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Research Paper

Designing And Analyzing An Elevated Atmospheric Tank: A Case Study

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ABSTRACT

The pressure vessels design process is a modular approach which is always advised and encouraged to comply by related codes and regulations to conform to a more uniform methodology. Such processes, in order to maintain the individual contributions and innovations, need to be monitored and modified regularly. Documented case studies of different, unique approaches, in compliance of the current existing codes and in a more academic context is one of the monitoring methods which could be explored. In this paper, design and analysis of a uniquely shaped atmospheric tank, in compliance of two suggested codes of ASME VIII Div.1 and API 620, is investigated and by implementing the design procedure, comparing both codes' criteria, a detailed and comprehensive approach is laid out for design engineers to explore the experimental code selection and design methodology for low pressure vessels applied by PV Elite software. A comparison of both intended codes was also carried out and applied for each step within the scope of the paper. The results demonstrated a more in-depth vision of both codes and their reliability through risk evaluation of design parameters.

Keywords: Design method; Elevated atmospheric tank; Standard codes; Pressure vessel.

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1 INTRODUCTION

WHEN selecting the appropriate standard code for pressure vessel design, recognized handbooks and codes often rely on the expertise and experience of the design engineer. It is generally assumed that an experienced engineer is best suited to analyze and apply the codes. This is because pressure vessels are inherently hazardous, and even with strict rules and considerations, their design and fabrication may be affected by unknown and uncontrollable conditions. Therefore, an informed and experienced human factor should be included in all aspects of the process. However, relying on a design engineer's experience as a qualitative characteristic may contradict the main purpose, which is to increase the reliability of the design. This is because quantitative measurements are typically more reliable when it comes to evaluating design reliability.

The issue of relying solely on the design engineer's experience is particularly prevalent when safety issues are less likely to occur. For instance, in the case of low-pressure vessels such as elevated atmospheric tanks that only contain drinking water, the rare probability of failure cases can cause even experienced design engineers to blindly follow the codes, disregarding basic engineering assumptions and the ultimate goal of providing an optimized and efficient design. This approach does not improve the chances of an optimized and safe design, which ultimately means that the failure probability of an elevated atmospheric water tank, no matter how rare, will remain the same. This contradicts the mission of standard codes, which is to maximize safety.

To develop more quantitative approaches that reduce the need for an experienced engineer's qualities, specific case studies and approaches should be documented and studied, particularly on how to choose an appropriate code when the designer is not bound by the employer or any entity. Access to different methods and judgment criteria could act as an instruction guideline that reduces the chances of failure cases. In this paper, we carry out a comparison of two codes, API 620 and ASME VIII Division 1, by studying the design and analysis procedure of an elevated atmospheric tank as a case study. The design was performed by an experienced team in compliance with ASME VIII Div.1 and then compared with API 620 instructions. The case study focuses on an atmospheric tank that contains drinking water and has a unique shape specified by the employer, which challenges the selection process of a reference code. This tank operates within the pressure scope criterion of both ASME VIII Div.1 and API 620, but its shape is not directly guided by either code, relying heavily on the designer's experience. Thus, this case study is the ideal example to address the issue of a lack of case studies to act as suggestive instruction guidelines. Furthermore, by considering a step-by-step design and analysis approach, we present the limitations and advantages of both codes.

The existing literature on the topic of this paper either proposes alternative standard codes or examines case studies and proposes solutions without delving into the design process in much depth.

The selection of an appropriate code for pressure vessel design is closely related to the overall design process. One common approach to pressure vessel design is to start by defining basic parameters, such as the pressure differential between inside and outside of the vessel, [1]. Another approach is to categorize pressure vessels into three divisions based on design pressure, ranging from 15 Psi to over 10,000 Psi, [2], or based on their applications, such as simple pressure vessels, gas cylinders, unfired pressure vessels, boilers, valves, pipework, and miscellaneous equipment, [3]. Each category corresponds to a specific set of standard codes for design and analysis.

To provide more quantitative approaches to pressure vessel design that reduce the need for experienced engineers, it is useful to adopt a recognized design engineering approach that optimizes the given objectives within partly conflicting constraints. The planning and design process can be divided into four main phases: planning and task clarification, conceptual design, embodiment design, and detail design, [4]. While these steps have been applied in this paper's design process, they have not been discussed in detail to avoid straying from the main topic.

This paper aims to provide a thorough investigation of the process of selecting an appropriate code for pressure vessel design. We begin by presenting the details of the case study and the relevant background information in the first section. The literature review is then presented, followed by the separate Design and Analysis sections. In the Design section, we outline a clear approach to demonstrate the levels of design, beginning with the specification of information, followed by the conceptual, embodiment, and detail design process for each level. Each level follows a unique methodology recommended by recognized industry handbooks. The Analysis section follows a similar structure, with each level of analysis defined and calculated for the specific case study. We use *PVElite* software to perform the analysis according to the standards of both ASME VIII div.1 and API 620.

In the Discussion section, we provide a systematic comparison of the applicability of each code for the case study, examining the essential criteria of each and comparing their instructions and conditions within the scope of the case study. After a qualitative comparison of quantified values, we condition the case study to the criteria of API 620 and analyze it using *PVElite* software. The end results are compared and a conclusive argument presented.

2 BACKGROUND

The elevated atmospheric tank described in this section was constructed to serve as an independent water supply for an industrial park. With a capacity of 600,000 liters and a height of 27 meters, the tank was a crucial infrastructure project for the developing area, which was expected to attract many companies from different industries. The decision-making process for the project involved various factors, including landscape, weather conditions, and city management.

The project was executed as an engineering, procurement, and construction (EPC) contract, with turnkey delivery. Several conditional criteria had to be taken into account during the design and construction phases. Firstly, the region was known to be windy, (wind speed of 120 km/h), requiring careful consideration of the structural design. Secondly, the tank's foundation had to be built on a specific site that required soil analysis to confirm its suitability. Thirdly, the tower's design was required to be iconic to represent the industrial park. Finally, the water stored in the tank was intended for drinking purposes, necessitating strict adherence to drinking water standards and the provision of adequate pressure to ensure reliable supply.

To ensure the necessary pressure difference (Head) for the water, a conical vertical-shaped tower with a skirt for footing was proposed, as is typical for water tank design. The initial parameters for the design were based on the required water capacity and the conditional criteria, and are shown in Table I. To carry out the basic and conceptual design calculations for each part, the engineering team chose to follow the standards of several institutes, including UBC for the foundation, ASTM for the vessel's body material, ASME Section VIII for the vessel, and AWS-D1.1 for connection and welding. Although API 620 is typically suggested for such projects, the team chose ASME Section VIII due to their experience and the reliability factors deemed necessary.

2.1 Literature review

The design and analysis of pressure vessels, particularly for atmospheric and elevated tanks, have garnered some research attention due to the critical role these vessels play in industrial applications. Various studies have explored the application of ASME codes, focusing on ensuring compliance with standard design protocols and evaluating structural stability. The use of finite element analysis (FEA) and code compliance in design processes has been a prominent focus, as seen in studies where ASME Section VIII is applied alongside FEA to assess pressure vessel thicknesses and structural integrity under different operational pressures, [5].

Research on the impact of environmental loads, such as extreme winds and seismic forces, highlights the importance of structural resilience in storage tanks. Studies have shown that elevated tanks face unique challenges due to environmental stresses, with vulnerabilities under high wind loads and seismic activity, necessitating careful design considerations for atmospheric storage tanks to prevent failure and containment loss, [6-7]. Furthermore, the seismic fragility of tanks supported on various structures has been widely investigated, with findings suggesting that tank shape and foundation design significantly influence seismic performance, [8].

The comparison of ASME Section VIII, Divisions 1 and 2, in the selection of pressure vessel design codes has also been a subject of analysis. Studies emphasize that the choice between these divisions often depends on factors beyond structural requirements, such as administrative and certification complexities, with detailed evaluations showing cost benefits of certain divisions under specific conditions, [9]. Likewise, for low-pressure vessels, the need for specialized codes has been argued due to the limitations of ASME codes for pressures below 15 psi, with researchers advocating for more flexible standards to accommodate these designs, [10].

The importance of shape and structural configuration in elevated water tanks has also been extensively discussed. Research comparing tank shapes under various loads concludes that circular designs tend to outperform other configurations under both seismic and wind forces, [11-12]. Additionally, spherical tanks have been studied for seismic resilience, with findings underscoring the need for distinct design considerations that factor in sloshing effects, as these can substantially impact structural responses during seismic events, [13].

Efforts to streamline the external pressure design process of cylindrical shells in pressure vessels, particularly through simplified calculations, have been proposed as alternatives to the iterative ASME approach. These methods, supported by analytical models, offer designers more accessible means of determining structural requirements without compromising on reliability, [14]. Moreover, studies focused on stress analysis and stress concentration factors reveal the challenges of maintaining structural integrity in vessels with openings or additional attachments, suggesting optimizations in material selection and structural geometry to mitigate these stresses, [15].

The complexity of pressure vessel failure mechanisms, especially for low-pressure tanks, has also been explored in depth. Studies document cases where failures were unrelated to the conventional risks of combustion or vacuum, broadening the understanding of potential hazards and underscoring the need for thorough hazard assessments in vessel design, [16]. These findings align with the goal of evaluating risk and reliability in pressure vessel designs, particularly in applications requiring strict adherence to code specifications for both safety and operational efficiency.

3 DESIGN APPROACH

Designing pressure vessels requires a comprehensive understanding of the different failure modes and types of corrosion, as well as the properties of materials used in pressure vessels. The choice of materials and the design considerations depend on the specific application and operating conditions, such as pressure, temperature, and exposure to corrosive environments. Carbon steel, low-alloy steel, and stainless steel are among the materials that can be used for pressure vessels, and designers must consider factors such as cost, corrosion resistance, and mechanical properties. In addition to material selection, designers must also account for potential fatigue failure, which can result from cyclic loading due to pressure fluctuations, temperature changes, or other environmental factors. To prevent fatigue failure, appropriate design features such as reinforcement at high-stress points and the use of appropriate welds must be incorporated, [3].

To facilitate the process designation of power and process plants, the widely-used Kraftwerk Kennzeichen system (KKS) plays a crucial role in equipment/spares inventories, Process and Instrumentation Diagrams (PIDs), and plant labeling. In the context of pressure vessel design, this paper proposes a four-level design process that aligns with the KKS classification system.

The four levels are level 0 for the operation site, level 1 for low-pressure/atmospheric tanks, level 2 for shell, head, and support, and level 3 for openings such as flanges, nozzles, plates, bolts, and piping, [3].The use of KKS allows for a better understanding of the design levels and methodology applied in this paper.

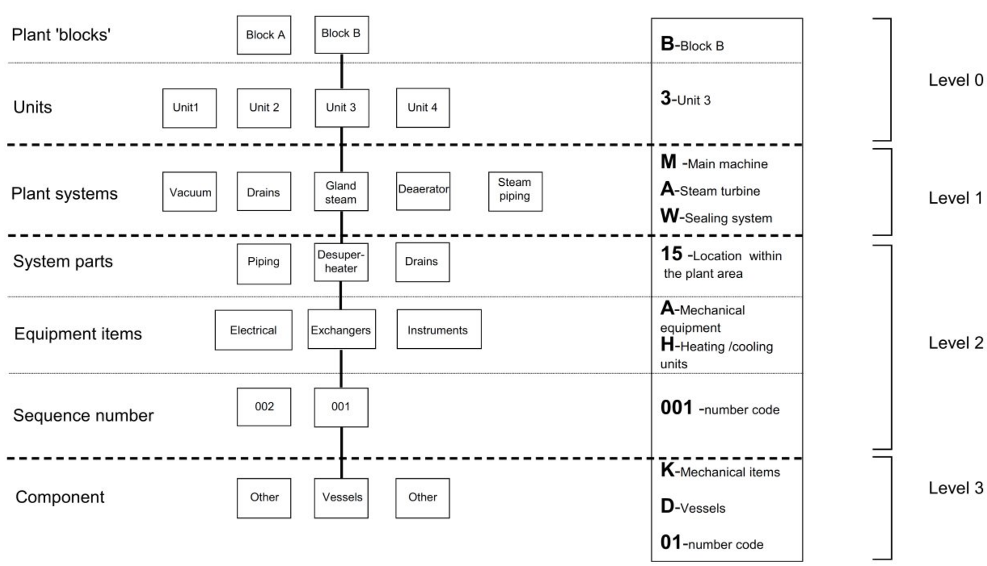

Fig. 1
Structure of the KKS plant classification system applied as the design approach.

3.1 Foundation Design

The design process of the storage tank started with the determination of the location of the tank and the analysis of the soil conditions in that area. The design parameters involved the soil sample analysis and foundation design, as well as the consideration of the geographical and regional seismic plans. For example, in the case of a vertical tank, the base plate is the critical stress point, and the foundation design must prioritize the tank's shape and the soil conditions. Design procedures were applied by using Plaxis software for modeling.

Once the soil conditions were assessed, the appropriate foundation type was selected. In this case, a pile type foundation was suggested in poor soils, where an elevated pile cap is used to allow air circulation under the tank, thus preventing the freezing of the soil and its upheaval, [17].

Atmospheric cylindrical tanks, which operate without pressure or with very little pressure, are generally cylindrical, perpendicular to the ground with a flat bottom and a fixed or floating roof. For this stage of design, a flat bottom conical tank was considered, [18].

For the purpose of foundation, there are three main components which are considered: Base plate, concrete and anchor bolts. The base plate and anchor design are investigated in the base plate section. Concerning the concrete, the initial design was based on a reinforced concrete design. The compressive strength of cylindrical specimens is used in the design and construction of structures, is also used to assess the quality and suitability of materials for specific applications. The mechanical specifications of these components are demonstrated in **Error! Reference source not found.**, so the load case calculations and design selection criteria are provided.

To ensure a safe and efficient foundation design, geotechnical studies were conducted to gather critical information about the soil properties in the area. Soil mechanics laboratory studies were conducted to determine the soil bearing capacity based on load. The modulus of subgrade reaction was calculated by applying a known load to a small area of soil and measuring the corresponding deflection of the soil. To determine the foundation bearing coefficient for the 900x900 foundation, calculations were applied. The laboratory provided a 76.2 cm diameter plate bearing coefficient, but it was not applicable for the foundation. The total load on the foundation was obtained by summing the structure's dead load, liquid weight filled in the tank, and foundation weight. The average stress on the foundation was calculated by dividing the total load by the foundation area. The soil settlement due to the plate's load was also determined, and the plate bearing coefficient was calculated accordingly (See **Error! Reference source not found.**). Finally, the foundation coefficient was calculated by dividing the plate bearing coefficient by a safety factor.

Table 1
Mechanical specifications of foundation material.

	Steel	Concrete	Reinforcement	Bolts
Yield strength	S_y=240 MPa	-	S_y=400 MPa	-
Tensile strength	S_u=360 MPa	-	S_u=1.25f_y kN/m^3	S_u=800-1000 MPa
Elasticity module	E_c=210 GPa	E_C=25 GPa	E_c=200 GPa	E_c=210 GPa
Shear modulus	G=81 GPa	-	-	G=81 GPa
Poisson's ratio	V_C=0.3	V_C=0.2	V_C=0.3	V_C=0.3
thermal expansion and contraction coefficient	α=12x10^{-6} C^{-1}	α=1x10^{-5} C^{-1}	-	α=12x10^{-6} C^{-1}
compressive strength of cylindrical specimen	-	S'_C=25 MPa	-	-
unit weight of volume	-	Y_y=25 kN/m^3	-	-

Table 2
Foundation coefficient calculations based on deadload.

Fabricated Steel structure (Weight)	Estimated based on initial basic design parameters	73.75 Tons
Liquid weight filled in the tank - Shop test + Water(full)	Estimated based on initial basic design parameters	671.87 Tons
Foundation weight	W=V x 2.4=A x B x H x 2.4=9.0 x 9.0 x 1.2 x 2.4	233.28 tons
Total load on the foundation	$\sum w_i$	978.81 tons
Average stress on the foundation	$\sigma_{plate} = \frac{w_i}{A} = \frac{978.81}{9 \times 9}$	$1.208 \frac{kg}{Cm^2}$
Soil settlement due to plat's load	δ	0.258 cm
Plate bearing coefficient	$k_{plate} = \frac{1.208}{0.258}$	$4.682 \frac{kg}{cm^3}$
foundation coefficient	$k_{foundation} = \frac{4.682}{5.0}$	$0.936 \frac{kg}{cm^3}$

In the design of a vessel foundation, pile design and analysis are critical steps in ensuring that the structure has a solid foundation capable of supporting its weight. In this case study, the ALL-Pile VER.6 software was used to calculate the pile loading capacity in both tension mode and under pressure. The software considers various factors such as soil layers and profile, as well as the disposition of the piles to determine the maximum load capacity of the piles, which is crucial for further analysis. To determine the maximum load capacity of the piles, the CSI-COL software was utilized. This software analyzed a concrete pile section with a diameter of 90 cm and eight reinforcements of type AIII-25. The load-displacement curve and the maximum axial load of a pile were displayed in a graph, along with a three-dimensional plot of anchor and the maximum axial load of the pile. The analysis also included the control of pile settlement to ensure the stability of the structure.

Furthermore, the bending anchor and rebar calculations were vital in the pile design and analysis. The bending anchor is a crucial component for piles that require lateral support. The rebar calculations, on the other hand, determine the number and spacing of reinforcement bars in the pile. The control of the punching shear in a pile is another crucial aspect to prevent the failure of the pile under sudden load conditions.

3.2 Low pressure/atmospheric tanks Design

The design of the atmospheric tank to provide 600 m^3 of water for the industrial town was initially based on determining the optimal shape and estimated height required for the tank. The resulting design featured a skirt and conical head to maximize the tank's structural stability. Given the height of the water column, the design team opted for a low-pressure/atmospheric tank framework that adhered to the guidelines set out in ASME SECTION IIIV – DIV.1. The design team applied two primary considerations suggested by [3], (Matthews, 2001) for vessel design, which are detailed in the following section. Also, a third governing factor was also taken into account in order to cover all aspects of the design.

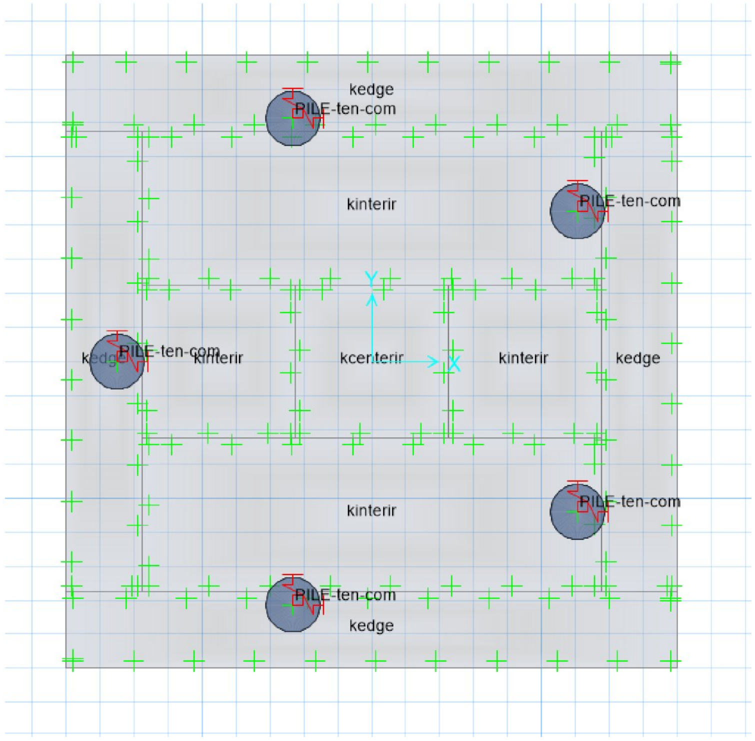

Fig. 2
The plan view of a foundation zoning determining the soil modulus factor.

3.2.1 Design Factors

The optimum L/D (length to diameter) ratio for a vessel is determined by the question of what vessel proportions will give the lowest weight for a given volume. The maximum volume for the minimum surface area, and weight, is a sphere, but spheres are generally more expensive to build.

To consider the shape of the cone to be ellipsoidal or torispherical, ASME VIII Div.1 has not studied the difference between ellipsoidal and equivalent torispherical heads in detail, therefore by referring to [19], and the comparison made on the elastic–plastic behaviors as their failure modes, i.e., buckling and plastic collapse, ellipsoidal shape was chosen.

The estimated optimum L/D ratio for the given vessel was calculated using Optimum Vessel Size criteria in [2], resulting in a diameter of 6 m and a length of 27 m. However, it is impossible to determine exactly what proportions will yield the lowest overall cost, since there are many more variables that enter into the ultimate cost of a vessel, [20].

For the design of the vertical tower with a water capacity of 600 m^3, the maximum allowable working pressure (MAWP) at the base plate needed to be determined to calculate the required thickness (as calculated in section 3.2.3). It's been suggested by [14], that External Pressure Design of Shells should be followed by more direct approach rather than iterative and tedious outline of ASME VIII div.1, but not to stray from the scope of this paper, Based on the Optimum Vessel Size approach of , [2], the optimal vessel size was then calculated. See Eq. (1).

$$F = \frac{P}{CSE}; \tag{1}$$

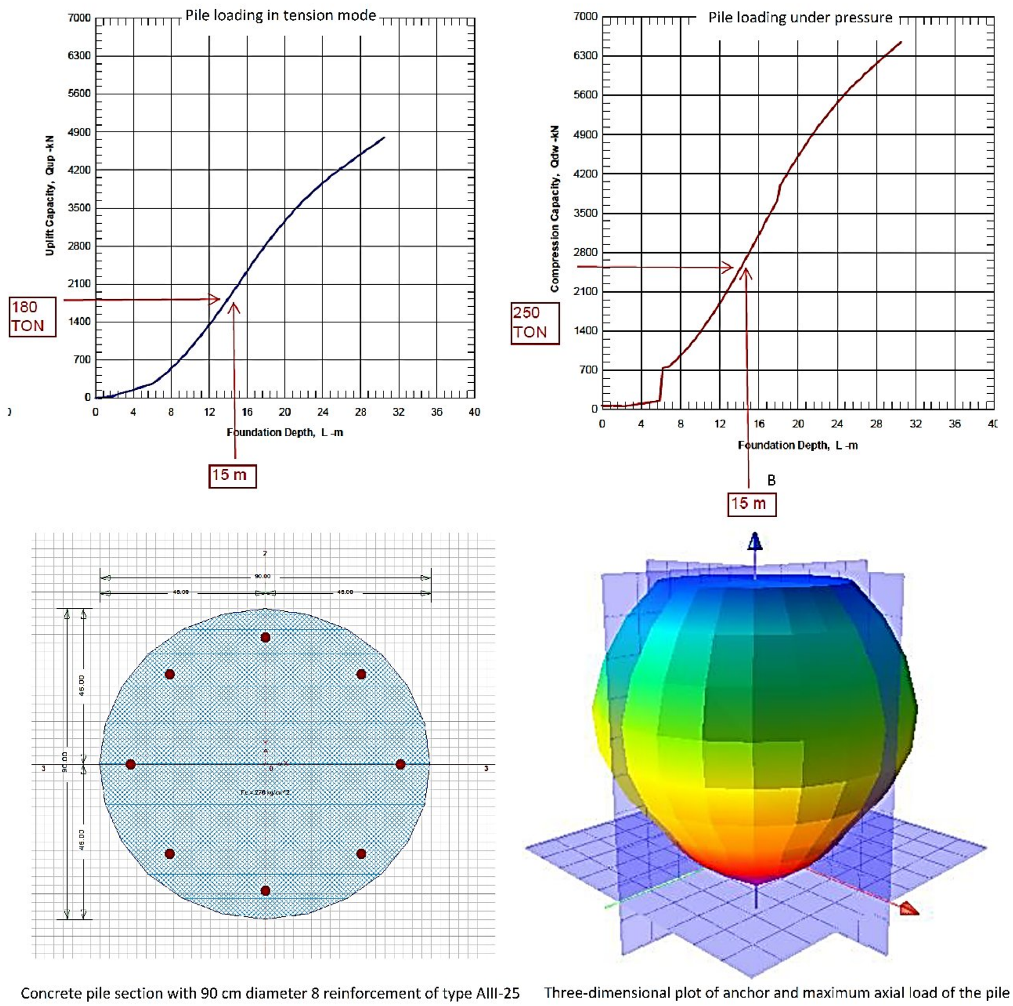

Fig. 3
Pile design and analysis's modeling and calculations.

In which P (Design pressure) is 0.245 MPa, C (corrosion allowance) and E (Joint Efficiency) are given (in section 3.2.2), S (Stress value of the material) is 260 MPa (Refer to section 3.2.3). Using the calculated F value and the sizing chart in Optimum Vessel Size approach, [2], a diameter of 6 m and length of 27 m were estimated. Modified dimensions were proposed based on the employer's initial criteria for the shape.

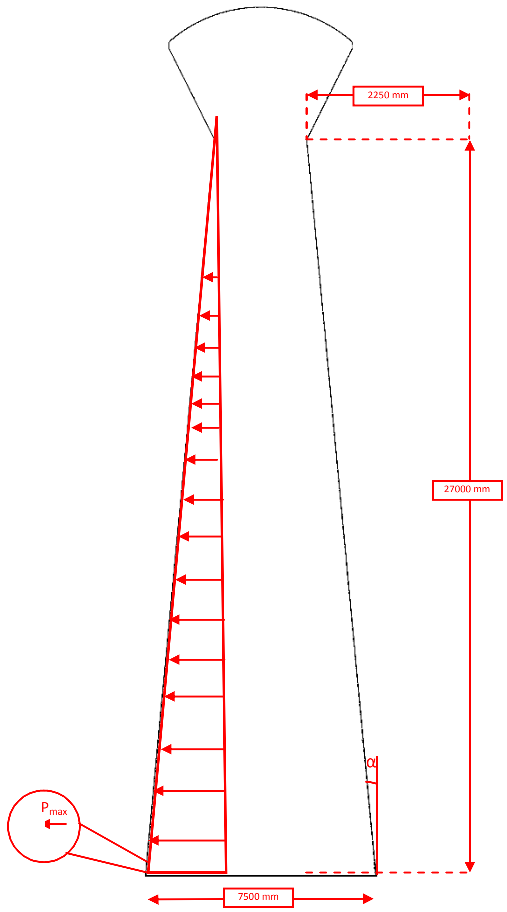

Fig. 4
The optimized vessel dimensions in the stage of basic design.

3.2.2 Safety factors

In pressure vessel design, safety factors are largely determined by considerations of corrosion and joint efficiency. To extend the lifespan of vessels exposed to corrosion, erosion, or abrasion, the vessel wall thickness is typically increased beyond the minimum specified by design formulas, or other protective measures are implemented. According to Code UG-25b, no standard corrosion allowance is mandated except for vessels with a thickness under 6.35 mm, for which an allowance of at least one-sixth of the minimum thickness is required. For vessels with a known corrosion rate, the desired lifespan informs the corrosion allowance; a rate of 0.127 mm per year (equating to approximately 1.524 mm over 12 years) is generally deemed adequate for vessels and piping. When corrosion rates are indeterminate, the designer's expertise guides the allowance. Major vessels are often designed for 15-20 years, while minor vessels are designed for 8-10 years, [2].

In this case study, with the vessel containing water rated as highly corrosion-resistant to metallic materials, [21], and a target lifespan of 15-20 years, a corrosion allowance of 3 mm was applied.

Joint welding methods in vessel construction are influenced by the welding context, Code regulations, and economic factors. Accessibility, for instance, affects welding feasibility; manual welding is unfeasible for small-diameter vessels, and backing strips are retained when applied. For large vessels lacking manholes, the final joint may require external-only welding. Code stipulations, such as those in UW-27, dictate acceptable joint types, their efficiencies, and designs based on service conditions and materials. Where flexibility exists, cost-effective welding options are preferred, such as V-edge torch cutting over J or U preparations, [2].

In this study, based on the vessel's dimensions and UW-12 requirements, limitations were dictated solely by Code and economic considerations. A butt joint, achieved via double-welding or equivalent techniques to ensure weld quality on both interior and exterior surfaces, was selected per UW-12, with a Type B examination method (spot examination) yielding a welded joint efficiency of 0.85.

3.2.3 Governing Factors

Based on the specifications of the tall tower described in, [2], the governing factors that determine the minimum required thickness were investigated by two main criteria.

Maximum stress theory: For analyzing the strength of tall towers under various loadings, the maximum stress theory has been applied. Considering the maximum applied pressure, the thickness required for the tower, according to ASME IIIV -Div. 1, for conical shells and body, [2]. By considering the maximum applied pressure on the base plate with a 25 meters column of water which was resulted to be 0.245 MPa.

$$P_{max} = \rho g h; \tag{2}$$

Regarding the materials selection, SA-516 Grade 70 and SA-283 Grade C were chosen based on the process engineer team's criteria, which tower's application and the operating pressure of the tower, approved by the design team, would put its classification in 15 psi (1 bar) working pressure. Calculating the minimum thickness based on ASME VIII Div.1, [2], suggested the formula for conical section.

$$t = \frac{\rho D}{2\, Cos\alpha\, (SE - 0.4P)}; \tag{3}$$

Considering that D represents the base plate outside diameter, α was calculated based on geometrical dimensions (See **Error! Reference source not found.** - The optimized vessel dimensions in the stage of basic design), Also S, yield strength of the selected material was 260 MPa. Therefore, a minimum thickness of 4.1 mm was calculated. With consideration of reliability factor, 170 MPa yield strength was suggested which resulted in 6.3 mm for the minimum thickness required at the base plate section.

Wind load: In the design of a tall vertical tower, wind load is also critical factor that must be considered in the calculations. The first step of design involves following the instructions of the ASCE-02 Standard to calculate the required thickness, stress, moment, and wind force, [2]. In this case, the tower's shape is double conical, with a diameter of 3 meters at first cone section and a height of 27 meters. Using these parameters and the ones provided from ASCE-02, the wind load was calculated to be 101.85 kN, the wind pressure 203.76 kPa, and the moment 1222.010 kilo joules (See **Error! Reference source not found.**). After analyzing the structure based on UBC-97, it was concluded that the minimum required thickness is close to 1 millimeter, which ensures the safety of minimum required thickness designed by Maximum stress theory.

Considering the weight and financial optimization, it was decided to use increasing plate thickness sizes from top to bottom, applying the minimum thickness required to the top. For instance, in exteriors of rockets and missiles or pressure vessels, the existence of pressure gradient along the longitudinal direction of the cylinder makes the engineers use cylindrical shell with variable thickness, [22]. For fabrication and installation purposes, the vessel shell and head were divided into 4 body modules, 1 head module, and 1 overhead.

The thickness starts at 15 mm at the base and decreases to 8 mm toward the head module. The overhead module, including the cone section, requires a minimum thickness of 7.5 mm to withstand external forces, as recommended by PV ELITE and calculated per UG-33(f) of ASME VIII (Cone calculations). Unlike the rest of the tower, which has a progressively decreasing thickness, the cone section also needs to meet seismic load requirements discussed later in Section 4-2, Vessel Analysis, and in Table 9. To ensure adequate strength under both external and seismic forces, a final thickness of 10 mm was selected for the cone section module (See **Error! Reference source not found.** - Shell thickness modular design section in the basic design).

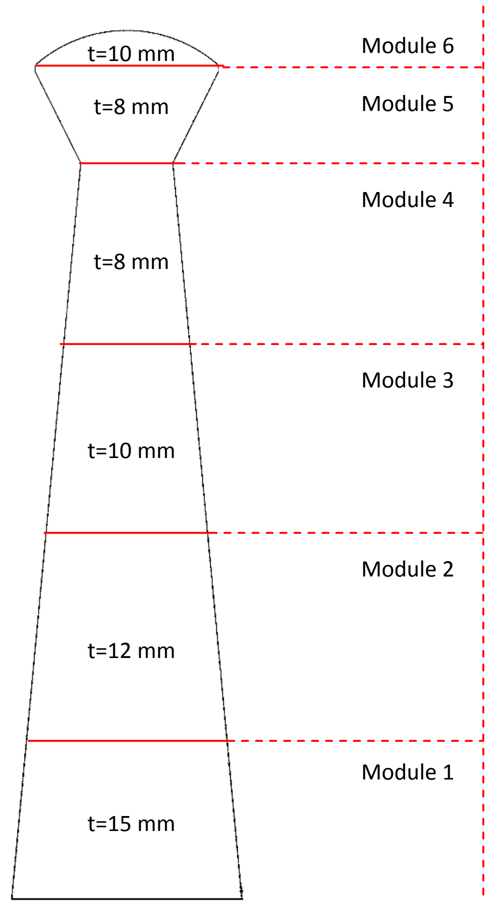

Fig. 5
Shell thickness modular design section in the basic design.

3.3 Shell, Head and Support

For The design and analysis of pressure vessel components, a comprehensive understanding of the governing factors for their minimum required thickness, stresses induced in them, and their life under cyclic load, the PVElite software was used for conceptual and basic design, which also allowed an accurate testing and analysis of numerous load cases. Considering the circumferential, longitudinal, and meridional stresses, the case study is categorized to be a thin cylinder, and thus, the analysis follows the thin cylinder formulae, [23].

3.3.1 Design data

Following the initial calculations and taking into account, the employer's stipulated requirements, a datasheet was devised based on the guidelines outlined in the ASME VIII Div.1, also suggested by [2].

Table 3
Minimum required thickness calculated based on wind loads[†].

	C = Shape factor, ASCE std, 7-02
	D = Width of the vessel with insulation, etc., ft.
	E = Efficiency of welded joints
	F = Wind force, ASCE Std. 7-02
Force: $F = P(DH)$	G = Gust factor, ASCE Std. 7-02
	H = Lever arm. Ft.
Pressure: $P = q_z G C_f$	H_T = Distance from base to section under consideration, ft.
Moment: $M = PDhH$	h = Length of vessel or vessel section, ft.
Stress: $S = \frac{12M}{R^2 \pi t}$	M = Maximum moment (at base) ft.lb.
Required Thickness: $t = \frac{12M}{R^2 \pi S E}$	M_T = Moment at height h_T ft.lb.
	P = Wind pressure at height, ASCE Std. 7-02
	q_z = Velocity pressure at height, ASCE Std. 7-02
	R = Mean radius of the vessel, in.
	S = Stress value of vessel material or actual stress, psi.
	t = required thickness of the shell, in.

Table 4
Tall tower deflection formula[‡].

	Δ_M = Maximum deflection (at the top), in.
	D_1 = Width of the tower with insulation, etc. ft.
	E = Modulus of elasticity, psi
$\Delta_M = \frac{P_W D_1 H (12H)^3}{8EI}$	H = Length of vessel, included skirt, ft.
	$I = R^3 \pi t$, moment of inertia for thin cylindrical shell(when R> 10t)
	R = Mean radius of the tower, in.
	t = Thickness of skirt, in.
	P_w = Wind pressure, psf.

[†] All the symbols are defined according to ASCE std. 7-02 and are not to be mistaken with others defined in nomenclature section.
[‡] All the symbols are defined according to Short Cut Method for Calculating Tower Deflection S. S. Tang and are not to be mistaken with others defined in nomenclature section.

Table 5
Design Datasheet.

Design Code	ASME VIII DIV.1
Service	Water Supply (Tower)
Fluid	Water
Operating Pressure	1.034 Bar
Internal/External design pressure	1.013 Bar
Operating Temperature	Ambient temperature
Design Temperature	1.013 C
Hydrostatic test pressure	1.32 Bar
Joint Efficiency	0.85
Radiography	Full
Corrosion Allowance	3.0 mm
Capacity	600 m^3
Wind Load	120 km/h
Seismic design load	8

3.3.2 Thickness Design

PV Elite software was used to design the dimensions of each module, and a total of 21 sub-modules were designed considering the available sizes of plates in the market and the tower's height.

Basic design: The geometric or nominal capacity of the tank refers to the total volume it can contain, whereas the working capacity is the volume between the Low Liquid Level (LLL) and the High Liquid Level (HLL). The effect of the tank height-to-diameter (H/D) ratio on tank stability, shear, and moment due to wind and seismic loading provides that a ratio of H/D = 2/3 is optimal for both wind and seismic loading, [17].

Typically, vessels supported by rings or lugs are contained within a structure and are subject to seismic movement. For elevated temperature design, a totally loose ring system can be fabricated and held in place with shear bars to avoid any interaction between the shell and the support rings, [20].

Anchor Bolts: With consideration of these frameworks and the design team's experience, a base plate with ring support were considered for the tower. The tower's anchor bolts must be installed in multiples of four, with a minimum of eight bolts recommended for tall towers. The spacing between the bolts is critical for the strength of the foundation. Anchor bolts should not be closer than 25.4 cm to ensure the strength of the foundation. The design team selected SA-36 as the material for the bolts, with a maximum allowable stress of 114452 kPa. To ensure reliability, maximum tension and required area for each bolt (See **Error! Reference source not found.**) were considered. An ideal requirement of 40 bolts, each with a size of 1 ¼ inches, was initially calculated. However, using 48 bolts of 1 1/8 inches resulted in only a 3% difference in stress per anchor bolt (see **Error! Reference source not found.**). Given that 48 bolts of 1 1/8 inches weigh less than 40 bolts of 1 ¼ inches, this configuration was both structurally and financially advantageous. Additionally, it meets all code requirements and ensures base plate integrity. Therefore, 48 anchor bolts were chosen, providing a reliability factor of 32.9%.

Table 6
Designed pressure vessel parameters based on PVElite calculations using design datasheet.

	Coordination Y (Vert.)	Element Height (mm)	Element Diameter (mm)	Minimum thickness (mm)	Element Metal Weight (kg)	Surface Area (cm^2)
Skirt	1474.93	1474.93	7500	15	13357.3	336123
M1	2949.86	1474.93	7224	15	3841.86	331163
M2	4424.79	1474.93	6947	15	3692.09	318284
M3	5899.72	1474.93	6671	15	3542.76	305439
M4	7374.65	1474.93	6395	12	2713.49	292314
M5	8849.58	1474.93	6119	12	2593.88	279455
M6	10324.5	1474.93	5842	12	2474.12	266578
M7	11799.4	1474.93	5566	12	2354.68	253734
M8	13274.4	1474.93	5290	10	1861.98	240702
M9	14749.3	1474.93	5014	10	1762.28	227841
M10	16224.2	1474.93	4737	10	1662.5	214966
M11	17699.2	1474.93	4461	10	1562.93	202122
M12	19174.1	1474.93	4185	8	1170.12	189090
M13	20649	1474.93	3909	8	1090.36	176228
M14	22123.9	1474.93	3632	8	1010.55	163354
M15	23608.8	1484.9	3356	8	940.534	152067
M16	24021.4	412.584	3109	8	248.269	40146.5
M17	25343.2	1321.73	4324	8	1057.06	170896
M18	26664.9	1321.73	5648	8	1438.3	232383
M19	27020.3	355.437	6000	8	450.632	72791.5
Head	27070.3	50	6000	10.5	4111.2	402825

Base plate: Quick Footing software was used to design and analyze the base plate, which confirmed a thickness of 100 mm. The calculations for the base ring were analyzed using PV Elite software. Based on the analysis, the required thickness for the base ring was found to be 65.10 mm, while the actual thickness entered by the user was 80.00 mm. The required thickness for the top ring/plate as a fixed beam was 34.00 mm, and for continuous top ring (based on, [20]) was 44.02 mm. However, the actual top ring thickness entered by the user was 60.00 mm. The required gusset thickness plus corrosion allowance was found to be 13.34 mm, while the actual gusset thickness entered by the user was 20.00 mm. Finally, the required weld sizes for the double fillet weld between the base ring and skirt, gusset and skirt, and top plate and skirt were calculated to be 7.20 mm, 7.20 mm, and 8.51 mm, respectively.

Table 7
Calculating number of anchor bolts for the base plate based on maximum tension and required area for one bolt.

$T = \frac{12M}{A_B} - \frac{W}{C_B}$ = 300.25 (Maximum tension lb./lin.in.) $B_A = \frac{TC_B}{S_B N}$ (Required area of one bolt sq.-in.) $S_B = \frac{TC_B}{B_A N}$ (Stress in anchor bolt psi.)	A_B = Area within the bolt circle, sq. in. C_B = Circumference of bolt circle in. M = Moment at the base due to wind or earthquake, ft. lb. N = Number of anchor bolts S_B = Maximum allowable stress value of bolt material psi. w = Weight of the vessel during erection, lb.

Table 8
Number of anchor bolts calculated to reach the most reliable number.

N	B_A (sq.-in.)	B_A (sq.-in.) – based on (Megyesy, 2008)	S_B (psi)	Reliability factor
4	4.327	4.736	15165	8.6%
8	2.163	2.418	14852	10.5%
16	1.082	1.412	12716	23.4%
24	0.721	1.008	11875	28.5%
32	0.541	0.669	13420	19.2%
40	0.433	0.669	10736	35.3%
48	0.361	0.537	11146	32.9%

3.4. Openings (flanges, nozzles, plates, bolts, piping)

In this paper, the calculations and design for the tower are limited to the scope of the project, taking into account its size and application. Since there were only two nozzles designed in the first module and limited piping installed within the tower, additional calculations and design for these components are not discussed here. However, they will be taken into consideration when evaluating the final test results.

4 TEST AND ANALYSIS

During the analysis phase of the project, two main steps are given priority - the soil and foundation analysis, and the case loadings on the tower. The soil and foundation analysis helps in assessing the soil's bearing capacity and the foundation's ability to support the tower's load. The case loading analysis involves studying the wind and seismic forces that may act on the tower, which can lead to deformation or failure if not appropriately addressed. According to the mechanical principles of the structural theory, the static and dynamic response of a structure is related to its stiffness. Any sort of changes in the stiffness of the structure will be accompanied by changes in the static and dynamic response. By investigating the response of structures, one can determine the characteristics of defects, [24].

4.1 Foundation Analysis

The weight of the vessel can result in compressive stress only when there is no eccentricity, and the resultant force coincides with the axis of the vessel. The weight is usually insignificant and does not control the design. The weight can be calculated for various tower conditions by considering the erection weight, operating weight, and test weight, [2].

Quick Footing Software was used to control the dimensions of the pile under service load, earthquake, and wind loads, while other parameters were controlled using SAFE ver.14 Software. Soil sample analysis from labs confirmed the procedure, with a maximum amount of q_{all}=1.60 kg/cm^3 for strip foundation.

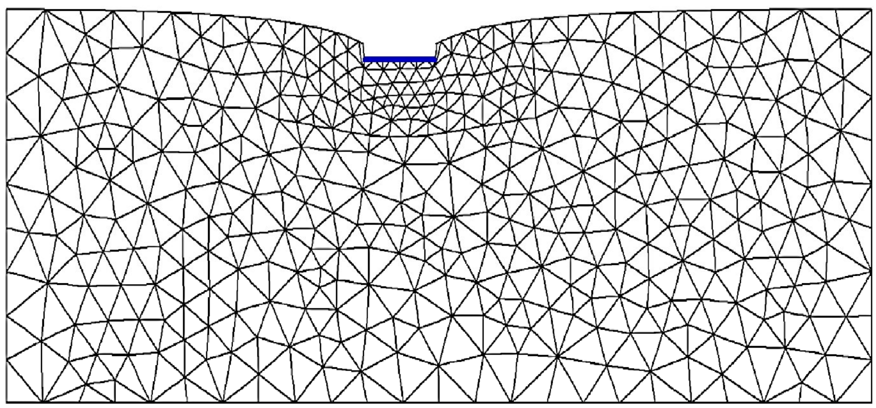

Fig. 6
Plot of deformed mesh to analyze foundation soil.

4.2 Vessel Analysis

ASME Code Requirements dictate that pressure vessels designed and constructed to VIII-1 rules, except for those tested in accordance with the requirements of UG-101, must pass a hydrostatic or pneumatic test before being U-stamped. The test pressure should be at least 1.3 times the maximum allowable working pressure, [20].

The primary load cases are defined and analyzed here to provide a better understanding of the load cases test result calculated by PVElite software. The high stresses at intersections are caused by shear stresses and moments that exist to maintain compatibility at the junctions. PVElite uses the welding research council (WRC) 107 formula to calculate these local stresses, [9].

Wind load

The wind load calculation presented here is based on the 1997 UNIFORM BUILDING CODE (UBC-97) published by the International Code Council. In accordance with this code, design wind pressure for buildings and structures can be determined, [2]. See Eq. (4).

$$P = C_e C_q q_s I_W \tag{4}$$

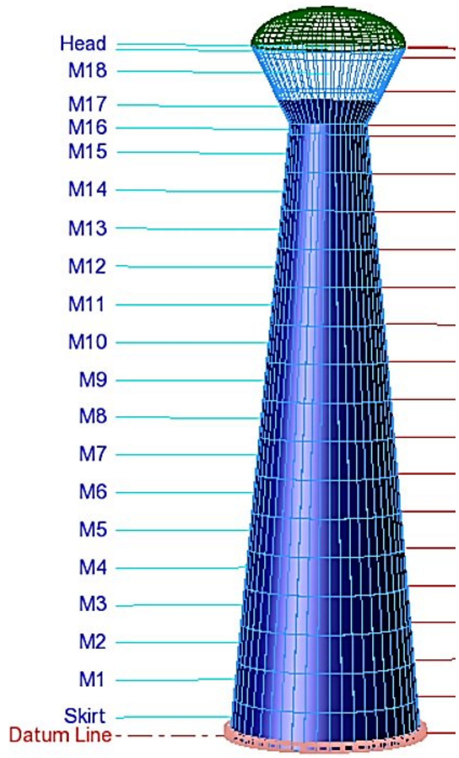

Fig. 7
Level of liquid (water) considered within the tower for analysis.

For a cylindrical vessel, C_q (combined height, exposure, gust factor) is 0.8, and with C_e (pressure coefficient) = 1.07, q_s (wind stagnation pressure at the standard height of 33 ft.) = 16.40, and I_w (importance factor) = 1.15, the calculated design wind pressure is 16.26 lbs/sq.ft. Multiplying this value by the area of the structure, 5166 sq.ft., gives a wind force of 38106.29 Kg, which is designated for wind load testing.

Seismic load

The design method for a tower under seismic forces is based on the Uniform Building Code, 1997 (UBC). The base shear is the total horizontal seismic shear at the base of the tower, which is distributed throughout the length of the tower. The required corroded vessel thickness is calculated using the allowable tensile stress of vessel plate material, and based on this thickness, the seismic load factor is compatible with the design thickness, [2]. Regarding the failure modes, the following modes must be considered to avoid such as Elephant's foot buckling, Roof damage, Failure of base plate, Anchor bolt failure and Nozzle (attached piping) failure, [25]. For the given tower, the base shear is calculated to be 4.26 kN and the maximum moment at the base is 627033.85 kN.m. The total weight of the tower is 89999.98 kg. Considering these values, a minimum corroded thickness of 9.14 mm is required to withstand the seismic loading (See **Error! Reference source not found.**). See Eq. (5-8).

Table 9
Required corroded vessel thickness calculated based on seismic load of 1997 (UBC).

Notations

Parameter	Description	Quantity
C	Numerical coefficient	0.035
D	Outside diameter of vessel, ft.	24.66
E	Efficiency of welded joints	0.85
F_t	F_t= 0.07 TV	73.75
H	Length of vessel including skirt, ft.	88.58
I	Occupancy importance coefficient	1
M	Maximum moment (at the base), ft-lb.	341105115.69
M_X	Moment at distance X, ft-lb.	
R	Mean radius of vessel, in.	73.81
R_W	Numerical coefficient	2.9
S	Site coefficient for soil characteristics	1.5
S_t	Allowable tensile stress of vessel plate material, psi.	1.1
T	Fundamental period of vibration, seconds =C xH%	1.1
t	Required corroded vessel thickness, in. $\frac{12M}{\pi R^2 S_t E}$	0.46 (9.14 mm)
V	Total seismic shear at base, lb.	957.87
W	Total weight of tower, lb.	198416
X	Distance from top tangent line to the level under consideration, ft.	
Z	Seismic zone factor	0.4

$$v = \frac{ZIC}{R_W} w; \tag{5}$$

$$M = \left[F_t H + (V - F_t)\left(\frac{2H}{3}\right)\right]; \tag{6}$$

$$M_x = [F_t X] \quad for\ X \leq H/3; \tag{7}$$

$$M_x = \left[F_t H + (V - F_t)\left(X - \frac{H}{3}\right)\right] \quad for\ X > \frac{H}{3}; \tag{8}$$

Vibration

The maximum allowable period of vibration (T_a) and the actual period of vibration (T) were calculated based on the tower's height (H), outside diameter (D), weight (W), seismic shear (Vg), and required corroded vessel thickness (t), [2]. The results show that the actual period of vibration is 0.9865 seconds, while the maximum allowable period of vibration is 108.36 seconds. Since the actual vibration period is well within the allowable limit, there is no concern for fatigue failure due to vibration. See Eq. (9-10).

$$T = 0.0000265\left(\frac{H}{D}\right)^2 \sqrt{\frac{WD}{t}}; \tag{9}$$

$$T_a = 0.80\sqrt{\frac{WH}{V_g}}; \tag{10}$$

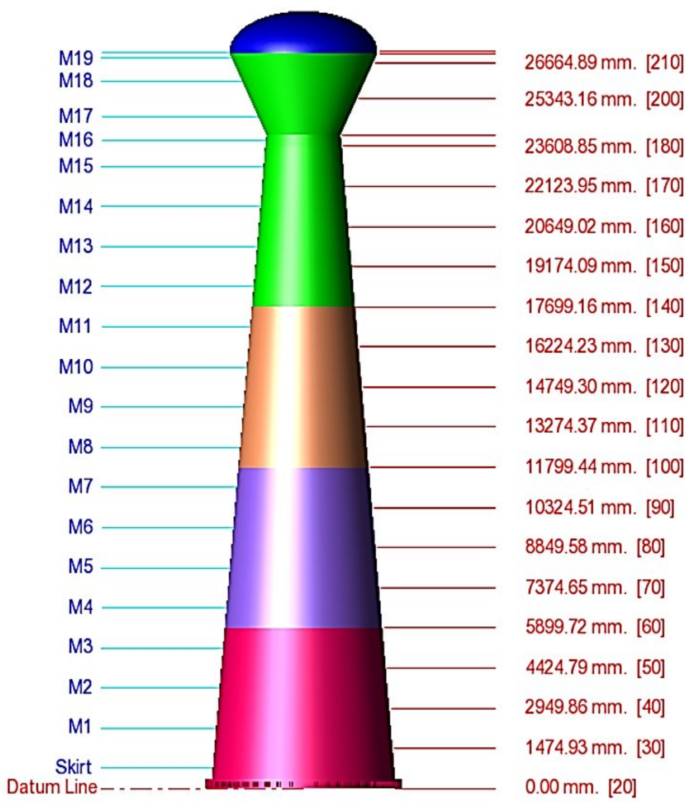

Fig. 8
PVElite model of the elevated atmospheric tank (colors demonstrated modules with different thickness).

Stress Analysis for Combined Loading Cases in Vertical Vessels

The stresses resulting from wind load, earthquake load, internal pressure, and weight of the vessel need to be investigated in combination to establish the governing stresses. The area of geometric discontinuity, such as the junction of the cylinder with its hemispherical head, are the most susceptible areas for crack initiation, where the stress field changes along the axial and, depending on the load case under consideration, also along the circumferential direction. Consequently, it is worth considering cracks at this connection, [26]. The stresses are calculated at various locations such as the bottom of the tower, joint of the skirt to the head, bottom head to the shell joint, and changes of diameter or thickness of the vessel. The positive and negative signs represent tension and compression, respectively. To ensure the tower's safety, it was designed for either wind or earthquake load, whichever was greater, [2]. PVElite software has a set of pre-defined load cases which consist of different external pressures applied, the abbreviations are explained in nomenclature section, the applied load cases applied according to ASME VIII-1 are defined as described in Appendix 1.

The structural test results have provided data on the stress values for each node in the structure under different load cases. Stress types such as longitudinal stress, bending stress, and axial stress have been identified, and the stress ratios for each node have been calculated. The governing stress ratio for each load case is determined by examining the ratio where the compressive stress is greater than the tensile stress. For clarity, the results generated by PVELITE, define the starting of each module as a node, (as shown in **Error! Reference source not found.**), which is also identified with a coordination of the height. The following is a descriptive analysis of the generated results:

Load Case 1 shows that most of the nodes have compressive stress values, with only a few nodes having tensile stress.

In Load Case 2, only three nodes have non-zero stress values, with one node having both compressive and tensile stress.

Load Case 3 shows that the component is experiencing predominantly compressive stress, with very low tensile stress in some areas.

Load Case 4 reveals that most nodes have compressive stress, with only a few having both compressive and tensile stress.

Load Case 5 shows no significant difference in stress distribution between tension and compression, with all nodes having zero stress ratio.

For Load Case 6, the stress ratios are all zero, indicating a predictable behavior of the structure under applied loads. The stresses are uniformly distributed throughout the structure, with no combination of stresses resulting in shear stresses.

Load Case 7 results in a mix of tensile and compressive stresses across different nodes. Node 20 has the highest stress ratio, with a greater susceptibility to tensile stress than compressive stress. The rest of the nodes show varying levels of stress, with some experiencing higher stresses than others.

Load Case 8 has Node 20 as the governing stress ratio, which determines the stress ratio for this set.

Load Case 9 produces a combination of tensile and compressive stresses, with Node 10 experiencing the highest stress ratio.

Load Case 10 experiences higher compressive stress than tensile stress, with Node 30 having the highest governing stress ratio.

Load Case 11 shows a shift from compression to tension as we move from node 10 to node 220.

Load Case 12 provides stress ratios ranging from 0.0023 to 0.1541 across different nodes, indicating the importance of understanding these values for optimizing designs and ensuring structural integrity.

Load Case 13 records stress values and ratios for different nodes, with the stress being higher in the compressive direction than in the tensile direction.

Load Case 14 shows node 20 experiencing the highest stress value,

and Load Cases 15, 16, and 17 demonstrate consistently higher tensile stress ratios, suggesting that the structure is experiencing greater tensile stress than compressive stress.

Load Cases 18 and 19 are being analyzed for stress ratios under different loading conditions, including forces from the front and back sides of the structure, internal pressure, out-of-plane forces, and external pressure.

The results suggested that the governing load case for this structure is Load Case 4, which includes a combination of loads from different directions on M1 (See **Error! Reference source not found.**). In the end, by considering the load case tests, the design stage was confirmed and passed on for fabrication (**Error! Reference source not found.**).

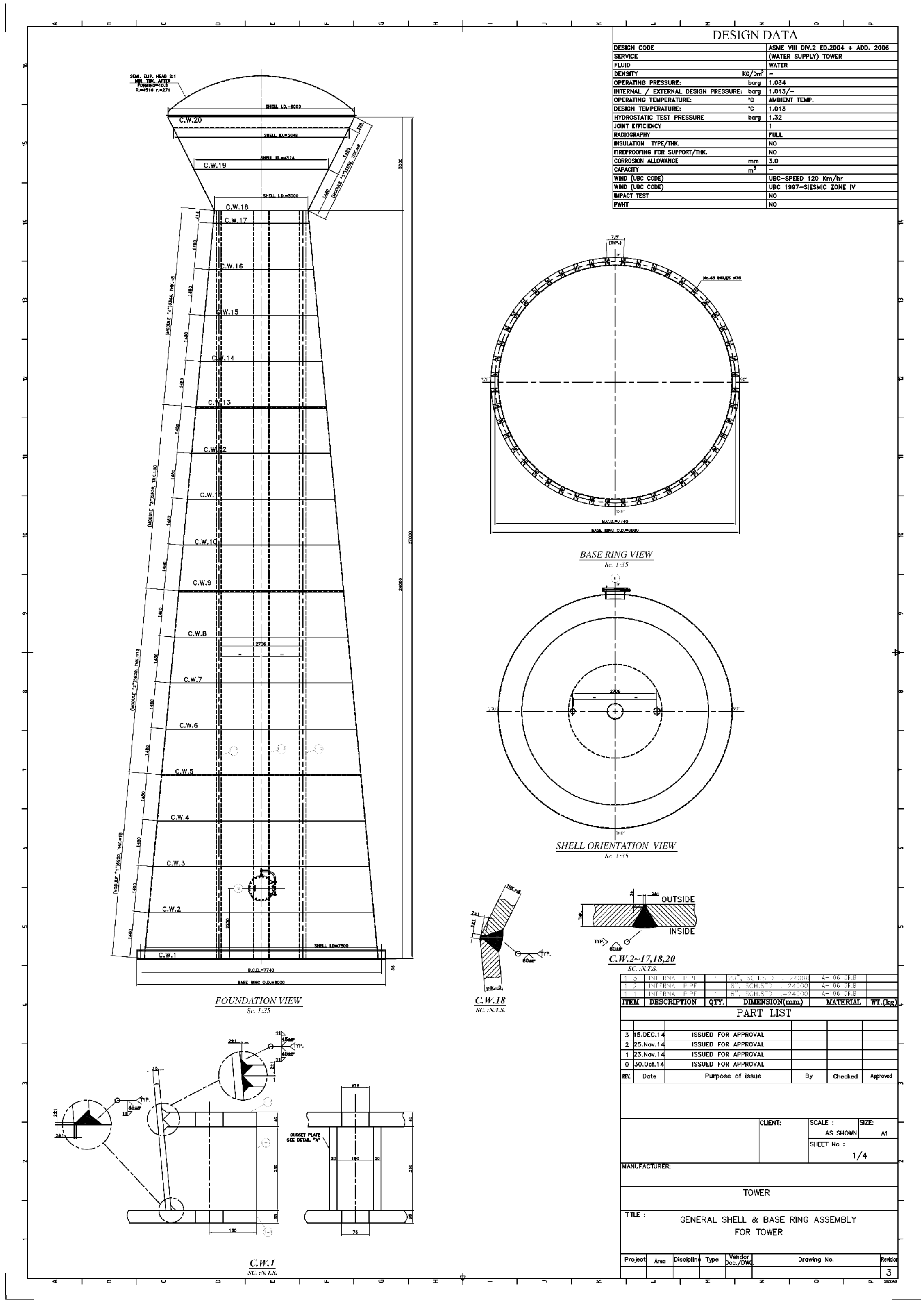
DESIGN DATA
DESIGN CODE
ASME VIII DIV.2 ED.2004 + ADD. 2006
SERVICE
(WATER SUPPLY) TOWER
FLUID
WATER
DENSITY KG/Dm3
OPERATING PRESSURE: barg
1.034
INTERNAL / EXTERNAL DESIGN PRESSURE: barg
1.013/-
OPERATING TEMPERATURE: °C
AMBIENT TEMP.
DESIGN TEMPERATURE: °C
1.013
HYDROSTATIC TEST PRESSURE barg
1.32
JOINT EFFICIENCY
1
RADIOGRAPHY
FULL
INSULATION TYPE/THK.
NO
FIREPROOFING FOR SUPPORT/THK.
NO
CORROSION ALLOWANCE mm
3.0
CAPACITY m3
WIND (UBC CODE)
UBC-SPEED 120 Km/hr
WIND (UBC CODE)
UBC 1997-SIESMIC ZONE IV
IMPACT TEST
NO
PWHT
NO
BASE RING VIEW
SHELL ORIENTATION VIEW
FOUNDATION VIEW
C.W.18
C.W.2~17,18,20
C.W.1
OUTSIDE
INSIDE
PART LIST
ITEM
DESCRIPTION
QTY.
DIMENSION(mm)
MATERIAL
WT.(kg)
3 15.DEC.14 ISSUED FOR APPROVAL
2 25.Nov.14 ISSUED FOR APPROVAL
1 23.Nov.14 ISSUED FOR APPROVAL
0 30.Oct.14 ISSUED FOR APPROVAL
REV. Date Purpose of issue By Checked Approved
CLIENT:
SCALE : AS SHOWN
SIZE: A1
SHEET No : 1/4
MANUFACTURER:
TOWER
TITLE : GENERAL SHELL & BASE RING ASSEMBLY FOR TOWER
Project Area Discipline Type Vendor Doc./DWG. Drawing No. Revision
3

Fig. 9

Final detailed plan of atmospheric water tower.

5 DISCUSSION

In the context of pressure vessel design, the selection of a reference code is a crucial step that depends not only on the intended development purposes of the code but also on industry experience. Among the available codes, the ASME codes are generally considered more extensive and conservative. However, in the present study, reference, [2], indicates that both API 620 and ASME VIII DIV.1 are appropriate codes for the conditions of the project. To elaborate further, this paper focuses on the procedural approach suggested by both codes for design and analysis, with particular attention given to their SCOPE and DESIGN instructions.

To gain a more comprehensive understanding of the differences between various pressure vessel codes, researchers have published several papers on the topic. For instance, a study details the variations in codes applicable to pressure vessels with pressures below 15 psi, [10]. By examining general features such as configuration and maximum pressure, it appears that API 620 is a more suitable option for an elevated atmospheric tank. Furthermore, while the minimum material thickness for both codes meets the project's basic assumptions and final designs, API 620's design parameters are not as conservative as ASME's. Welding certification is not a primary concern in this comparison since API 620 also refers to ASME IX.

5.1 Scope

Code content is essential to ensure that pressure vessels meet the required standards. Important points to consider include the roles of the manufacturer and the purchaser, the use of independent inspection organizations, and the technical requirements for design. The code also includes details on materials of construction, such as plate, forged parts, bar sections, and tubes, and their specific requirements for low temperature applications. The code also contains manufacturing, inspection, and testing requirements, including material identification and traceability, NDT, welder approvals, pressure testing, and the content of the vessel's documentation package, [3].

Tank classification is based on the internal pressure above the stored product (vapor space), with API 650 code applicable for atmospheric pressure tanks that are flat bottom cylindrical tanks with cone roofs. For internal pressures above 0.004 bar but less than 0.17 bar, the provisions of Appendix F in API 650 are considered. Tanks with internal design pressures above 0.17 bar and less than 1.03 bar are designed as flat bottom cylindrical tanks with dome roofs per API 620, with the maximum allowable external pressure at 0.0043 bar. Liquids with vapor pressures > 1.5 psia are stored in internal or external floating roof tanks under API 650. Above 1.03 bar or the API limitations, tanks are designed as spherical or cigar tanks under ASME code section VIII DIV 1 or DIV 2. Alternatively, cryogenic tanks can be used to store large amounts of liquefied gases at low temperatures and slight pressure, requiring refrigeration units and cost-effectiveness for large storage needs in liquefied gas export facilities, [17].

API 620 defines its scope as covering large, field-assembled storage tanks of the type described in section 1.2, which contain petroleum intermediates (gases or vapors) and finished products, as well as other liquid products commonly handled and stored by various industries. While the general section of API 620 considers petroleum, especially in the form of gas, as the main containment, it expands to cover other liquids, including water, considering the maximum specific gravity of 1. In contrast, ASME VIII follows a different approach and does not consider the containment in its general scope (U1), stating that "pressure vessels are containers for the containment of pressure, either internal or external. This pressure may be obtained from an external source, or by the application of heat from a direct or indirect source, or any combination thereof."

API 620 further specifies that it covers the design and construction of large, welded, low-pressure carbon steel above-ground storage tanks, including flat-bottom tanks, that have a single vertical axis of revolution. This suggests that API 620 is supportive of the design of vertical atmospheric storage tanks, while ASME does not exclude these parameters. However, ASME VIII div.1 expands two articles (U1-C-2-F) and (U1-C-2-H), which specify classes of vessels not included in the scope of that division.

Article U1-C-2-F defines vessels for containing water under pressure, including those containing air, the compression of which serves only as a cushion when none of the following limitations are exceeded: a design pressure of 300 psi (2 MPa); a design temperature of 210°F (99°C). On the other hand, article U1-C-2-H refers to vessels not exceeding the design pressure at the top of the vessel with no limitation on size, provided that vessels have an internal or external pressure not exceeding 15 psi (100 kPa) and combination units have an internal or external pressure in each chamber not exceeding 15 psi (100 kPa) and differential pressure on the common elements not exceeding 15 psi (100 kPa). The footnote in ASME VIII DIV.1 includes information on water properties, stating that the water may contain additives, provided the flash point of the aqueous solution at atmospheric pressure is 185°F or higher, which shall be determined by the methods specified in ASTM D93 or in ASTM D56, whichever is appropriate.

In this case study, the primary goal was to provide a drinking water supply for an industrial town, and the process engineering team dictated a definitive need for certain chemical properties to be maintained. Therefore, by referring to (Megyesy, 2008), paint properties were designed with epoxy cover. Considering this solution, and the fact that article U1-C-2-H is not applicable since this was a vertical tower, it could be concluded that both API 620 and ASME VIII DIV.1 are subject to the engineer's judgment and experience, with API 620 stating that "the tanks described in this standard are designed for metal temperatures not greater than 250°F and with pressures in their gas or vapor spaces not more than 15 lbf/in.2 gauge," and ASME VIII DIV.1 offering further clarification regarding the circumstances under which these parameters could be considered in other sections.

The painting section of, [2], demonstrate that painting steel surfaces for the preservation of the material by retarding corrosion through preventing contact with corrosive agents and utilizing rust inhibitive and electro-chemical properties of the paint. Successful paint jobs require thorough surface preparation by removing mill scale, rust, dirt, grease, oil, and foreign matter. The selection of paint systems must consider technical aspects as well as economic considerations, balancing the cost of surface preparation against the increased life of the vessel. The section also provides recommended paints for high temperature applications, [2].

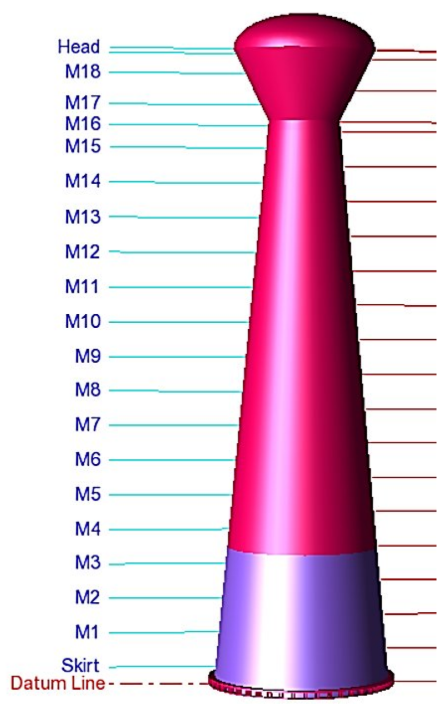

Fig. 10
PVElite modelling of the tower demonstrating selected materials for different sections.

5.2 Material

Table 10
A comparison of design articles between ASME VIII-1 and API 620, discussing the parallel notions within the scope of the case study[§] .

ASME VIII div1.	API 620	ASME VIII div1.	API 620
UG-16 General	5.1 General	Minimum thickness required for vessels is 1.5 mm	_ Each pressure part shall be designed for the most severe combination of pressure or vacuum - The volume of the vapor space above the high liquid design level shall be not less than 2 % of the total liquid capacity
UG-20 Design Temperature	5.2 Operating Temperature	_ maximum temperature used in design shall be not less than the mean metal temperature (through the thickness) expected under operating conditions for the part considered -The Minimum design metal temperature (MDMT) marked on the nameplate shall correspond to a coincident pressure equal to the MAWP	The temperature of the liquids shall not exceed 250 °F (121.11 °C)
UG-21 Design Pressure	5.3 Pressures Used in Design	shall be designed for at least the most severe condition of coincident pressure (including coincident static head in the operating position) and temperature expected in normal operation	_Above Maximum Liquid Level - shall not exceed 15 lbf/in.2 gauge (1.0342 bars) -Below Maximum Liquid Level - shall be designed for the most severe combination of gas pressure (or partial vacuum) and static liquid head affecting the element -The weight for liquid storage shall be assumed to be the weight per ft3 of the specified liquid contents at 60 °F
UG-22 Loadings	5.4 Loads	a. Internal or external pressure b. Weight of the vessel and contents c. Static reactions from attached equipment, piping, lining, insulation, d. The attachment of internals, vessel supports, lugs, saddles, skirts, legs e. Cyclic and dynamic reactions due to pressure or thermal variations f. Wind pressure and seismic forces g. Impact reactions due to fluid shock b. Temperature gradients and differential thermal expansion i. Abnormal pressures caused by deflagration .	"dead load (DL) - hydrostatic and pneumatic tests (Ht) - loads from connected piping (Lp) - loads from platforms and stairways (Ls) - minimum roof live load (Lr) - pressure (Pg) - pressure (Pv) - seismic (E)" - For tank components located more than 80 ft above ground, use ASCE 7 to determine wind pressures. a) DL + Pg + Pl b) DL + WL + 0.7Pg c) DL + WL + 0.4Pv d) DL + Pv + 0.4(Lr or S) e) DL + 0.4Pv + (Lr or S) f) DL + 0.7Pg + Pl + E + 0.1S g) DL + Ht h) DL + Ls i) DL + Lp + Pg + Pl

[§] (ASME Boiler and Pressure vessel Commitee, 2021), [30], (API620, Design and Construction of Large, Welded, Low-pressure Storage Tanks, 2018), [31].

UG-23 Maximum Allowable Stress Values	5.5 Maximum Allowable Stress for Walls	_The maximum allowable stress value is the maximum unit stress permitted in a given material used in a vessel constructed under these rules - the induced maximum general primary membrane stress does not exceed the maximum allowable stress value in tension - For the combination of earthquake loading, or wind loading with other loadings, primary membrane stress shall not exceed 1.2 times the maximum allowable stress permitted - Values for yield strength, SY , as a function of temperature: (1) If allowable stress is established based on the 662/3% yield criterion, then yield strength, SY , shall be taken as 1.5S/f . (2) If the allowable stress is established based on yield criterion between 662/3% and 90%, then the yield strength, SY , shall be taken as 1.1S/f .	_meridional and latitudinal forces with regards to specifications of the materials - Maximum Tensile Stresses - Maximum Compressive Stresses - Maximum Shearing Stresses - Maximum Allowable Stresses for Wind or Earthquake Loadings - Allowable Stress for Tests
UG-25 Corrosion	5.7 Corrosion Allowance	Vessels or parts of vessels subject to thinning by corrosion, erosion, or mechanical abrasion shall have provision made for the desired life of the vessel by a suitable increase in the thickness of the material over that determined by the design formulas, or by using some other suitable method of protection	additional metal thickness in excess of that required by the design computations shall be provided, or some satisfactory method of protecting these surfaces from corrosion shall be employed. The added thickness need not be the same for all zones of exposure inside and outside the tank
UG-26 Linings	5.8 Linings	Corrosion resistant or abrasion resistant linings, whether or not attached to the wall of a vessel, shall not be considered as contributing to the strength of the wall except as permitted	When corrosion-resistant linings are attached to any element of the tank wall, including nozzles, their thickness shall not be included in the computation for the required wall thickness
UG-27 Thickness of Shells Under Internal Pressure	5.10 Design of Sidewalls, Roofs, and Bottoms	_ The minimum required thickness at the thinnest point after forming22 of ellipsoidal, torispherical, hemispherical, conical, and toriconical heads under pressure on the concave side - Ellipsoidal Heads With t s/L ≥ 0.002. The required thickness of a dished head of semiellipsoidal form, in which half the minor axis (inside depth of the head minus the skirt) equals one□fourth of the inside diameter of the head skirt, shall be determined - The required thickness of an ellipsoidal head having pressure on the convex side, either seamless or of built□up construction with butt joints,	_Free-body analysis denotes a design procedure that determines the magnitude and direction of the forces that must be exerted by the walls of a tank - made at successive levels from the top to the bottom of the tank for the purpose of determining the magnitude and character of the meridional and longitudinal unit forces that will exist in the walls of the tank at critical levels under all the various combinations of gas pressure (or partial vacuum) and liquid head to be encountered in service, which may have a controlling effect on the design - the determination of optimum shapes and sizes is frequently a trial-and-error procedure requiring considerable experience and judgment - Flat bottoms of cylindrical tanks that are uniformly supported on a ringwall, grade, or concrete-slab foundation are pressure-resisting membranes but are considered nonstressed because of support from the foundation. - All bottom plates shall have a minimum nominal thickness of 1/4 in.

shall not be less than that determined	- Bottom plates shall be ordered to a sufficient size so that when they are trimmed, at least a 1 in. width will project beyond the outside edge of the weld that attaches the bottom to the sidewall plate, or 2 in. width will project outside the sidewall plate, whichever is greater. - Bottom plates under the sidewall that are thicker than 3/8 in. shall be butt-welded. The butt-welds shall be made using a backing strip 1/8 in. thick or more, or they shall be butt-welded from both sides. Welds shall be full fusion through the thickness of the bottom plate. The butt-weld shall extend at least 24 in. inside the sidewall. - At each level of the tank selected for free-body analysis as specified in and for each condition of gas and liquid loading that must be investigated at that level, the magnitude of the meridional and latitudinal unit forces in the wall of the tank shall be computed from the following equations - The thickness of the tank wall at any given level shall be not less than the largest value of t as determined for the level by the methods prescribed in 5.10.3.2 through 5.10.3.5. In addition, provision shall be made by means of additional metal, where needed, for the loadings other than internal pressure or possible partial vacuum enumerated - A measure of 3/16 in. plus the corrosion allowance. - In tanks that have cylindrical sidewalls and flat bottoms, the uplift that results from the pressure acting on the underside of the roof combined with the effect of design wind pressure, or seismic loads if specified, must not exceed the weight of the sidewalls plus the weight of that portion of the roof that is carried by the sidewalls when no uplifts exists unless the excess is counteracted by a counterbalancing structure such as a concrete ringwall, a slab foundation, or another structural system. - The counterbalancing structure, which may be a foundation or support system, shall be designed to resist uplift calculated as described in 5.11.2 based on 1.25 times the internal design pressure plus the wind load on the shell and roof based on its projection on a vertical plane. If seismic loads are specified, uplift shall be calculated using internal design pressure plus the seismic loads. Wind and seismic loads need not be combined. (5-30) - Flat-bottom Tanks without Counterbalancing Weight (5-32) -Unless otherwise required, tanks that may be subject to sliding due to wind shall use a maximum allowable sliding friction of 0.40 times the force against the tank bottom.

Regarding the materials, neither of the codes disqualifies the material used in this project, but there are some articles that propose an argument for the topic of this paper.

Consequently, the choice of materials is based on several factors, including notch toughness, area at fracture elongation and reduction, availability, ageing and brittleness under operating circumstances, and resistance to fatigue. Safety factors, such as yield strength at design temperature, creep strength at design temperature, and maximum tensile strength at room temperature, are also taken into consideration. The most commonly used materials for pressure vessel construction are carbon steel, Hastelloy, stainless steel, nickel alloys, and titanium. Cast iron is not used due to its inability to be welded, and aluminum is not suitable for high-pressure applications despite its high corrosion resistance and light weight, [27].

When dealing with carbon steels, ASME VIII-1 has two options regarding toughness requirements, where the first option exempts the material from impact testing if certain conditions are met. For thicker carbon steel vessels or vessels made of low alloy steel, designers need to evaluate brittle fractures in accordance with the rules of UCS-66, [28].

ASME Section II provides an extensive material database for selecting appropriate materials, which is referred to by ASME Section VIII for designing pressure vessels. In this context, ASME VIII focuses mainly on design methodology and conditions. On the other hand, API 620 suggests a methodology for material selection, which requires all plates that are subject to pressure-imposed membrane stress or that are important to the structural integrity of a tank, including bottom plates welded to the cylindrical sidewall of flat-bottom tanks, to conform to specifications that provide a high level of resistance to brittle fracture at the lowest temperature expected in the locality where the tank is installed. Additionally, a list of materials is proposed with specific conditions based on this approach.

In this study (See **Error! Reference source not found.** - PVElite modelling of the tower demonstrating selected materials for different sections), the plate material section of each code is considered, and the selected material for the project complies with both ASME Section VIII and API 620. While API 620 specifies the thickness and design metal temperature, which is also subject to impact testing, code A 285 imposes limitations such as "Grade C only, with a maximum nominal thickness of 3/4 in". These limitations confirm the conditions of the detailed design of the tower, but they also indicate less reliability compared to ASME VIII.

5.3 Design

When designing pressure vessels in accordance with the ASME Code, Section VIII, Division 1, designers use rules and do not require a detailed evaluation of all stresses, but they must consider all loadings. The Code establishes allowable stresses, and higher allowable stresses are permitted if appropriate analyses are made. Stress analysis is the determination of the relationship between external forces applied to a vessel and the corresponding stresses, and the designer must be familiar with the various types of loadings and their stresses in order to accurately understand the results of the analysis. It is not necessary to find every stress but rather to know the governing stresses and how they relate to the vessel or its respective parts, attachments, and supports. The strength/failure theory utilized, the types and categories of loadings, and the hazard the stress represents to the vessel determine how these stresses are interpreted and combined and what allowable stresses are applied, [20], (Moss & Basic, 2012). To summarize, the design of pressure vessels is typically based on cylindrical shell theory, taking into account various practical requirements and design criteria such as the probability of weld flaws and defects. Basic vessel design is concerned with internally pressurized, welded steel, unfired vessels operating at room temperature and above, with limitations placed on design tensile stress of the vessel material. Different design codes have varying numerical factors and symbol sets, with US practice typically using "S" or "F" to represent stress and European practice using "σ". These considerations are crucial for ensuring the safe and efficient operation of pressure vessels in a variety of industries, [3], (Matthews, 2001).

ASME VIII DIV.1 Subsection A provides detailed guidelines for designing pressure vessels, with specific instructions given in terms of UG-16 to UG-55. API 620 also has a chapter dedicated to design, and similarly, this paper covers comparable sections related to its topic (see **Error! Reference source not found.**).

The tower in question was designed according to the, [2], handbook with reference to ASME VIII, as shown in the table. It is clear that API 620 also covers the design, which indicates that ASME's method of bounding the design to material properties and geometry can accommodate a wide range of pressure vessel shapes and functionality. However, the flexibility of the ASME handbook may be a disadvantage in terms of design and loading testing time and expertise required, while API 620 provides more specific limitations and conditions for shape, conditions, and loading. These constraints are supported by material properties and geometry, which enhance the reliability and commitment of the design. In the subsequent discussion, we will explore two specific articles of these codes that differentiate them and gain a more precise understanding of their design approaches.

5.3.1 Thickness design

Using PV Elite in accordance with ASME VIII Div.1, the necessary calculations were performed as discussed in the previous sections. The engineer's judgement and adherence to the essentials of the code led to the consideration of the maximum applied pressure in a vertical tower at the bottom, and the minimum required thickness was designed accordingly. The rest of the modules were designed based on this governing element, except for the juncture discontinuity which was modified only due to wind loads. In this approach, the tower is analyzed as a solid object, and the forces are applied uniformly.

Table 11
Load case parameters defined according to API 620.

DL	Dead load represented by the weight of the tank or tank component including any insulation lining or corrosion allowance unless otherwise noted
H_t	Hydrostatic and pneumatic tests Which refer to the load generated by conducting the tests specified in 7.18;
L_p	Loads from connected piping
L_s	Loads from platforms and stairways
Lr	Minimum roof live load with a value of 20 lb./ft^2 on the horizontal projected area of the roof;
P_g	representing the maximum positive gauge pressure given in 5.3.1;
P_v	representing the maximum partial vacuum given in 5.3.1. The maximum partial vacuum shall be at least 1 in.;
€	which refers to the seismic loads given in Annex L;
Pl	which is the gauge pressure (lb./in.^{2}) resulting from the liquid head of the liquid with the density given in 5.3.3. All liquid levels from empty to the maximum liquid level shall be considered

API 620 recommends a free-body analysis, which conforms to the modular design of the tower but also necessitates the design and analysis of each body separately and under the most severe conditions, resulting in high reliability. Therefore, API 620 provides two formulas to calculate forces applied to each body, regardless of their shapes, except for discontinuity junctures. These forces represent the magnitude of the meridional and latitudinal unit forces in the tank wall, computed. See Eq. (11-12).

$$T_1 = \frac{R_2}{2}\left(P + \frac{w+F}{A_t}\right); \tag{11}$$

$$T_2 = R_2\left(P - \frac{T_1}{R_1}\right); \tag{12}$$

While API 620 provides more instructions regarding the maximum allowable stress for wind, earthquake and hydrostatic tests according to article 5.5.6-7, [29], (API620, API STD 620, 2013). These criteria will be considered in the next part, but they also contribute to API 620's focus on tank thickness, which aims to maintain a certain level of reliability regardless of the mechanical specifications of the material.

5.3.2 Load cases

In contrast to PV Elite, ASME VIII DIV.1 does not provide predefined combinations of loadings. Instead, the code's design requirements for the maximum allowable operating pressure and thickness are based on the individual loading modules. PV Elite, on the other hand, provides a default set of loading combinations that comply with ASME's requirements. These combinations have been rigorously tested and are discussed in detail in the preceding section.

The design of atmospheric storage tanks against external wind loads governed by API-620 and API-650 considers the shell buckling. API-650 and EN 1993-1-6, specify that wind pressure varies both along the circumference and in height. However, due to the typical dimensions of tanks, wind pressure along the height can be assumed constant. This assumption raises questions about the accuracy and safety of the design, which require further investigation and analysis. In addition to tank overturning and wind pressure, the potential impact of debris or flying projectiles must also be considered in the design of atmospheric storage tanks. The impact force F_i of an object transported by strong winds can cause damage to the tank. The physical properties of the object and impact velocity are used to calculate the impact force. Therefore, it is essential to evaluate the potential impact of debris or flying projectiles when designing atmospheric storage tanks to ensure their structural integrity and safety. In summary, based on API 620, the fragility assessment of atmospheric storage tanks involves characterizing both the storage tank and natural hazard, it is also crucial to differentiate between damage and failure when assessing the vulnerability of atmospheric storage tanks. While damage refers to any deformation or impairment suffered by the tank, it does not necessarily imply a loss of containment. On the other hand, failure occurs when a crack or opening in the tank leads to a release of the stored material, which can be instantaneous in the case of a total collapse, [7].

New test results were obtained by modifying the loading conditions in PV Elite according to the API 620 procedure. Parameters are defined per **Error! Reference source not found.**.

Load Cases result analysis complying by API 620: The analysis of the six load cases shows that the stresses in the members vary depending on the combination of loads applied. Load Case 1 (EW+IP+HW) produces the highest tensile stresses at all nodes, with a maximum tensile stress of 20.99 at Node 200, and relatively high compression stresses, with a maximum compressive stress of 88.92 at Node 190. The members are not being loaded close to their tensile capacity, but they are being loaded closer to their compressive capacity. Load Case 2 (EW+WL+IP) results in lower stresses than Load Case 1, with the members being loaded closer to their tensile capacity and further from their compressive capacity. Load Case 3 (EW+WL+NP) produces the lowest stresses of the three load cases, with the members being loaded close to their tensile capacity and further from their compressive capacity. Load Case 4 (HW+IP+NP) includes only longitudinal stresses due to hydrotest pressure and internal pressure, with no weight load. The stresses are highest at the top and bottom nodes and lowest in the middle nodes, ranging from 119.62 MPa to 152.38 MPa in compression and from 0 MPa to 179.27 MPa in tension. Load Case 5 (EW+IP+HW+OW) includes longitudinal stresses due to weight, internal pressure, hydrotest pressure, and weight in operating condition. The stress values are generally higher than in Load Case 4, particularly at the top and bottom nodes, with tension stresses ranging from 88.92 MPa to 179.27 MPa and compression stresses ranging from 60.01 MPa to 67.43 MPa. Load Case 6 (EW+IP+HW+EQ) includes longitudinal stresses due to weight, internal pressure, hydrotest pressure, and bending stress due to earthquake moment in operating condition. The stress values are similar to Load Case 5, with slightly lower compression stresses and slightly higher tension stresses. Load Case 7 (EW+HP) produces a maximum tensile stress value of 26.66 at Node 210 and a maximum compressive stress value of 88.92 at Node 190. The stress values decrease as we move from Node 20 to Node 220, with the stress ratio decreasing accordingly. The stress values are generally higher than those for Load Case 4 but lower than those for Load Case 5 and Load Case 6. Load Case 7 is dominated by the longitudinal stress due to hydrotest pressure and the longitudinal stress due to weight. It also includes bending stress due to weight and longitudinal stress due to weight, which are not present in Load Case 4 and Load Case 5. However, Load Case 7 does not include any external pressure or earthquake moment loads, which are present in Load Case 5 and Load Case 6, respectively. In conclusion the tests demonstrated that the Governing Element is M5.

Comparing Absolute Maximum of all the stress ratio, under both codes, and considering their load case, it demonstrates that ASME VIII tests the structure under more sever conditions, also it demonstrates that the maximum stress ratio is extremely close to 1 and Element M5's likelihood to fail.

Considering the load case results, it can be concluded that ASME suggest more conservative approach with more flexibility, whereas API 620 offers a less conservative method with less flexibility.

6 CONCLUSIONS

In this paper, we carried out a comparison of two codes, API 620 and ASME VIII Division 1, by studying the design and analysis procedure of an elevated atmospheric tank as a case study. The design was performed by an experienced team in compliance with ASME VIII Div.1 which was outline in the paper and then the process was compared with API 620 guidelines.

By comparing the two codes, in Scope, material and design section, it was concluded that while both codes can be applied for an elevated atmospheric tank, they approach the issues of safety and reliability in different methods. ASME VIII-1 relies on a conservative approach while providing the designer with more flexibility to choose and apply, but API 620 is both less conservative and flexible. API 620 provides a set of simple instruction within a limited scope which is more suitable for engineers from diverse fields and even junior engineers. At the same time API 620 raises the reliability through rigorous sessions of tests which might not be appliable for many parts according to ASME VIII-1.

Ultimately, the purpose of this paper was to provide a detailed demonstration of design approach and the differences of using both codes to act as an experimental guideline for other designers and engineers.

This paper was aimed to provide details of design and analysis process for an elevated atmospheric tank complying by pressure vessel guidelines and all the references of certified handbooks were provided to ensure a thorough investigation of the methods are available. By having access to these approaches and a better grasp of the know-how regarding pressure vessels design, selecting an appropriate code for each unique approach is much more accessible.

The paper's scope was not considered for wider subjects such as high-pressure vessels, which is advised and encouraged for other engineers and researchers to pursue. By considering the fact that code application comparison, not only provides a more tangible guideline for new engineers and designers but also provide a more in-depth approach to prepare design papers, which wouldn't happen normally since most design related papers in the field of pressure vessel are concerned with specific mechanical design and less with design methodology. Also considering the fact that selection and applying a more reliable design method contributes largely to decreasing the likelihood of failure and catastrophic failure cases. Also, it is highly advised that by sharing more case studies from different engineers of different experiences and background, more data would be available for institutions like ASME to modify and redefine their approaches and codes, if necessary, in more specific instances.

Nonetheless this study could also be expanded into other sections of both ASME VIII-1 and API 620, like fabrication and inspection to address more known and unknown issues. By considering different objectives and intended benefactor of each code, a closer study of their mechanism and application using case studies is always beneficial.

In conclusion, the paper provided details about design of an elevated low-pressure vessel and covered most steps of design and analysis, applied by PVElite software. A comparison of both intended codes was also carried out and applied for each step within the scope of the paper. The results demonstrated a more in-depth vision of both codes and their reliability through risk evaluation of design parameters.

NOMENCLATURE

P	Pressure, psi.
C	corrosion allowance, in.
S	stress value of the material, psi.
E	Joint efficiency
IP	Longitudinal Stress due to Internal Pressure
EP	Longitudinal Stress due to External Pressure
HP	Longitudinal Stress due to Hydrotest Pressure
NP	No Pressure
EW	Longitudinal Stress due to Weight (No Liquid)
OW	Longitudinal Stress due to Weight (Operating)
HW	Longitudinal Stress due to Weight (Hydrotest)
WI	Bending Stress due to Wind Moment (Operating)
EQ	Bending Stress due to Earthquake Moment (Operating)
EE	Bending Stress due to Earthquake Moment (Empty)
HI	Bending Stress due to Wind Moment (Hydrotest)

HE	Bending Stress due to Earthquake Moment (Hydrotest)
WE	Bending Stress due to Wind Moment (Empty) (no CA)
WF	Bending Stress due to Wind Moment (Filled) (no CA)
CW	Longitudinal Stress due to Weight (Empty) (no CA)
VO	Bending Stress due to Vortex Shedding Loads (Ope)
VE	Bending Stress due to Vortex Shedding Loads (Emp)
VF	Bending Stress due to Vortex Shedding Loads (Test No CA.)
FW	Axial Stress due to Vertical Forces for the Wind Case
FS	Axial Stress due to Vertical Forces for the Seismic Case
BW	Bending Stress due to Lat. Forces for the Wind Case, Corroded
BS	Bending Stress due to Lat. Forces for the Seismic Case, Corroded
BN	Bending Stress due to Lat. Forces for the Wind Case, Uncorroded
BU	Bending Stress due to Lat. Forces for the Seismic Case, Uncorroded

APPENDICES:

Load Case 1:	NP+EW+WI+FW+BW	(A1)
Load Case 2:	NP+EW+EE+FS+BS	(A2)
Load Case 3:	NP+OW+WI+FW+BW	(A3)
Load Case 4:	NP+OW+EQ+FS+BS	(A4)
Load Case 5:	NP+HW+HI	(A5)
Load Case 6:	NP+HW+HE	(A6)
Load Case 7:	IP+OW+WI+FW+BW	(A7)
Load Case 8:	IP+OW+EQ+FS+BS	(A8)
Load Case 9:	EP+OW+WI+FW+BW	(A9)
Load Case 10:	EP+OW+EQ+FS+BS	(A10)
Load Case 11:	HP+HW+HI	(A11)
Load Case 12:	HP+HW+HE	(A12)
Load Case 13:	IP+WE+EW	(A13)
Load Case 14:	IP+WF+CW	(A14)
Load Case 15:	IP+VO+OW	(A15)
Load Case 16:	IP+VE+EW	(A16)
Load Case 17:	NP+VO+OW	(A17)
Load Case 18:	FS+BS+IP+OW	(A18)
Load Case 19:	FS+BS+EP+OW	(A19)

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