical head required additional thickness to maintain adequate pressure containment.

TABLE 2: Summary of analytical pressure-vessel thickness calculations.

Component	Thickness [in.]	Allow. [in.]	Total [in.]
Shell	2.576	0.125	2.701
Ellipsoidal head	2.445	0.125	2.570
Conical head	2.975	0.125	3.100

These analytical thicknesses were used as the baseline wall dimensions for the CAD model and for the preliminary structural evaluation. The finite element analysis was then used to check the stress distribution of the shell-and-head assembly under the same 1500 psi internal design pressure.

4.4. Nozzles

Detailed nozzle reinforcement calculations were not required for this introductory pressure-vessel design project. However, nozzles were included in the CAD model because the selected three-phase separator application requires inlet and outlet connections for the gas, oil, and water phases. The model includes six simplified nozzles to represent these functional connections. The diameter of the nozzle was found as 6 in [15].

Although nozzle reinforcement was not analyzed structurally, the nozzle weight was included in the total vessel weight and cost estimate. The nozzle weight was estimated using Table 2-14 of the *Pressure Vessel Design Manual*, which lists approximate nozzle and manway weights by pressure rating and nominal size [7]. For a 6 in nozzle at a 1500 psi rating, the table gives an estimated nozzle weight of 215 lb. Since the vessel includes six nozzles, the total nozzle weight is

$$W_n = 6(215\text{ lb}) = 1290\,\text{lb}. \quad (16)$$

The reference table used for this estimate is included in Appendix A.

4.5. Weight Estimation

The vessel weight was estimated from the calculated wall thicknesses, component surface areas, and the density of SA-516 Grade 70 carbon steel. Detailed substitution calculations are provided in Appendix A. The density used for all weight estimates was $\rho = 0.2833\,\text{lb/in}^3$.

The cylindrical shell weight was estimated as

$$W_s = \pi(D_i + t_s)L_s t_s \rho. \quad (17)$$

The conical head weight was estimated using

$$W_c = A_c t_c \rho, \quad (18)$$

where t_c is the total conical head thickness including corrosion allowance, and A_c is the estimated conical head surface area:

$$A_C = \pi[0.5(D_i + d_i)]\sqrt{L_C^2 + 0.5(D_i - d_i)^2}. \quad (19)$$

"Procedure 2-5: Design of Cones" of the *Pressure Vessel Design Manual* gives the equation for the length of the cone along the shell, L_c as

$$L_c = \sqrt{L^2 + (R_i - r_i)^2}, \quad (20)$$

where L is the length of the cone along the axis of the cone, given by

$$L = [0.5(D_i - d_i)]\tan\alpha. \quad (21)$$

The 2:1 ellipsoidal head weight was estimated using

$$W_E = 1.084(D_i + t_E)^2 t_E \rho, \quad (22)$$

where t_E is the total ellipsoidal head thickness including corrosion allowance.

The total estimated assembly weight was calculated as

$$W_{\text{total}} = W_s + W_c + W_E + W_n + W_{\text{saddles}}. \quad (23)$$

TABLE 3: Estimated vessel weight breakdown.

Component	Estimated Weight [lb]
Cylindrical shell	15722.88
Conical head	3372.60
2:1 ellipsoidal head	2018.19
Nozzles	1290.00
Saddle supports	380.00
Total	22783.67

5. SADDLE SUPPORTS

Detailed saddle support stress calculations were not required for this project. However, saddle supports were included in the CAD model because horizontal pressure vessels are commonly supported by two saddles. The saddle geometry was selected from the standard saddle dimensions provided in Chapter 4 of the *Pressure Vessel Design Manual*. If the calculated vessel outside diameter fell between two listed values, the next larger outside diameter was selected for a conservative preliminary CAD design. The saddle angle was selected within the recommended range of 120 deg to 168 deg.

The saddle weight was included in the total vessel weight and cost estimate. Table 2-20 of the *Pressure Vessel Design Manual* gives the approximate combined weight of two carbon steel saddles and baseplates as a function of vessel diameter [7]. For a vessel inner diameter of 48 in, the table gives a combined saddle weight of

$$W_{\text{saddles}} = 380\,\text{lb}. \tag{24}$$

The reference table used for this estimate is included in Appendix A.

6. MANUFACTURING COST ESTIMATION

6.1. Cost Model

The manufacturing cost was estimated using the total vessel weight and a lumped fabrication cost per pound. The raw SA-516 Grade 70 steel plate cost was first estimated using a quoted material cost of approximately \$800 per ton:

$$C_{\text{steel}} = \frac{\$800}{2000\ \text{lb}} = \$0.40/\text{lb}. \tag{25}$$

The raw material cost was calculated as

$$C_{\text{raw}} = W_{\text{total}} C_{\text{steel}}. \tag{26}$$

Using $W_{\text{total}} = 22\,783.67$ lb, the raw material cost is

$$C_{\text{raw}} = \$9,113.47. \tag{27}$$

A lumped manufacturing cost of \$5/lb was then used to estimate the total vessel cost. This estimate includes material procurement, forming, welding, nozzle fabrication, saddle fabrication, inspection, testing, and general fabrication labor:

$$C_{\text{total}} = W_{\text{total}} C_{\text{mfg}}, \tag{28}$$

where $C_{\text{mfg}} = \$5/\text{lb}$. The resulting total manufacturing cost is

$$C_{\text{total}} = \$113,918.33. \tag{29}$$

Detailed cost calculation steps and unit conversions are provided in Appendix A.

TABLE 4: Cost estimate summary.

Cost Item	Estimated Cost [\$]	Basis
Raw SA-516 Gr. 70 material	9,113.47	\$0.40/lb
Total manufacturing estimate	113,918.33	\$5/lb

6.2. Cost Discussion

The raw material cost represents only the steel plate cost and is much lower than the total manufacturing cost because it does not include forming, welding, machining, nozzle fabrication, saddle fabrication, inspection, testing, or labor. The lumped manufacturing estimate of \$5/lb is therefore used as the final cost estimate for the vessel. The total cost is primarily driven by the vessel weight, which depends on the required shell and head thicknesses. Additional cost drivers include the use of two different head geometries, the six pressure-rated nozzles, saddle support fabrication, weld inspection, and hydrostatic testing.

7. TESTING AND INSPECTION REQUIREMENTS

After fabrication, the vessel must be tested and inspected to verify that the manufactured pressure boundary is free from critical defects. Finite element analysis evaluates the idealized design geometry, but it does not guarantee that weld defects, surface cracks, lack of fusion, porosity, or fabrication damage are absent in the completed vessel. Therefore, hydrostatic testing and nondestructive examination are included in the evaluation plan.

7.1. Dye Penetrant Testing

Dye penetrant testing may be used to detect surface-breaking flaws such as cracks, seams, laps, and porosity near welds and nozzle attachments. The surface is cleaned, penetrant is applied, excess penetrant is removed, and a developer draws trapped penetrant out of surface defects. This method is useful for accessible weld regions because it provides a simple visual indication of surface flaws.

7.2. Spot Radiography

Spot radiography is used to inspect selected welded joints using X-rays or gamma rays. Since welded joints are generally treated as less efficient than seamless parent metal, radiographic examination helps verify weld quality and supports the selected joint efficiency used in the design calculations. For ASME Section VIII, Division 1 vessels, spot examination of welded joints is associated with UW-52 requirements [14].

7.3. Acoustic Emission Testing

Acoustic emission testing may be used as a nondestructive method for detecting active crack growth or structural degradation while the vessel is under load. Sensors mounted on the vessel surface detect transient elastic waves produced by active flaws. This method is useful because it can monitor the material response during stressing and can help identify growing damage in real time [12].

7.4. Hydrostatic Testing

Hydrostatic testing verifies the pressure-boundary integrity of the vessel by filling it with liquid and pressurizing it above normal operating pressure. For ASME Section VIII, Division 1 vessels, hydrostatic testing is commonly associated with UG-99 requirements [13]. During the test, the vessel is monitored for leakage, pressure loss, or visible deformation. Because liquid is nearly incompressible compared with gas, hydrostatic testing is generally preferred over pneumatic testing when practical.

A typical hydrostatic test pressure can be expressed as

$$P_{\text{test}} = 1.3\,\text{MAWP}\left(\frac{S_T}{S_D}\right), \tag{30}$$

where S_T is the allowable stress at test temperature and S_D is the allowable stress at design temperature.

7.5. Nondestructive Examination

The evaluation plan may include:

- visual inspection of welds,
- radiographic testing or ultrasonic testing of major seams,
- magnetic particle or liquid penetrant testing of nozzle and attachment welds,
- hardness testing after welding or heat treatment if required,
- dimensional inspection of shell roundness, nozzle location, and saddle fit-up.

8. CAD MODEL AND DRAFTING SHEET

8.1. CAD Model Description

The CAD model was created in OnShape. The model includes the cylindrical shell, conical head, 2:1 ellipsoidal head, simplified nozzles, and two saddle supports. The full assembly CAD views are shown in Figs. 1 and 2. Separate component views of the vessel body, nozzle, and saddle support are shown in Figs. 3, 4, and 5, respectively.

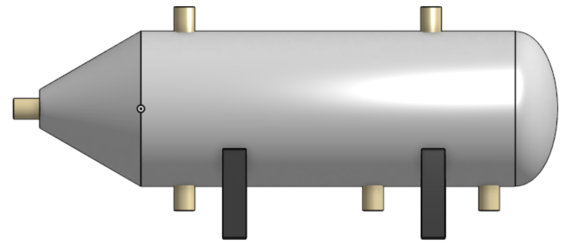

FIGURE 1: Horizontal assembly view of the pressure vessel CAD model.

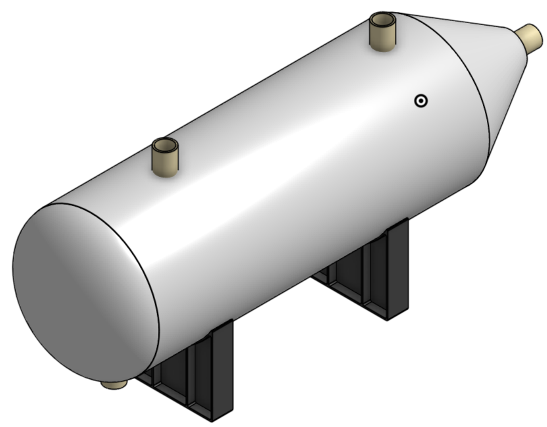

FIGURE 2: Isometric assembly view of the pressure vessel CAD model.

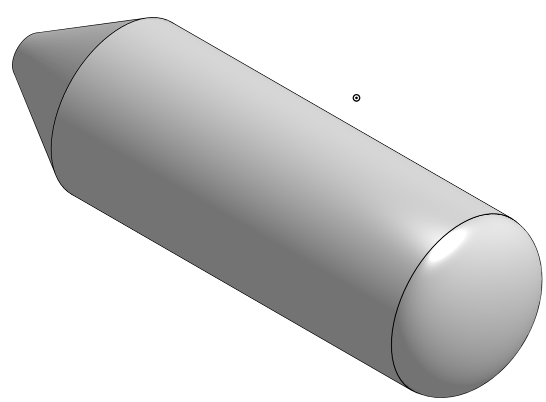

FIGURE 3: Separate CAD view of the pressure vessel body, including the cylindrical shell, conical head, and 2:1 ellipsoidal head.

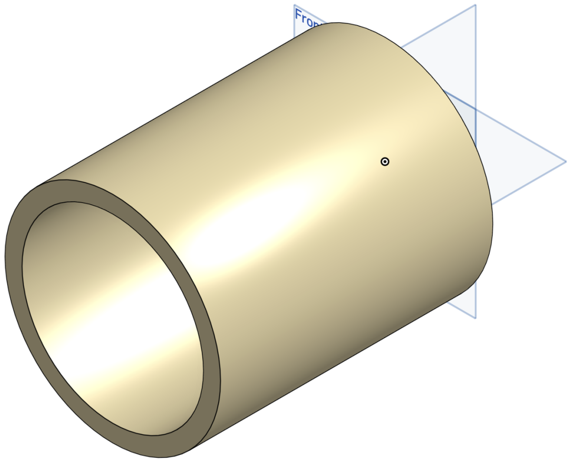

FIGURE 4: CAD detail view of the simplified nozzle geometry.

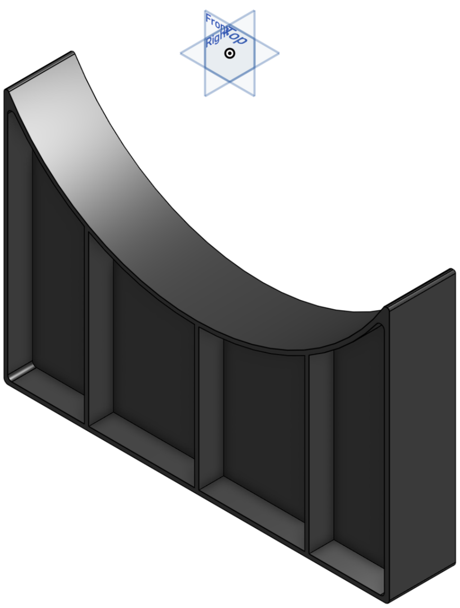

FIGURE 5: CAD detail view of the saddle support geometry.

8.2. Drafting Sheet

A detailed drafting sheet was created to document the main vessel dimensions, head geometry, nozzle locations, saddle support placement, and material information. The full drawing is provided in Appendix B.

9. STRUCTURAL ANALYSIS

The structural analysis was limited to the pressure-retaining shell-and-head assembly. Saddle support stresses and nozzle reinforcement stresses were not evaluated because the project requirement focuses primarily on the internal pressure stresses in the cylindrical shell and heads. The finite element model applies the design pressure to the internal surfaces of the shell and heads and evaluates the resulting stress and displacement fields.

9.1. Boundary Conditions and Loads

TABLE 5: Finite element model boundary conditions and loads.

Item	Description
Geometry analyzed	Cylindrical shell and heads
Internal pressure	1500 psi internal surfaces
Material model	Linear elastic SA-516 Grade 70
Excluded from analysis	Nozzles and saddle supports
Primary output	Von Mises stress and displacement
Acceptance basis	Compare with allowable/yield stress

9.2. FEA Results

The finite element analysis was performed in Fusion for the pressure-retaining shell-and-head assembly under the design internal pressure of 1500 psi. The model used a linear elastic steel material model with a yield strength of 207 MPa. The mesh contained 15,674 nodes and 7,333 elements. The nozzles and saddle supports were not included in the stress evaluation because the project scope focused on the cylindrical shell and heads. The resulting stress, displacement, and safety factor plots are shown in Figs. 6–8.

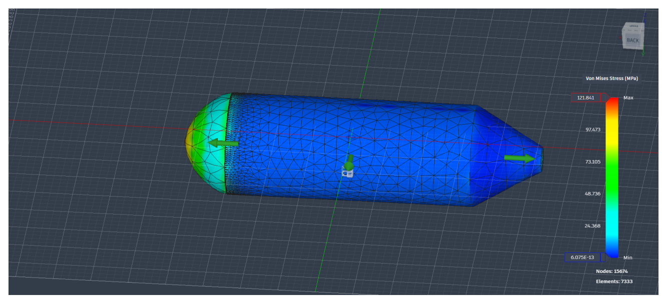

FIGURE 6: Von Mises stress distribution from finite element analysis.

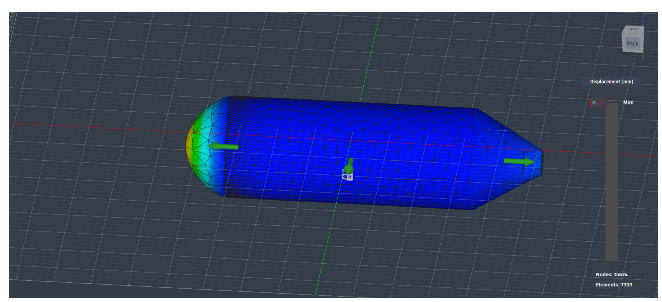

FIGURE 7: Total displacement distribution from finite element analysis.

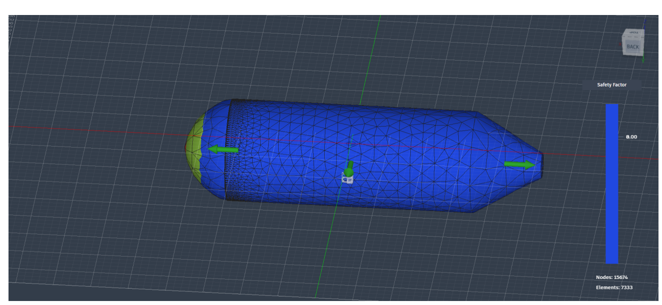

FIGURE 8: Factor of safety distribution from finite element analysis.

TABLE 6: Summary of structural analysis results.

Result	Value
Max. von Mises stress	121.841 MPa
Max. displacement	0.486 mm
Min. factor of safety	1.699
Min. 3rd principal stress	−34.611 MPa
Mesh size	15,674 nodes / 7,333 elements
Critical stress region	Shell-to-head transition

The maximum von Mises stress was 121.841 MPa, which is below the material yield strength of 207 MPa. This gives a minimum factor of safety of approximately 1.699 based on yielding. The maximum displacement was 0.486 mm, which is small compared with the overall vessel diameter and length. Therefore, the preliminary FEA result indicates that the shell-and-head assembly remains within the elastic range under the specified internal pressure.

The highest stresses occur near geometric transition regions, especially where the shell connects to the heads. This behavior is expected because shell-to-head junctions introduce changes in curvature and stiffness, which can create localized stress concentrations. The conical head region is also expected to show higher stress than the straight cylindrical shell because of the geometry change.

10. RESULTS AND DISCUSSION

10.1. Analytical Design Results

The analytical calculations determined a required cylindrical shell thickness of 2.576 in. Including the corrosion allowance of 0.125 in, the total shell thickness used for the design was 2.701 in. The required 2:1 ellipsoidal head thickness was 2.445 in, giving a total ellipsoidal head thickness of 2.570 in after corrosion allowance. The required conical head thickness was 2.975 in, giving a total conical head thickness of approximately 3.100 in after corrosion allowance.

Using these calculated thicknesses, the estimated component weights were 15 722.88 lb for the cylindrical shell, 3372.60 lb for the conical head, and 2018.19 lb for the 2:1 ellipsoidal head. Including the estimated nozzle and saddle weights, the final total vessel weight was estimated as 22 783.67 lb. The corresponding lumped manufacturing cost estimate was $113,918.33.

10.2. Comparison Between Analytical and FEA Results

The analytical calculations determined the required shell and head thicknesses using pressure-vessel design equations, while the FEA model evaluated the stress distribution of the shell-and-head assembly under internal pressure. The analytical method is used as the primary sizing method, and the FEA result is used as a numerical check on stress behavior.

The FEA result showed a maximum von Mises stress of 121.841 MPa. This is close to the allowable stress value used in the hand calculations, 17 500 psi, or approximately 120.7 MPa, but it remains below the material yield strength of 207 MPa. The small difference between the FEA maximum stress and the analytical allowable stress is reasonable because the FEA captures localized stresses near geometric transitions, while the hand calculations are based on simplified shell and head pressure formulas.

Overall, the FEA supports the analytical design by showing that the pressure-retaining components remain below yield under the design pressure. However, because the maximum stress occurs near a transition region and a mesh convergence study was not performed, the result should be treated as preliminary rather than a final code-certified stress evaluation.

10.3. Design Tradeoffs

The major design tradeoffs include:

- increasing thickness improves strength but increases weight and cost,
- hemispherical heads reduce stress but are more expensive to manufacture,
- ellipsoidal heads provide a practical balance between strength and manufacturability,
- conical heads may be simpler for some geometries but can introduce high discontinuity stresses,
- nozzle reinforcement improves local strength but increases welding and inspection requirements,
- saddle placement affects longitudinal bending and local shell stress.

10.4. Potential Design Change

The chosen vessel design possesses one 2:1 ellipsoidal head and one conical head. If we were to replace the conical head with another 2:1 ellipsoidal head, it would be possible to reduce costs. Using the calculated weights for each and the overall cost per lb of raw steel, we find that one conical head and one ellipsoidal head weigh a total of 5309.79 lb. This gives an estimated raw price of $2156.32. On the other hand, two 2:1 ellipsoidal heads would weigh a total of 4036.38 lb and cost $1614.55. This is a decrease in weight for the overall vessel of 1354.41 lb and a decrease in raw cost of $541.76, or 6%. However, this percentage decrease in raw cost must be compared to the potential decrease in vessel integrity, given the analytical conclusion that the ellipsoidal head is the vessel's weak point.

11. CONCLUSION

This project developed a preliminary design of a horizontal pressure vessel for use as a three-phase oil–gas–water separator. The main analytical design focused on the cylindrical shell, conical head, and 2:1 ellipsoidal head under an internal design pressure of 1500 psi. The calculated shell thickness including corrosion allowance was 2.701 in, while the total conical and ellipsoidal head thicknesses were approximately 3.100 in and 2.570 in, respectively. The total estimated vessel weight, including simplified nozzles and saddle supports, was 22 783.67 lb, and the total manufacturing cost was estimated as $113,918.33.

A CAD model was created in OnShape to represent the shell, heads, simplified nozzles, and saddle supports. The structural analysis was performed on the pressure-retaining shell-and-head assembly under internal pressure. The maximum von Mises stress from FEA was 121.841 MPa, which is below the material yield strength of 207 MPa, giving a minimum factor of safety of 1.699. The highest stresses occurred near shell-to-head transition regions, which is expected because changes in curvature and geometry create local stress concentrations. Overall, the design satisfies the project objectives by combining analytical pressure-vessel sizing, weight estimation, cost estimation, CAD modeling, and preliminary finite element evaluation.

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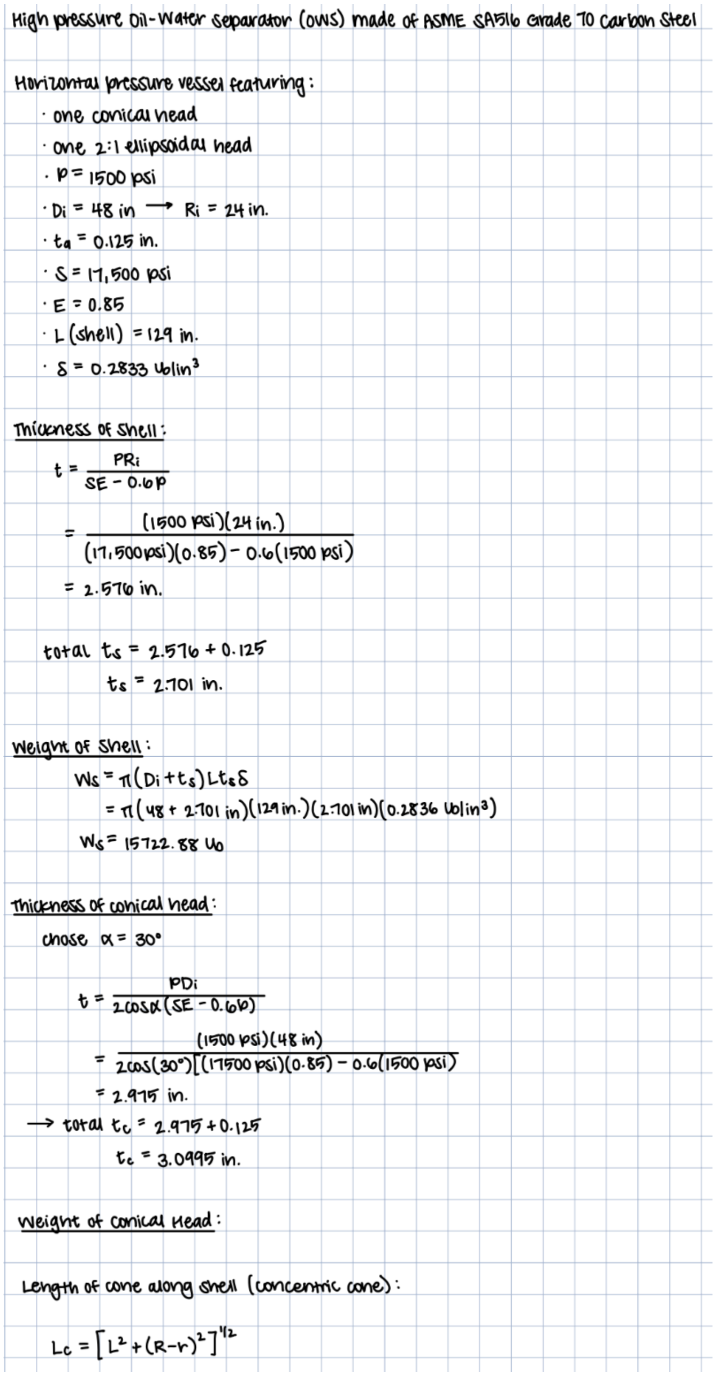
APPENDIX A. HAND CALCULATIONS

Detailed hand calculations for the shell thickness, shell weight, conical head thickness, conical head weight, 2:1 ellipsoidal head thickness, ellipsoidal head weight, nozzle weight estimate, saddle weight estimate, and total vessel weight are provided in Figs. 9–11.



High pressure Oil-Water Separator (OWS) made of ASME SA516 Grade 70 Carbon Steel

Horizontal pressure vessel featuring:
- one conical head
- one 2:1 ellipsoidal head
- $P = 1500$ psi
- $D_i = 48$ in $\longrightarrow$ $R_i = 24$ in.
- $t_a = 0.125$ in.
- $S = 17{,}500$ psi
- $E = 0.85$
- L (shell) $= 129$ in.
- $\delta = 0.2833$ lb/in^3

Thickness of Shell:

$$t = \frac{PR_i}{SE - 0.6P}$$

$$= \frac{(1500\text{ psi})(24\text{ in.})}{(17{,}500\text{ psi})(0.85) - 0.6(1500\text{ psi})}$$

$$= 2.576\text{ in.}$$

total $t_s = 2.576 + 0.125$

$t_s = 2.701$ in.

Weight of Shell:

$$W_s = \pi(D_i + t_s)Lt_s\delta$$

$$= \pi(48 + 2.701\text{ in})(129\text{ in.})(2.701\text{ in})(0.2836\text{ lb/in}^3)$$

$$W_s = 15722.88\text{ lb}$$

Thickness of conical head:

chose $\alpha = 30°$

$$t = \frac{PD_i}{2\cos\alpha(SE - 0.6P)}$$

$$= \frac{(1500\text{ psi})(48\text{ in})}{2\cos(30°)[(17500\text{ psi})(0.85) - 0.6(1500\text{ psi})]}$$

$$= 2.975\text{ in.}$$

$\rightarrow$ total $t_c = 2.975 + 0.125$

$t_c = 3.0995$ in.

Weight of Conical Head:

Length of cone along shell (concentric cone):

$$L_c = [L^2 + (R-r)^2]^{1/2}$$

FIGURE 9: Hand calculations for shell thickness, shell weight, and conical head thickness.

$$L = [0.5(D_i - d_i)]\tan\alpha$$
$$= [0.5(48-6)]\tan(30°)$$
$$= 12.124 \text{ in.}$$

$$L_c = [(12.124 \text{ in})^2 + (24 - 3 \text{ in})^2]^{1/2}$$
$$L_c = 24.25 \text{ in.}$$

$$W_c = A_c t_c \delta$$

$$A_c = \pi[0.5(D_i + d_i)][L_c^2 + (0.5(D_i - d_i)^2]^{1/2}$$

From OWS chart, inlet/outlet (nozzle) diameter is 6 in. for the 1,000 model.
↳ use 6in. for d_i

$$A_c = \pi[0.5(48+6)][24.249^2 + (0.5(48+6)^2)]^{1/2}$$
$$= 3836.78 \text{ in}^2$$

$$W_c = (3836.78 \text{ in}^2)(3.0995 \text{ in})(0.2836 \text{ lb/in}^3)$$
$$W_c = 3372.6 \text{ lb}$$

Thickness of 2:1 Ellipsoidal Head:

$$t = \frac{PD_i}{2SE - 0.2P}$$
$$= \frac{(1500 \text{ psi})(48 \text{ in})}{2(17500 \text{ psi})(0.85) - 0.2(1500 \text{ psi})}$$
$$= 2.445 \text{ in.}$$

→ total $t_E = 2.445 + 0.125$

$$t_E = 2.5698 \text{ in.}$$

Weight of 2:1 Ellipsoidal Head

$$W_E = 1.084(D_i + t_E)^2 t_E \delta$$
$$= 1.084(48 + 2.5698 \text{ in.})^2(2.5698 \text{ in.})(0.2833 \text{ lb/in}^3)$$
$$W_E = 2018.19 \text{ lb}$$

Weight of Nozzles:

Table 2-14 in the design manual gives the weights of nozzles
From the OWS sizing chart, the inlet & outlet diameter is 6 in., which equals nozzle diameter.

For $p = 1500$ psi & rating size = 6", table 2-14 gives:

$W_n = 215$ lb
Our pressure vessel has 6 nozzles:
total $W_n = 6(215)$
$W_n = 1290$ lb.

FIGURE 10: Hand calculations for conical head weight, 2:1 ellipsoidal head thickness, ellipsoidal head weight, and nozzle weight estimate.

Weight of Saddles

Table 2-20 in the design manual gives us the total weight of 2 saddles for $D_i = 48$ in.

$W_{saddles} = 380$ lb

Total Weight

$$W_{tot} = W_S + W_C + W_E + W_n + W_{saddles}$$
$$= (15722.88 + 3372.6 + 2018.19 + 1290 + 380) \text{ lb}$$
$$\boxed{W_{tot} = 22783.67 \text{ lb}}$$

Cost of raw material:

Steel plates SA516 Gr 70 : $\frac{\$800}{\text{ton}} \cdot \left| \frac{1 \text{ ton}}{2000 \text{ lb}} \right| = \$0.4/\text{lb}$

$$\text{cost} = W_{tot} (\text{lb}) \times \$0.4/\text{lb}$$
$$= 22783.67 \text{ lb} \left(\frac{\$0.4}{\text{lb}} \right)$$

$$\text{cost} = \$9113.47$$

Total manufacturing cost:

$\text{cost} = \$5/\text{lb}$

$$\text{Total cost} = W_{tot} (\text{lb}) \times \frac{\$5}{\text{lb}}$$
$$= 22783.67 \text{ lb} (\$5/\text{lb})$$
$$\boxed{\text{Total cost} = \$113,918.33}$$

FIGURE 11: Hand calculations for saddle weight and total estimated vessel weight.

APPENDIX B. DRAFTING SHEET

The detailed pressure vessel drafting sheet is shown in Fig. 12. The drawing includes the overall vessel geometry, principal dimensions, nozzle locations, saddle supports, and material callouts.

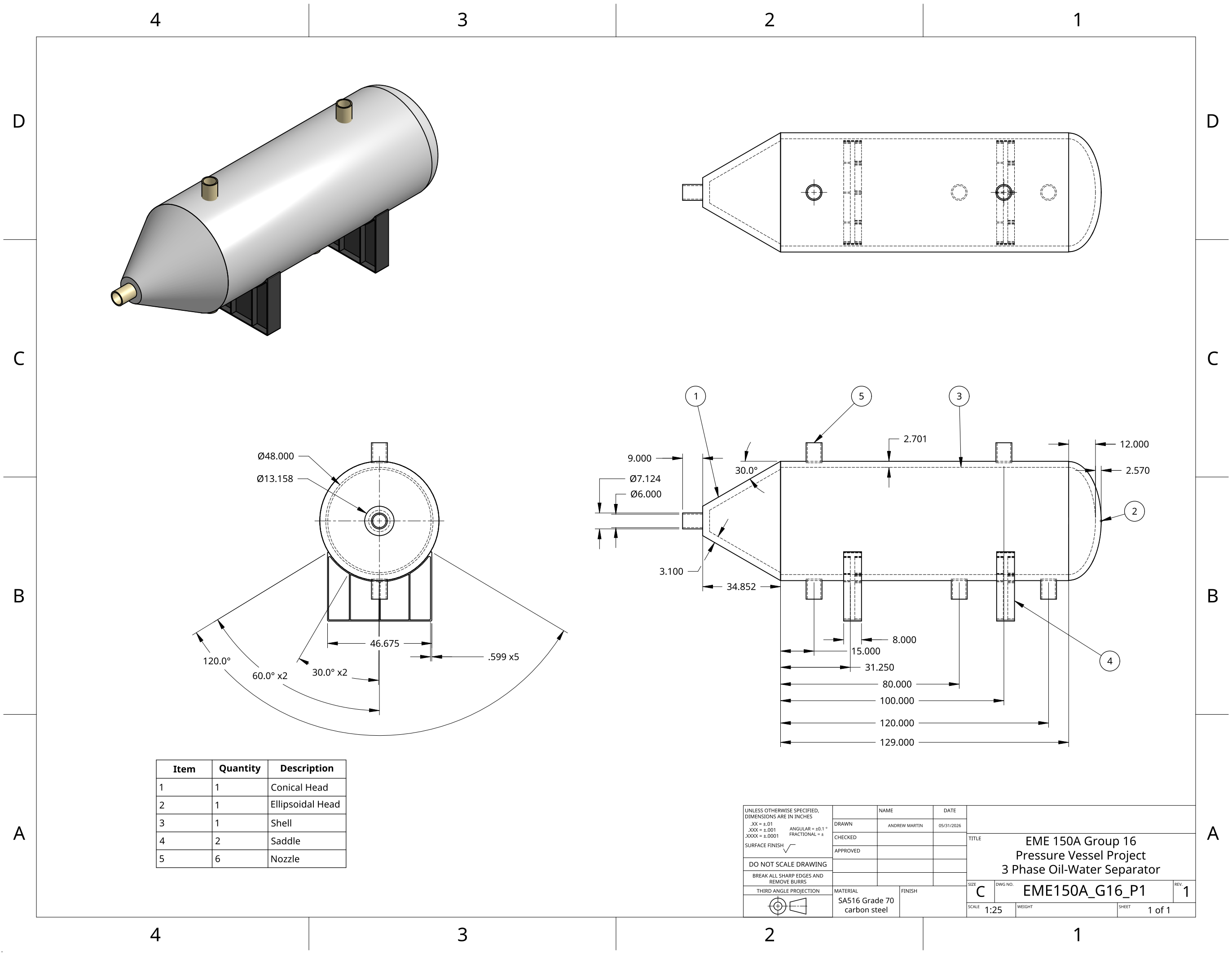


FIGURE 12: Detailed drafting sheet for the horizontal pressure vessel assembly.