

EVALUATION OF DBF AND DBA METHODS FOR HEAD GEOMETRIES IN MULTI-FUNCTIONAL PMMA HYPERBARIC CHAMBERS

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Received: August 27, 2025; Revised: February 03, 2026; Accepted: March 02, 2026

Abstract

Pressure vessels are widely used in various industrial and commercial fields. These include diving cylinders, recompression chambers, and hyperbaric oxygen therapy chambers. Hyperbaric chambers are a special group of pressure vessels that operate at a specific pressure for treatment. In this study, analyses were carried out on the design of a cylindrical hyperbaric test chamber to be used for devices such as submarine equipment or a bedside monitors that require performance tests under pressure, using DBF and DBA methods. The body and head thicknesses calculated using the DBF method in accordance with ASME PVHO-1 standards were revised and reduced within the constraints and rules specified in the formulas. Thickness analyses of the chamber with a diameter of 800 mm and a length of 1500 mm were re-evaluated using the DBA method for two different head geometries (spherical and hemispherical), considering the safety conditions accepted in ASME PVHO-1 standards. PMMA (Polymethyl methacrylate) was chosen as the cabin material for its lightweight properties. The chamber was analysed for two different pressure values: 10 bar operating pressure (MAWP) and 16.5 bar test pressure (MAP). The analyses resulted in beneficial outcomes in terms of both manufacturing ease and cost. The hemispherical head geometry demonstrated superior performance than the spherical head geometry in terms of stress distribution, resistance to test pressure, geometric complexity, and safety factor. Consequently, the thickness values, which have been proven to be safe through analysis, can be used instead of the numerically obtained results.

Keywords: HBOT; ASME-PVHO; PMMA; Finite Element Method

Introduction

Hyperbaric chambers were first designed in the 18th century for medical applications, such as the treatment of decompression sickness. Today, these chambers are also used for different purposes, such as pressure tests. Hyperbaric chambers are

generally divided into two basic design types: monoplace (single occupancy) and multiplace (multiple occupancy). Monoplace chambers offer a structure optimised for the treatment of a single individual, while multiplace chambers have a larger

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DOI: <https://doi.org/10.55766/sujst11258>

Suranaree J. Sci. Technol. 33(4):010445(1-16)

interior space to allow multiple patients to be treated simultaneously. Modern hyperbaric chambers not only fulfil technical requirements but are also equipped with ergonomic features aimed at improving patient comfort. These features include adjustable seating and lying areas, temperature and humidity control, efficient ventilation systems and sound insulation. In addition, transparent windows and screens used to provide visual contact with the external environment stand out as important design components to reduce the risk of claustrophobia.

Hyperbaric oxygen chambers are also known to be used in animal models. The cabin can be adapted for use with pets by integrating additional equipment, while also providing a suitable environment for devices such as subsea equipment or bedside monitors that need to be performance-tested under pressure.

Regardless of the intended use of hyperbaric chambers, there are important factors to be considered in the design process. In the design phase, criteria, such as the properties of the materials used, stresses that may occur, elastic stability and aesthetic appearance of the cabin should also be taken into consideration. In addition, the cost of the cabin is an important factor when compared with the expected lifetime and the service provided (Lingaiyah, 2003). In this context, material selection is of great importance for the safe functioning of the cabinets under high pressure. In one study, durable materials, such as API 5L Grade 70 carbon steel, were preferred to ensure that the chamber can withstand high pressure (Sazali & Ovinis, 2014). In this study, the design process was analysed in detail using ANSYS Static Structural software. In another study, steel, aluminium and PMMA materials were compared in the production of hyperbaric chambers for pets and PMMA was shown to be lighter, portable and cost effective (Altinbalik & Kabak, 2016). As known a hyperbaric cabin can be designed using the rules of DBF and DBA (Diamantoudis & Kermanidis, 2005; Natal Jorge & Fernandes, 2001). On the other hand, finite element package programmes have an important place in the design of pressure vessels and are widely used (Natal Jorge & Fernandes, 2001; Widera & Wei, 1999; Lu *et al.*, 2014). It was emphasized by Hurd *et al.* (2023) that the ASME PVHO-1 standards should be advanced beyond the empirical design tables originating from the 1970s. In their study, it was demonstrated that the Design-by-Analysis (DBA) methodology allows for the optimization of material thicknesses without compromising structural integrity, while operational costs can be reduced. The reliability of simulation-based designs is recognized as a fundamental factor in the development of modern

ASME codes. In a study conducted by Kemper, (2023), the manner in which ASME implements VVUQ concepts for pressure vessels was examined; it was emphasized that the transparent and reliable presentation of simulation results is of critical importance in cases where physical testing is limited. The Design-by-Analysis (DBA) approach applied in the mentioned study was conducted in alignment with this modern framework of simulation credibility highlighted in the literature. A hybrid approach was adopted by Satya Prakash *et al.* (2025) to minimize stress concentrations in pressure vessel components, and the accuracy of numerical analyses (FEA) in design revisions was demonstrated. Mentioned study supports current design optimisation work in the literature by applying the methodology in question specifically to PMMA material and ASME PVHO-1 standards. It was demonstrated by Chapai (2023) that conventional empirical formulations (WRC-537) yield overly conservative results in the design of pressure vessel components, whereas Finite Element Analysis (FEA) performed in accordance with ASME Section VIII, Division 2 provides significantly more realistic and optimized design data. Furthermore, it was shown that, for complex geometries, numerical analyses based on the Design-by-Analysis methodology enhance design accuracy while enabling a reduction in unnecessary material usage. In a sample study finite element analysis (FEA) was carried out to obtain the elastic stress distribution at cylinder to cylinder junction in pressurized shell structures by Aseer Brabin *et al.* (2010).

In the present study, engineering design and structural analysis of a cylindrical hyperbaric chamber with a diameter of 800 mm and a length of 1500 mm, designed for a working pressure of 10 bar (145 PSI), has been carried out. Such a hyperbaric chamber is used for conformity testing of submarine equipment or devices, such as bedside monitors in multiplace cabinets, which require performance tests under pressure. Although *metal* materials are generally used in such chambers, PMMA was preferred in this study due to its lightness, transparency and economy. In this study, the body and head thicknesses calculated according to ASME PVHO-1 standards were changed in accordance with the DBA philosophy and the results were interpreted. In industrial applications, the shell thicknesses selected typically incorporate high safety factors, leading to unnecessary weight and increased manufacturing costs in both *metal*-based and polymer-based cabin designs. Moreover, there is a limited amount of literature addressing the selection of head type and head thickness for such cabins, indicating a significant knowledge gap in

this area. Therefore, during the design process, two different geometries, semi-spherical and spherical, were evaluated for the head section of the cabin, and both head types were compared in terms of production, cost and structural advantages.

Materials Properties and Calculation Basis

To evaluate the mechanical performance of PMMA, the basic material properties are presented in Table 1.

Table 1. Mechanical properties of PMMA material

Property	Value	Unit
Density	1.18	g/cm ³
Elastic Modulus	2.4 -3.3	GPa
Poisson's Ratio	0.35 – 0.4	-
Yield Strength	65	MPa
Max Tensile Strength	70 – 110	MPa
Transparency	Up to 92%	-

Design Principles and Theoretical Analysis

The HBO cabinet designed in this study is multifunctional. It can be used as a R&D cabinet for pets, such as cats, dogs or rabbits by adding related equipment, and it is also suitable for equipment whose performance under pressure is to be determined (e.g., submarine equipment or bedside monitor). Furthermore, such a pressurised chamber can be used to investigate the behaviour of DNA telomeres or damaged tendons under high pressure. For this purpose, a design was made to withstand cabin dimensions $D_i = 800$ mm, $L = 1500$ mm and $P = 10$ bar (145 PSI) pressure. Since the design priority was to examine the effects of body thickness and head type, the cabin was designed without a door. PMMA material was used in the designed HBOT. While AD2000 Merkblatt standards were used in theoretical calculations involving both *metal* and acrylic materials in previous studies (Altinbalik & Kabak, 2016), the ASME PVHO-1 standard plays a critical role today, especially in the design of a hyperbaric chamber (HBOT) using polymethyl methacrylate (PMMA) material. Therefore, in the presented study, calculations were made according to the ASME PVHO-1 standard.

The mechanical properties of PMMA vary with temperature. Especially in materials, such as *metal* and acrylic glass, the modulus of elasticity decreases, the yield strength decreases and the thermal expansion increases as the temperature

increases. This can affect the strength of a material operating under design pressure. In pressure vessel design, the design temperature is critical for the system to operate safely, for long life and in accordance with the standards. In this study, the operating temperature was to be 38°C according to the table taken from the ASME PVHO-1 standard. Especially in medical and hyperbaric systems, the indoor temperatures are generally expected to be between 22 and 30°C. However, due to light sources, electronic devices and the number of users, these temperatures can approach 38°C momentarily. A design temperature of 38°C (100°F) is commonly used in standards, such as ASME PVHO-1. This temperature value is based on many technical and scientific justifications (Wu *et al.*, 2023; Nasraoui *et al.*, 2009; American Society of Mechanical Engineers - ASME, 2019). This temperature is very close to the thermal comfort limit of the human body and is a limit temperature that can be easily taken as a reference in engineering applications. In addition, coefficients, such as CF (Conversion Factor) and critical pressure graphs in the ASME PVHO-1 standard are generally based on 38°C temperature.

Although cabin designs are commonly based on an ambient temperature of 38°C, in certain operating conditions, the surrounding temperature may increase up to 45°C. As shown in Table 2, the first standard temperature value above 38°C is 52°C. When the design is carried out based on an ambient temperature of 52°C, the required shell thickness is 23 mm according to the new CF value and the corresponding data presented in Figure 1. Since the initial head thickness used at the beginning of the analysis was 25 mm, it is concluded that the selected thickness is suitable for operating temperatures higher than 38°C. The glass transition temperature of PMMA is approximately 114°C; therefore, the use of PMMA as a structural material is not considered suitable for applications at temperatures exceeding 52°C. However, because the thermal expansion coefficient of PMMA is significantly higher than those of steel and aluminium, elastomeric seals that allow for free thermal expansion are recommended.

Calculation of the Main Body

When calculating the thickness of the main body according to ASME PVHO-1 standard, the CF value corresponding to the design temperature of 38°C is first selected from Table 2. CF value is an empirical ratio of short-term critical pressure to design pressure for a given temperature.

Table 2. Conversion Factors for Acrylic Spherical Sector Head, Hemispherical Shape for Applications (ASME, 2019)

Operational Pressure Ranges	Temperature °F (°C)				
	50(10)	75(24)	100(38)	125(52)	150(66)
N = 1 250psi (1.7 MPa)	CF = 13	CF = 14	CF = 15	CF = 20	CF = 25

Short-term critical pressure (STCP) is determined by multiplying the operational pressure (P) by a conversion factor (CF).

$$STCP = CF * P \quad (1)$$

$$STCP = 15 * 145$$

$$STCP = 2175 \text{ psi}$$

CF: conversion factor

P: Pressure

STCP: Short-term critical pressure

During the calculations, it should be remembered that 1 bar pressure is equal to 14.5 psi. As already mentioned, the maximum working pressure is chosen as 10 bar.

STCP value calculated according to formula no.1 is substituted in the graph given in Figure 1 and t/D_i value is read.

$$\frac{t}{D_i} = 0.1425 \quad (2)$$

$$\frac{t}{800} = 0.1425$$

$$t = 114 \text{ mm}$$

D_i = Inner diameter of the cabin

t = thicknesses of shell, less corrosion allowance, in.

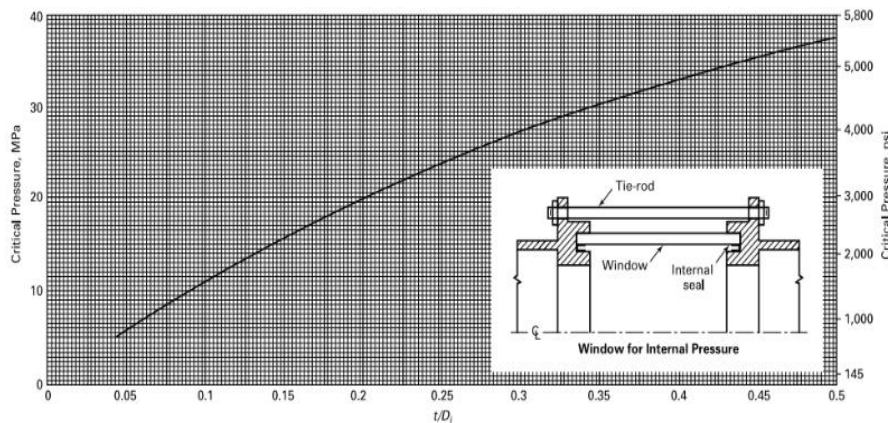
So the wall thickness of the main body was selected as 120 mm.

Calculations of the Heads

In this study, to optimise the structural strength and pressure distribution of the head section of the designed hyperbaric chamber, two head types were selected; hemispherical and spherical. Both head types offer different advantages and limitations in terms of structural performance, material utilisation, manufacturing difficulties and cost. Therefore, a comparison based on technical and economic criteria is necessary to determine which head geometry is more suitable for the design process. Table 3 presents a comparison of hemispherical and spherical heads in terms of basic engineering parameters.

Table 3. Comparison of Hemispherical and Spherical Heads in Terms of Structural and Manufacturing Parameters

Property	Hemispherical Head	Spherical Head
Pressure Resistance	High but lower than spherical head	Highest resistance
Material Usage	Requires more material compared to spherical head	Provides max strength with min material
Manufacturability	Easier to manufacture compared to spherical head	Difficult and costly to manufacture
Welding Process	Requires welding to connect with the cylindrical shell	Welding process is complex and expensive
Cost	Moderate	High cost
Transportation & Assembly	Easy to transport and assemble	Large structures may cause transportation difficulties

**Figure 1. Short-Term Critical Pressure of Cylindrical Acrylic Heads Pressurized Internally (ASME, 2019)**

Calculation of Hemispherical Head

Although the term ‘head’ is preferred in the tables and geometric classifications used in the ASME PVHO-1 standard, the use of this term is not limited to optical function. In fact, the elements defined under this heading act in many cases, as components, that directly carry structural loads and form a pressure boundary. In this context, the relevant geometries (e.g., hemispherical and spherical sector), although referred to as ‘window’ in the terminology of the standard, are the pressure vessel head due to their engineering functions and are treated as such in this study. These pressure-resistant transparent elements are mostly manufactured from transparent polymer materials (e.g., acrylic) with specific geometries, which are used for visual observation. However, the hemispherical and spherical sector forms used in the pressure vessels designed in this study are used for visual purposes and have a direct structural function. These spherical elements, which are structurally integrated into the cylindrical body, both carry the internal pressure load and form the pressure boundary by completing the closed volume. Hemispherical heads reduce stress concentration by providing a more even stress distribution against internal pressure and offer a cost advantage using less material compared to flat heads. Due to its geometry, this head provides maximum shape retention and a high safety factor. It also provides a long service life with less risk of fatigue damage under repeated pressure loading. For the theoretical wall-thickness calculations of the hemispheric head, the temperature value was determined first. Since the temperature value in the table taken from Table 4 of the ASME PVHO-1 standard for the main body was 38°C, the same temperature value was used. Under these conditions, the conversion factor (CF) was read from the table and then the head thickness was determined according to the formulae given below. On the other hand, the characteristics of the hemispherical head used in the hyperbaric chamber design are presented in Figure 2.

$$t \geq \frac{1}{2} \text{ in. (12.5 mm)} \quad (5)$$

$$\frac{t}{R_i} > 0.03 \quad (6)$$

$$D_o = (D_i + 2M) \quad (7)$$

$$R_m \geq \frac{1}{8} \text{ in. (3.2 mm)} \quad (8)$$

$$0.5 \text{ mm} \leq R_e \leq 0.125t \quad (9)$$

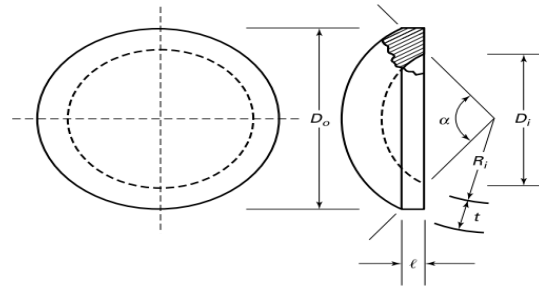


Figure 2. Hemispherical Head with Equatorial Flange

$$1.5t \leq M \leq 2.0t \quad (10)$$

$$R_o = R_i + t \quad (11)$$

R_i : Distance from the center of the spherical surface to the inner surface of the head.

R_o : Distance from the center of the spherical surface to the outer surface of the head.

R_m : The average of the inner and outer radii, depending on the specific application in the standard.

D_i : Diameter corresponding to the inner radius.

D_o : Diameter corresponding to the outer radius.

M or t : Thickness of shell, less corrosion allowance, in.

The theoretical calculation formulae and geometric ratios used in the design of the semi-spherical head are taken from ASME PVHO-1:2019 standard. In this study, the basic dimensions of the header (e.g., inner diameter and height) were predetermined according to the design requirements, and the formulas presented in the standard were used to calculate the required theoretical wall thickness according to the selected dimensions and to ensure structural control.

These formulae were also directly applied during the header geometry drawing in SolidWorks. In particular, parameters, such as inner radius (R_i), outer radius (R_o), average radius (R_m) and header depth, were determined by adhering to the ratios defined in the standard during the model creation. In the finite element analyses (FEA), the conversion factor (CF) value corresponding to the design temperature of 38°C was considered, and the safety of the design was evaluated by comparing theoretical thickness calculations with the analysis results (Table 4).

Table 4. Conversion Factors for Acrylic Spherical Sector Heads with Conical Edge, Hyperhemispherical Heads With Conical Edge, and NEMO-Type Heads With Conical Edge (ASME, 2019)

Operational Pressure Ranges	Temperature, °F (°C)				
	50(10)	75 (24)	100 (38)	125(52)	150(66)
N=1 2,500psi (17.2 MPa)	CF = 4	CF = 6	CF = 8	CF = 10	CF=16 1,500 psi (10.3 MPa)
N=2 5,000psi (34.5 MPa)	CF = 4	CF = 6	CF = 8	CF = 10 3,500 psi (24.1 Mpa)	3,000 psi (20.7 MPa)
N=3 7,500psi (51.7 MPa)	CF = 4				

$$\begin{aligned} TCP &= CF * P \\ STCP &= 8 * 145 \\ STCP &= 1160 \text{ psi} \end{aligned} \quad (3)$$

The t/D_i ratio corresponding to the 1160 psi value obtained as a result of the calculations was determined on the 180° curve in Figure 3. The reason for taking the 180° curve as a basis in the graph used in the calculations is that the semi-spherical head geometry corresponds to this angle and this form provides optimum load distribution and maximum structural strength under pressure. Since the curves for lower angles correspond to different geometrical forms and stress behaviours, calculations in hemispherical head designs are made only on the 180° curve.

$$\begin{aligned} \frac{t}{D_i} &= 0.0275 \\ \frac{t}{800} &= 0.0275 \\ t &= 22 \text{ mm} \end{aligned} \quad (4)$$

So the wall thickness of the hemispheric head was selected as 25 mm.

Calculation of Spherical Head

Among the head geometries that can be used in HBO chamber design, spherical sector heads stand out in terms of high compressive strength and structural efficiency. Similar to hemispherical heads, spherical heads minimise stress concentrations by providing homogeneous stress distribution under internal pressure. However, these heads are more difficult to manufacture and require special moulds and processes. For the spherical head, which is difficult to apply in large cabinets and is generally preferred in special designs, this head type was also selected to determine whether it would be suitable for use in the selected cabin dimensions. It provides long service life with low risk of fatigue damage under repeated pressure loading for limited applications.

Radius, geometric ratios and calculation formulas used in spherical head design are given in Figure 4 and the calculation formulae and geometric ratios used in spherical head design are taken from the ASME (2019) standard. In this study, the main dimensions of the head, such as inner diameter and depth were determined according to the design objectives, and the standard formulas were used to calculate the theoretical wall thickness suitable for the selected geometry and to check the structural adequacy. The same formulas were also applied during the drawing of the three-dimensional model of the spherical head in SolidWorks. In particular, details, such as the inner radius, angular arc, wall thickness and edge transition geometries of the head were modelled by adhering to the formal ratios defined in the PVHO-1 standard. In the analysis phase, FEA was applied considering the internal pressure, mechanical properties of PMMA and the

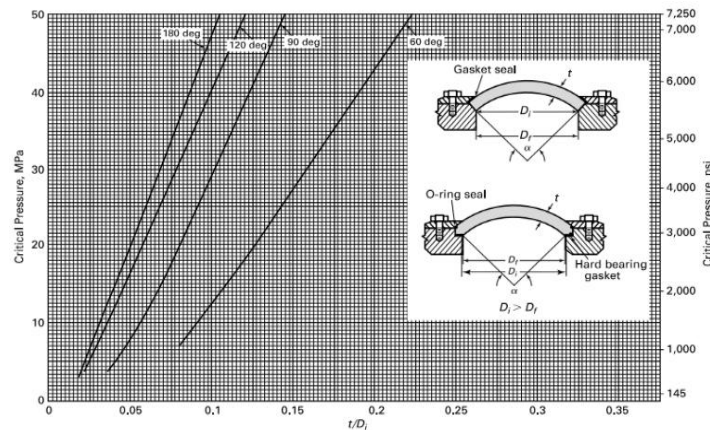


Figure 3. Short-Term Critical Pressure of Spherical Sector Acrylic Head (ASME, 2019)

conversion factor (CF) value corresponding to the design temperature of 38°C; the safety of the structure was verified by comparing the analysis outputs with theoretical calculations.

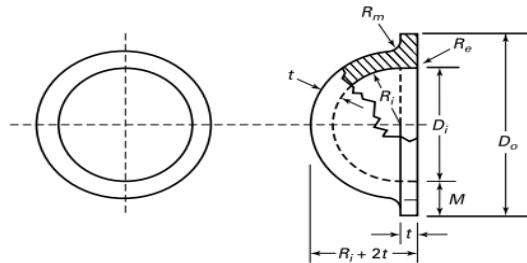


Figure 4. Spherical Sector Head with Square Edge

$$t \geq \frac{1}{2} \text{ in. (12.5 mm)} \quad (12)$$

$$30 \text{ deg} \leq a \leq 150 \text{ deg} \quad (13)$$

$$\frac{t}{R_i} > 0.03 \quad (14)$$

$$D_i = 2R_i \sin \frac{a}{2} \quad (15)$$

$$D_o = 2R_o \sin \frac{a}{2} \quad (16)$$

$$R_o = R_i + t \quad (17)$$

$$\ell = t \sin(90 \text{ deg} - \frac{a}{2}) \quad (18)$$

CF value is taken as 9 from Table 5 for 38°C temperature.

Table 5. Conversion Factors for Acrylic Spherical Sector Heads with Square Edge and Hemispherical Heads With Equatorial Flange (ASME, 2019)

Operational Pressure Ranges	Temperature, °F (°C)				
	50(10)	75(24)	100(38)	125(52)	150(66)
N = 1 2,500 psi (17.2Mpa)	CF=5	CF=7	CF=9	CF=11	CF=17 1,500 psi (10.3Mpa)
N=2 5,000 psi (34.5Mpa)	CF=5	CF=7	CF=9		3,000 psi (20.6 Mpa)
N = 3	CF=5				

$$\begin{aligned} STCP &= CF * P \\ STCP &= 9 * 145 \\ STCP &= 1305 \text{ psi} \end{aligned} \quad (19)$$

The t/D_i value is calculated using the 1305 psi value corresponding to the curve corresponding to 180 degrees from the Figure 2. As a result of this calculation, the head thickness is determined.

$$\begin{aligned} \frac{t}{D_i} &= 0.03 \\ \frac{t}{800} &= 0.03 \\ t &= 24 \text{ mm} \end{aligned} \quad (20)$$

So the wall thickness of the spherical head was selected as 25 mm.

Results and Discussion

For the Finite Element Analysis (FEA), the chamber was modeled with a Fixed Support at the lower base. Pressure loads of 10 bar and 16.5 bar were applied to the internal surface. The analysis employed high-precision, 10-node high-order quadratic elements. To accurately capture stress concentrations at the geometric discontinuity regions where the shell and head meet, a curvature-based mesh refinement was applied locally. The concentration of maximum stresses at the junctions is an expected mechanical behavior, and it was confirmed that the mesh is sufficiently fine to capture these stress gradients accurately.

Hemispherical Head

The theoretical wall thicknesses calculated according to ASME PVHO-1 standard were entered into the programme in SolidWorks environment and the stress distribution was obtained by applying 10 bar internal pressure, which is the working pressure of the designed cabinet. As seen in Figure 5, the maximum Von Mises stress value obtained as a result of the analysis is approximately 17.2 MPa, which is approximately one fourth of the yield strength of PMMA material. Moreover, the reported stress value corresponds to the maximum stress calculated by the analysis software. Examination of the color scale indicates that the stress in the main body is approximately 1.5 MPa, while the stress in the head section reaches 8.6 MPa. Accordingly, when evaluated based on the maximum stress obtained from the analysis, the cabin structure exhibits a safety factor of 3.78 but it shows that this thickness combination (120/25 mm) offers a high margin of safety from a structural point of view and at the same time may not be optimized in terms of cost and production. As it is, the cabin is too thick and too strong.

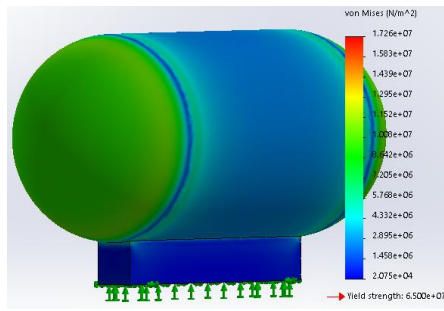


Figure 5. Stress Distribution Analysis of the Cabin with 120 mm Shell Thickness and 25 mm Hemispherical Head Thickness under 10 Bar Pressure

As a result of the analysis under 16.5 bar test pressure with 120 mm cylindrical body and 25 mm head thicknesses calculated according to PVHO-1 standard seen in Figure 6, the maximum Von Mises stress was approximately 29.8 MPa. This value corresponds to approximately 46% of the yield limit of the cabin material and as such, the cabin is 2.2 times safer. Similar to the results obtained in the initial analysis, the stresses acting on the shell and head regions in this analysis clearly indicate that the structure exhibits a significantly high safety factor. Although structural integrity was maintained, the stress increased by almost 60% compared to the working pressure analysis. This clearly shows the effect of test conditions on the design. However, the fact that the body thickness is quite high for both pressure conditions suggests that similar safety can be achieved with more optimized thicknesses.

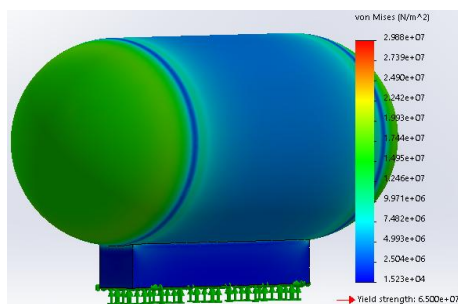


Figure 6. Stress Distribution Analysis of the Cabin with 120 mm Shell Thickness and 25 mm Hemispherical Head Thickness under 16.5 Bar Pressure

It is known that the average safety factor used in pressure vessels is generally selected between 1.5 and 4. According to the ASME standard, while the safety coefficient is taken as 4 in the old codes, it is generally in the range of 1.5 to 3.5 in the current ASME codes due to better knowledge of material

properties and sensitization of tests. The safety limit accepted by ASME PVHO-1 in the theoretical calculations for hyperbaric cabin manufacturing allows the maximum stress in the cabin wall to be 70% of the yield stress of the material (ASME, 2019). This means that the cabin can operate at least 1.5 times safer. This is a warning to the designers that the safety factor should not fall below this value. Since the coefficient of safety values obtained in previous analyses with this approach are high, it means that the main body of the cabin can operate at thickness values lower than the values obtained by theoretical calculations. However, considering the pressure levels involved in the operating conditions, it is evident that the shell thickness can be significantly reduced. To avoid time-consuming and iterative analysis procedures, the shell thickness was initially selected as 20 mm based on previous studies reported in the literature (Altinbalik & Kabak, 2016). The analysis results confirmed that the cabin remains structurally safe with a shell thickness of 20 mm; therefore, increasing the thickness beyond this value was considered unnecessary, as it would only lead to higher manufacturing costs without providing additional structural benefits. Moreover, although the cabinets are analysed with test pressure, in reality, only 60% of the test pressure is applied to a cabin. Thus, under an internal pressure of 10 bar, a design with a body thickness of 20 mm and a head thickness of 25 mm was planned and the cabinet was analysed.

Since the minimum thickness allowed in ASME standards is 12.5 mm, the first benchmark thickness value was 20 mm to be close to this value and the analyses were started from this value.

As seen in Figure 7, the maximum von Mises stress in the cabin was obtained as approximately 22.65 MPa, resulting in a safety factor of 2.87.

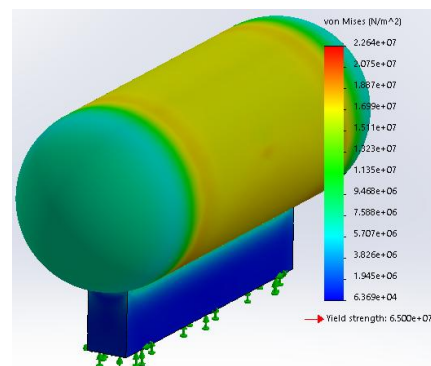


Figure 7. Stress Distribution Analysis of the Cabin with 20 mm Shell Thickness and 25 mm Hemispherical Head Thickness under 10 Bar Pressure

The maximum von Mises stress increased by approximately 32% compared to the shell thickness value calculated analytically (120 mm); however, the cabin still remains safe. The average stress was 17 MPa for the cabin shell and 5 MPa for the head section. Moreover, this reduced structure would provide significant advantages in terms of material usage and cost.

In Figure 8, the stress distribution in the cabin is shown when the test pressure of 16.5 bar is applied for the previously mentioned 20 mm shell thickness.

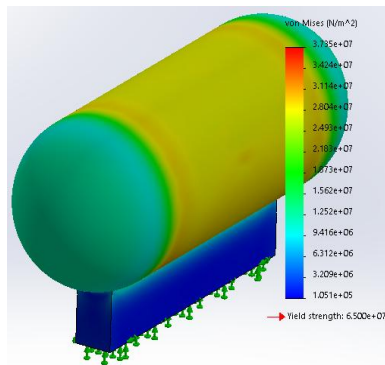


Figure 8. Stress Distribution Analysis of the Cabin with 20 mm Shell Thickness and 25 mm Hemispherical Head Thickness under 16.5 Bar Pressure

Examination of the colour scale indicates that the average stress is approximately 28 MPa in the shell region and around 10 MPa in the head region. The maximum stress of 37.3 MPa corresponds to 57.5% of the material's yield strength, and the safety factor remains higher than 1.5. Structural safety is ensured even under the test pressure with this thickness design. Even when the pressure is increased to 16.5 bar, a level not expected to be reached under operating conditions, the safety factor is calculated as 2.32 for the shell and 5.0 for the head. However, compared to the previous analyses, the stress has increased significantly, approaching the optimum level acceptable according to ASME standards.

The study was extended to investigate the effects of reducing the head thickness on the stress distribution, while keeping the shell thickness and head geometry constant after the reduction in shell thickness. Figure 9 shows the analysis results for the new design with a shell thickness of 20 mm and a head thickness of 15 mm under an operating pressure of 10 bar.

Examination of the colour scale indicates that the average stress is approximately 18 MPa in the

shell region and around 11 MPa in the head region. The maximum stress of 23 MPa corresponds to 35.5% of the material's yield strength, resulting in a safety factor of 2.8. Since there is neither a detachable nor a permanent joint located at the point where the maximum stress occurs, and the safety factor is high, this condition is not expected to pose a significant structural concern.

Despite the reduction of the head thickness from 25 mm to 15 mm, the cabin remains highly safe, and this optimization is feasible from a design perspective. Moreover, significant material savings are achieved for both the head and shell thickness values.

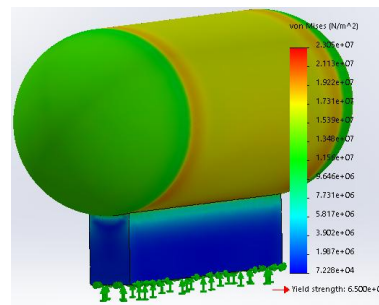


Figure 9. Stress Distribution Analysis of the Cabin with 20 mm Shell Thickness and 15 mm Hemispherical Head Thickness under 10 Bar Pressure

Examination of the colour scale indicates that the average stress is approximately 18 MPa in the shell region and around 11 MPa in the head region. The maximum stress of 23 MPa corresponds to 35.5% of the material's yield strength, resulting in a safety factor of 2.8. Since there is neither a detachable nor a permanent joint located at the point where the maximum stress occurs, and the safety factor is high, this condition is not expected to pose a significant structural concern.

Despite the reduction of the head thickness from 25 mm to 15 mm, the cabin remains highly safe, and this optimization is feasible from a design perspective. Moreover, significant material savings are achieved for both the head and shell thickness values.

Figure 10 shows the stress distribution when the test pressure is applied to the cabin with a shell thickness of 20 mm and a head thickness of 15 mm. As seen in the figure, the average stress in the shell is 28 MPa, which is consistent with Figure 8 (showing the 16.5 bar and 20/25 mm combination). The average stress in the head region is around 16 MPa. The maximum von Mises stress is approximately 38 MPa, corresponding to 58.5% of

the material's yield strength. Although the safety factor appears to have decreased, this thickness combination also passed the test pressure since the cabin will be subjected to operating pressure, and the ASME limit of 70% has not been reached. Structural safety is maintained, although it is observed that the safety margin has narrowed compared to previous configurations.

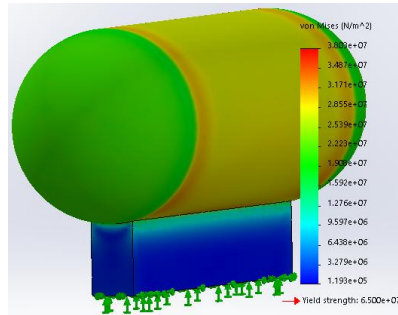


Figure 10. Stress Distribution Analysis of the Cabin with 20 mm Shell Thickness and 15 mm Hemispherical Head Thickness under 16.5 Bar Pressure

In this study, the minimum thicknesses of hemispherical and spherical heads were determined according to the conditions specified in Equations 5 and 12 of the ASME (2019) PVHO. Therefore, as the final design combination for the hemispherical head, the analysis results for shell and head thicknesses of 12.5 mm under 10 bar operating pressure and 16.5 bar test pressure are presented in Figures 11 and 12, respectively.

Examination of Figure 11 shows that the average stress in the shell is 27 MPa and in the head it is 17 MPa. The maximum von Mises stress obtained in this analysis is 34.30 MPa, which is the highest stress value occurring under operating pressure, and even in this case, the cabin has a safety factor of 1.9. Figure 12 presents the analysis results under the test pressure of 16.5 bar for the same thickness combination. The stress is 47 MPa in the shell and 30 MPa in the head, with a maximum stress of 56.6 MPa. This value corresponds to 87% of the material's yield strength. For brittle polymeric materials, such as PMMA, it is recommended that the maximum von Mises stress does not exceed 70% of the material's yield strength to ensure safe operation of the structure. Although yielding may not theoretically occur in the 12.5/12.5 mm hemispherical head design combination, it indicates that the structure is at risk of unexpected failure, especially under test pressure. When the shell and head are evaluated as an integrated structure, the stresses for both

pressure cases are observed to vary between 26–43 MPa in the shell and 19–33 MPa in the head. However, the maximum stress value reported by the analysis must be taken as the governing criterion. Accordingly, the considered thickness combination is deemed acceptable under operating pressure, while it is assessed as risky under test pressure. Therefore, this thickness combination is acceptable under operating pressure but is risky under test pressure. For this reason, these thicknesses can only be temporarily used for a constant operating pressure of 10 bar, but it is difficult for them to receive design approval (Table 6).

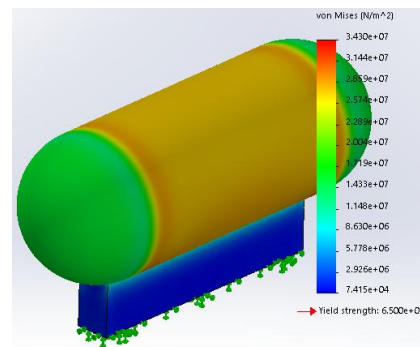


Figure 11. Stress Distribution Analysis of the Cabin with 12.5 mm Shell Thickness and 12.5 mm Hemispherical Head Thickness under 10 Bar Pressure

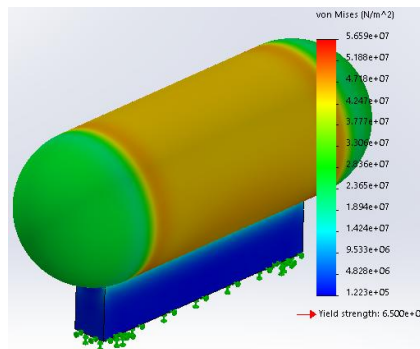


Figure 12. Stress Distribution Analysis of the Cabin with 12.5 mm Shell Thickness and 12.5 mm Hemispherical Head Thickness under 16.5 Bar Pressure

The table presents the stress values under operating and test pressures for the thickness combinations calculated and recommended using the DBF and DBA methods for the hemispherical head. As detailed in the analysis Figures, all combinations are extremely safe and usable under operating pressure conditions. Even at the minimum

thickness accepted by ASME, the cabin remains 1.9 times safer than required under operating pressure conditions. However, when both the shell and head thicknesses are 12.5 mm, the maximum stress under test pressure reaches 87% of the material's yield strength, making it unlikely to receive manufacturing approval in this configuration.

Spherical Head

Similar to the hemispherical head application, the spherical head application aimed to perform stress analyses for comparable shell and head thicknesses to compare both head options. As shown in Figure 13 the maximum von Mises stress at the most critical location was calculated as 28.23 MPa for the cabin with a shell thickness of 120 mm and a spherical head thickness of 25 mm under an internal pressure of 10 bar. This value corresponds to approximately 44% of the yield strength of PMMA, thus remaining within safe limits. However, it was observed that the stress obtained was approximately 64% higher compared to the value from the hemispherical head analysis with the same thickness. As in the test pressure analysis, no significant change was observed in the stress distribution of the main shell at this pressure level. However, while the average stress distribution in the hemispherical head was 7 MPa, it increased to 14 MPa in the spherical head. Although the stress level remains within safe limits, this indicates that the spherical head form is less advantageous in terms of stress concentration even under operating pressure. As shown in Figure 14, in the analysis conducted with a shell thickness of 120 mm and a spherical head thickness of 25 mm, calculated per the ASME PVHO-1 standard, the maximum von Mises stress under the test pressure of 16.5 bar was 42.28 MPa.

This value corresponds to approximately 65% of the yield strength of PMMA and remains within acceptable design safety limits. However, while the stress level in the hemispherical head analysis with the same thicknesses reached only 42%, it increased to 65% in this analysis, indicating that the spherical head geometry causes higher local stresses. This

result clearly demonstrates the effect of geometric shape on stress distribution. As only the head was changed, there was no significant change in the stress distribution of the main shell, as expected. Under the same conditions, the average stress distribution in the hemispherical head was around 10 MPa, whereas in the spherical head it was approximately 17 MPa. Although the cabin remains within safe limits, this indicates that even at theoretical thicknesses, the spherical head structure is subjected to higher stress levels.

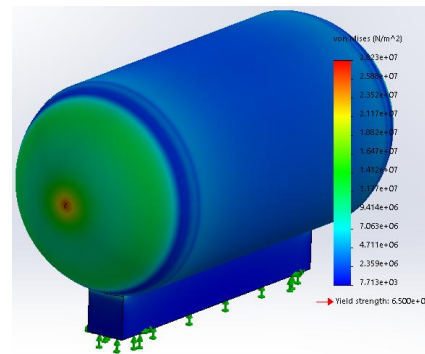


Figure 13. Stress Distribution Analysis of the Cabin with 120 mm Shell Thickness and 25 mm Spherical Head Thickness under 10 Bar Pressure

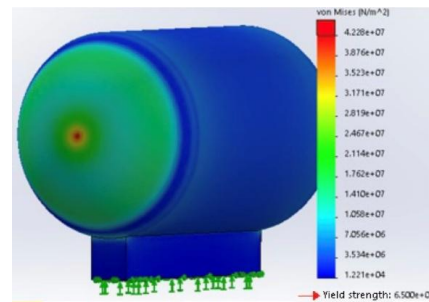


Figure 14. Stress Distribution Analysis of the Cabin with 120 mm Shell Thickness and 25 mm Spherical Head Thickness under 16.5 Bar Pressure

Table 6. Hemispherical Head Stress and Safety Summary Table

Pressure (Bar)	Shell Thickness (mm)	Head Thickness (mm)	Max Von Mises Stress	Ratio to Yield Stress	Safety Factor	Safety Status
10	120	25	17.2	26.5	3.77	Very Safe
16.5	120	25	27.4	42.2	2.37	Safe
10	20	25	22.5	34.8	2.88	Safe
16.5	20	25	37.35	57.5	1.74	Safe (Near Limit)
10	20	15	23.05	35.5	2.81	Safe
16.5	20	15	38.03	58.5	1.7	Safe (Near Limit)
10	12.5	12.5	34.3	52.8	1.90	Safe
16.5	12.5	12.5	56.59	87.1	1.14	Risky (Limit Exceeded)

When the color scale presented in Figure 15 is examined in the analysis conducted under an operating pressure of 10 bar with a shell thickness of 20 mm and a spherical head thickness of 25 mm, the maximum von Mises stress was 50% higher compared to the hemispherical head application.

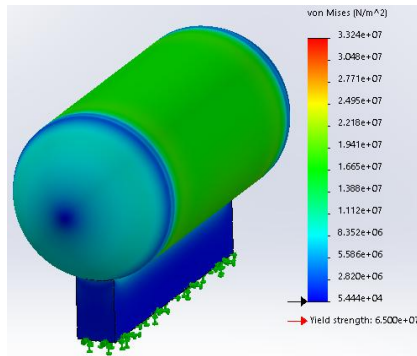


Figure 15. Stress Distribution Analysis of the Cabin with 20 mm Shell Thickness and 25 mm Spherical Head Thickness under 10 Bar Pressure

This reduced the safety factor from 3 in the hemispherical head to 2 for this head geometry; however, the cabin still remains safe in this configuration. The stress analysis results indicate that the cabin is safe and usable even under test pressure with this thickness combination. The design in this state meets the ASME PVHO-1 acceptance criteria. This value provides a good balance between optimum thickness and strength, suggesting that the head thickness could potentially be reduced further while still meeting the acceptance criteria.

In the analysis conducted at a test pressure of 16.5 bar with a shell thickness of 20 mm and a spherical head thickness of 25 mm, the maximum von Mises stress was 42.28 MPa, as shown in Figure 16. In such applications, the head geometry is a critical parameter; while the shell thickness affects the overall rigidity, the maximum stress point usually occurs in the head.

Changing the head geometry increased the maximum stress by 41% for the same thickness combination. Head thickness and geometry are critical parameters that directly affect cabin safety. Furthermore, in spherical heads, the highest stress concentration is generally observed either at the center of the head or near the flange connection region. As is well known from practical applications, the occurrence of maximum stress at the shell-head junction or at the center of the head is not considered to adversely affect the safety of the

vessel. However, the stress analysis results indicate that the cabin is safe and usable even under test pressure with this thickness combination. The design in this state meets the ASME PVHO-1 acceptance criteria.

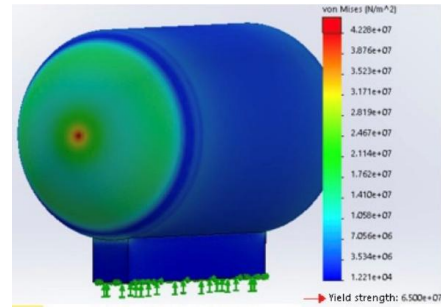


Figure 16. Stress Distribution Analysis of the Cabin with 20 mm Shell Thickness and 25 mm Spherical Head Thickness under 16.5 Bar Pressure

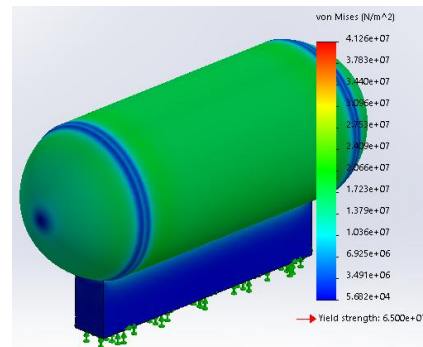


Figure 17. Stress Distribution Analysis of the Cabin with 20 mm Shell Thickness and 15 mm Spherical Head Thickness under 10 Bar Pressure

As shown in Figure 17, in the analysis conducted under an operating pressure of 10 bar with a shell thickness of 20 mm and a spherical head thickness of 15 mm, the maximum von Mises stress was 41.26 MPa. This value corresponds to 63% of the yield strength of PMMA and remains below the safe limit of 70%. Although the cabin is considered safe under these conditions, the high stresses induced by the spherical geometry provide a narrower safety margin than those with hemispherical heads of the same thickness. Furthermore, as explained in Figure 18, the cabin with this geometry and dimensions failed the test pressure analysis. This indicates that the structure is operating at a configuration that requires continuous monitoring. Such a head-thickness combination could be used in applications that do

not require extensive pressure cycling, such as single-use pressure tests for submarine equipment, validation tests for medical devices like patient monitors, or laboratory experiments with a limited number of repetitions.

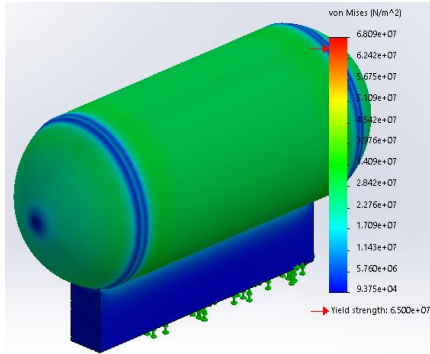


Figure 18. Stress Distribution Analysis of the Cabin with 20 mm Shell Thickness and 15 mm Spherical Head Thickness under 16.5 Bar Pressure

The next step in applying the thickness combinations used for hemispherical heads to spherical heads is the case with a shell thickness of 20 mm and a head thickness of 15 mm. When the head thickness was reduced to 15 mm, the maximum stress obtained in the analysis was 68 MPa, which exceeds the yield strength of the cabin material (104.8%). This indicates that the analysed structure would exhibit yielding behaviour or at least sustain significant damage under test pressure. The cabin is designed to operate under a maximum pressure of 10 bar and will not be exposed to 16.5 bar under operating conditions.

However, as previously explained, ASME requires the test pressure to be 1.5 times the operating pressure plus an additional 10%. In light of this information, this thickness combination is not acceptable from an engineering perspective for spherical heads. Particularly because spherical geometry provides a less advantageous stress distribution compared to hemispherical heads of the same thickness, the risk of failure is increased.

Table 7 presents the stress values under operating and test pressures for the thickness combinations calculated and recommended using the DBF and DBA methods for the spherical head. As detailed in the analysis figures, all combinations are usable under operating pressure conditions.

However, the combination of a shell thickness of 20 mm and a head thickness of 15 mm failed the test pressure application. Furthermore, since the maximum stress obtained under test pressure for the 15 mm head thickness exceeded the yield strength of the cabin material, the thickness combination of 12.5 mm shell and 12.5 mm head, which was analysed for the hemispherical head as shown in Figures 11 and 12, was not analysed for the spherical head.

Based on the numerical results presented in Tables 6 and 7, the strengths and weaknesses of two different head geometries with respect to five comparison parameters are summarized in an integrated manner in Table 8. Considering the five primary comparison criteria, as also shown in Table 8, the hemispherical head geometry demonstrates superior performance compared to the spherical head geometry in terms of stress distribution, resistance to test pressure, geometric complexity, and safety factor.

Table 7. Spherical Head Stress and Safety Summary Table

Pressure (Bar)	Shell Thickness (mm)	Head Thickness (mm)	Max Von Mises Stress	Ratio to Yield Stress	Safety Factor	Safety Status
10	120	25	28.23	43.4	2.3	Safe
16.5	120	25	42.28	65.0	1.53	Safe (Near Limit)
10	20	25	33.24	51.1	1.95	Safe
16.5	20	25	42.28	65.0	1.53	Safe (Near Limit)
10	20	15	41.26	63.5	1.57	Safe (Near Limit)
16.5	20	15	68.09	104.8	0.95	Fail (Yield Limit Exceeded)

Table 8. Structural and Design Comparison of Hemispherical and Spherical Heads

Property	Hemispherical Head	Spherical Head
Stress Distribution	More uniform and lower	Higher and more concentrated
Material Consumption	Slightly higher	More compact but risky due to high stress
Resistance to Test Pressure	Safe up to 12.5 mm thickness	Risky at 15 mm thickness and below
Geometric Complexity	More symmetrical and relatively easy to manufacture	Requires critical calculations for geometric compatibility
Safety Factor	Higher	More limited

The results obtained for both spherical and hemispherical heads, as well as the proposed thickness values, are consistent with the literature and have been evaluated as favourable and feasible by an industrial company engaged in practical applications related to this topic.

Conclusions

In this study, the structural behaviour of hemispherical and spherical head geometries and the cylindrical shell used in the design of hyperbaric oxygen (HBO) chambers was investigated under different thickness combinations using the DBF and DBA methods. It was demonstrated that supporting the analytical calculations in standards with engineering analyses enables designs to be both economical and safe. The standards preferred for pressure vessel design vary from country to country. Commercial manufacturers adopt the standards that are accepted in the country where the design is intended to be used. In the United States, ASME Section I and ASME Section II standards are predominantly employed, whereas the AD 2000 Merkblatt standard is used in Germany, and the EN 13445 standard is widely applied in other European countries.

Although these standards involve certain design differences, such variations are primarily based on differing approaches to safety factors. In the present study, a design methodology that is expected to be acceptable under all of these standards was adopted. Instead of conservative designs leading to high manufacturing costs, the analyses demonstrated that pressure vessels with identical geometries can be designed to be lighter and more cost-effective while maintaining structural safety. Furthermore, the analyses were repeated for two different head geometries.

The theoretical thickness values obtained from the formulas specified in the ASME PVHO-1 standard were taken as reference in the numerical analyses. In this reference configuration, very low von Mises stresses were obtained for both head types under both 10 bar operating pressure and 16.5 bar test pressure. The safety factor for the thickness values obtained from analytical calculations ranged between 3.77 and 1.53. This indicates that, particularly for hemispherical head selection, standard calculations provide a high safety margin but may also lead to unnecessary material use. In the second stage, combinations involving reduction in shell and head thicknesses were evaluated. Since the minimum thickness allowed by ASME standards is 12.5 mm, the first comparative thickness value was 20 mm due to its

proximity to this limit, and the analyses were initiated from this value. Considering the values permitted by ASME, analyses were conducted by reducing the shell thickness to 20 mm and the head thicknesses to 25 mm and 15 mm. For hemispherical heads, the maximum stress values obtained under both operating and test pressures ranged between 35% and 58% of the material's yield strength, thus meeting the safety criteria. For spherical heads with the same shell and head thickness values, this range was between 51% and 105%. As indicated in the tables, the spherical head with a thickness of 15 mm exceeded the safe limits under test pressure. In contrast, for the same 15 mm head thickness and 16.5 bar test pressure, the hemispherical head remained safe with a safety factor of 1.7. This result revealed that the spherical head geometry causes higher local stresses and should be evaluated carefully, especially at lower thicknesses. In conclusion, the analyses demonstrated that for the safe operation of the design, thickness and head geometry is a critical parameter.

In the final stage, analyses were conducted with the shell and head thicknesses set to the minimum standard value of 12.5 mm. For the hemispherical head, a safety factor of 1.92 was obtained under operating pressure and 1.15 under test pressure. For brittle polymeric materials, such as PMMA, the safety factor accepted by ASME for a safe design is 1.42, indicating that this configuration should not be considered safe under test conditions.

In conclusion, both theoretical calculations and finite element analyses have demonstrated that thickness selection plays a critical role in ensuring structural safety. Compared to the theoretical values of 120 mm shell thickness and 25 mm head thickness, thickness combinations of 20 mm for the shell and 15-25 mm for the head support safe operation and contribute to material and cost optimization.

As a result, it has been demonstrated that the shell thickness obtained as 120 mm can be reduced to 20 mm based on the literature review and the conducted analyses. Analyses were also carried out for the minimum allowable design thickness of 12.5 mm; however, the use of this shell thickness is not recommended. The selection of head thickness varies depending on which pressure value is considered as the governing design criterion by practicing engineers. In the case of a spherical head, a head thickness between 15 mm and 25 mm can be safely selected for an operating pressure of 10 bar, whereas for a MAP of 16.5 bar, it is not recommended to reduce the thickness below 25 mm. When a hemispherical head geometry is

selected, engineers are provided with a broader range of design options. No significant difference is observed between spherical and hemispherical head geometries with respect to shell thickness selection. However, for head thickness selection, values ranging from 15 mm to 25 mm can be safely preferred for both pressure levels (10 bar and 16.5 bar). As a result of this study, a different perspective has been provided, facilitating the work of design engineers on the subject.

When evaluated comprehensively, hemispherical heads are structurally more advantageous. This study has demonstrated that the hemispherical head geometry should be preferred in hyperbaric chambers, especially for applications requiring high safety.

The shell thicknesses calculated in accordance with the relevant standards for hyperbaric chamber design are resistant to 10,000 cycles.

The hyperbaric chambers designed for the presented study are typically used once per day, depending on the daily schedule. In the present study, the proposed shell and head thicknesses were determined to withstand a working pressure of 10 bar under actual operating conditions. Considering this pressure and the proposed minimum thicknesses, the maximum stresses acting on the chamber are presented in Tables 6 and 7. Based on fatigue tests reported in the literature for PMMA material, the chamber's fatigue life is estimated to be approximately $n = 10,000$ cycles, depending on the stresses to which it is subjected, and is therefore compliant with current standards. Furthermore, in a study utilizing 40 years of field data and nonlinear Finite Element Analysis (FEA) results, Hurd *et al.* (2023) demonstrated that PMMA windows exhibit significantly higher durability than the current design limits (10 years / 10,000 cycles).

Fatigue tests need to be repeated for the thicknesses proposed by the DBA method in this study, so that new service life recommendations should be developed accordingly in future studies.

Funding

The authors received no specific financial support for the research, authorship, and/or publication of this article.

Author Contributions

Conceptualization: M.D.D., M.T.A.; Methodology: M.D.D., M.T.A.; Investigation: M.D.D., M.T.A.; Formal Analysis: M.D.D., M.T.A.; Data Curation:

M.D.D., M.T.A.; Writing - Original Draft: M.D.D., M.T.A.; Writing - Review & Editing: M.D.D., M.T.A.; Supervision: M.T.A. All authors have read and agreed to the published version of the manuscript.

Ethics Statement

Ethical approval was not required for this study because it did not involve human participants, animals, identifiable personal data, or materials requiring institutional ethical review.

Declaration of Generative AI Use

The authors used Generative AI tools solely for grammar checking and improving the writing flow of the manuscript. No AI-generated content was used in the scientific design, data analysis, interpretation of results, or conclusions. The authors take full responsibility for the accuracy, integrity, and originality of the manuscript.

Conflict of Interest

The authors declare that they have no conflicts of interest related to this work.

Data Availability Statement

All relevant data supporting the findings of this research are included in the manuscript; no additional data are required.

Acknowledgement

The authors would like to express their sincere gratitude to Osman ÇALPAN and HPOTECH for their valuable support and contributions to this study.

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