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DESIGN AND STATIC ANALYSIS OF PRISMATIC PRESSURE VESSELS ACCORDING TO TS 13445-3 STANDARD

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ABSTRACT

The focus of the presented work is on the design and analysis of prismatic pressure vessels. Engineering calculations, design parameters and performance criteria of pressure vessels are available in the literature in detail, but there are deficiencies in the evaluation of the manufacturing processes of prismatic pressure vessels and the mathematical formulae required by these processes in the light of standards. In this context, it was investigated how a special prismatic pressure vessel, a square profile hyperbaric oxygen chamber, could be optimised in terms of structural strength, stress distribution and safety requirements. The material selection and design of such a chamber has been evaluated using DBA and DBF methods. The performance of the prismatic hyperbaric oxygen chamber in current applications is explained, supported by various engineering analyses and simulations. It was observed that the cabin designed in accordance with TS 13445-3 standards is 2 times safe under operating conditions and the amount of displacement is commercially acceptable. It is shown with SolidWorks analysis.

1. INTRODUCTION

The sizes and shapes of pressure vessels cover a wide range, from large cylindrical vessels to parts in aeroplanes with small hydraulic units. Efforts to produce steam in boilers, which led the industrial revolution in the 1800s, formed the basis of today's pressure vessels [1]. A pressure vessel is defined as a container with a pressure difference between the inside and outside. In most cases, the internal pressure is higher than the external pressure. Pressure vessels are used to store and transport liquids, vapours and gases under high pressure. Such vessels are used for various purposes both in industry and in specialized applications; for example, diving cylinder, recompression chamber, reactor technology, chemical industry, distillation towers, submarine habitats and hyperbaric oxygen treatment chambers [2]. While some pressure vessels may be buried underground or at ocean depths, most are placed on the ground or on supporting platforms. These vessels are usually spherical or cylindrical in shape with dome-shaped ends [3].

Hyperbaric chambers are a special type of pressurised vessel used for various treatment processes of humans and animals. These chambers can adjust or change the internal pressure inside the container. Unlike other pressurised vessels, it is possible to add or remove an object while hyperbaric chambers are under pressure. Although science-based applications of hyperbaric technology are a relatively recent development, the use of compressed gas in medicine has a much longer history.

Although hyperbaric air began to be used as early as 1662, Paul Bert, the pioneer of pressure physiology, laid the scientific foundations of oxygen toxicity in 1878. Throughout history, hyperbaric chambers have been manufactured from various materials and the internal pressure has been controlled by different methods. The development of welding and compressor technologies has led to the evolution of the design of modern hyperbaric chambers [4]. Pressure vessels can be manufactured in various shapes, but spherical, cylindrical and conical forms are generally preferred in the industry. The spherical shape provides an ideal structure by distributing the internal pressure with equal stresses on both the inner and outer surface, and this provides an advantage in terms of structural strength compared to a cylindrical pressure vessel of the same thickness. However, the manufacturing and cost of the spherical shape can present some challenges. For this reason, the cylindrical shape, which offers advantages in terms of lower production costs and space utilisation, is more widely preferred [5].

In a study, a method is proposed to estimate the design pressure in prismatic storage tanks for liquefied natural gas. Two main types of vessels are used for gas transport: spherical and self-supporting MOSS type vessels and membrane type vessels. While the use of MOSS vessels is decreasing, the use of membrane vessels is increasing. This is due to the advantage of membrane type vessels, which provide more economical and high volume efficiency in limited space [6]. The volume efficiency of conventional

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pressure vessels is about 25% to 50% lower compared to prismatic pressure vessels, depending on their installation area. In the literature, simplified formulae developed for prismatic pressure vessels are presented in a way to show general validity for different sizes and thicknesses. In order to test the accuracy of these formulae, case studies were carried out, which were compared with Finite Element Method (FEM) results [7]. In one study, design guidelines were proposed for a rectangular container at 20 psi pressure [8]. In another study, a non-circular vessel was designed for waste storage under 3 bar pressure [9]. These studies have influenced Section VIII, Division 1 and Division 2 of the ASME Boiler and Pressure Vessel Code [10,11].

In the article written by Lam, Fontaine R., Ross F.L., and Chiu E.S., it was emphasised that hyperbaric oxygen therapy (HBOT) through the administration of 100% oxygen at pressures greater than 1.4 atmospheres contributes to the improvement of oxygenation and neovascularization and reduces inflammation in chronic wounds. The article cited a growing body of research supporting the effects of HBOT in accelerating wound healing and reducing the risk of adverse events such as amputation. However, it was stated that HBOT is not sufficiently recognized by many practitioners. In this context, the article provides a general introduction to HBOT, detailing the physiological and mechanistic processes behind the treatment and comprehensively discussing the current indications for HBOT [12].

The book, edited by Enoch Huang, covers in detail the general design principles of hyperbaric chambers and various application areas of oxygen therapy. In addition, the standards set by the American Society of Mechanical Engineers and Pressure Vessels (ASME-PVHO-1) for pressure vessels and the codes and standards recommended by the National Fire Protection Agency (NFPA 99) for hyperbaric chambers are comprehensively examined [13].

In the study by Kemper, Richards, Nappi, Thipparthi and Escobar, it was noted that Section VIII of the Boiler and Pressure Vessel Code introduces the use of acrylics as pressure vessel materials. It was also highlighted in this study that an ASME Codes and Standards task group is developing a ‘design by analysis’ (DBA) methodology for acrylics and other glassy polymers for pressure vessel components. The proposed DBA methodology proposes to make extensive use of Verification and Validation (V&V) techniques and Finite Element Method (FEM) in the design process to improve the effectiveness of glassy polymers as pressure vessel materials [14].

2. EXPOSITION

This study includes the strength calculations of the Square Type Hyperbaric Oxygen Therapy Chamber manufactured in accordance with the Pressure Vessels Directive 97/23/EEC, in accordance with EN 13445-3 Design Standards.

The pressure vessel for which design calculations are made is Hyperbaric Oxygen Therapy System and the operating and test conditions of the system are as follows

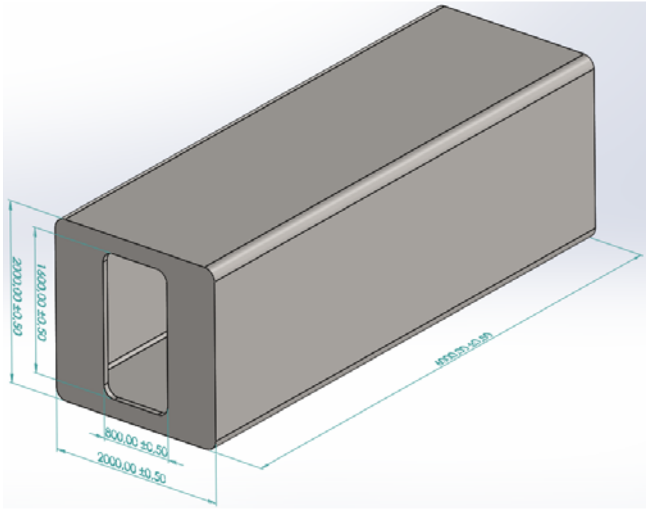


Fig. 1. Schematic view of pressure vessel

Table 1 *Operating Conditions of the System*

Design Pressure	*5,5 Bar*
Test Pressure	*8,25 Bar*
Material Code	*P 355 GH*
$R_{p_{0.2/t}}$	$315\ N/mm^2$
$R_{m/t}$	$470\ N/mm^2$
Standard Used	*DIN EN 13445-3*
Measure for corrosion	*3 mm*

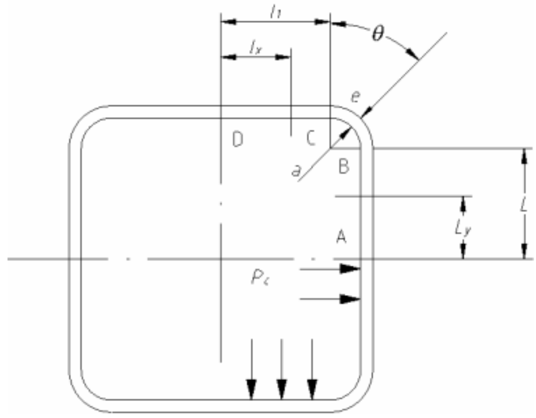


Fig. 2. Unreinforced vessels

$$f_d = min\left(\frac{R_{p_{0.2/t}}}{1.5};\frac{R_{m/20}}{2.4}\right)$$

$$f_d = min(210;195.8)$$

$$f_d = 195,8\,N/mm^2$$

$$f_{test} = \frac{R_{p_{0,2/t}}}{1,5}$$

$$f_{test} = 300\,N/mm^2$$

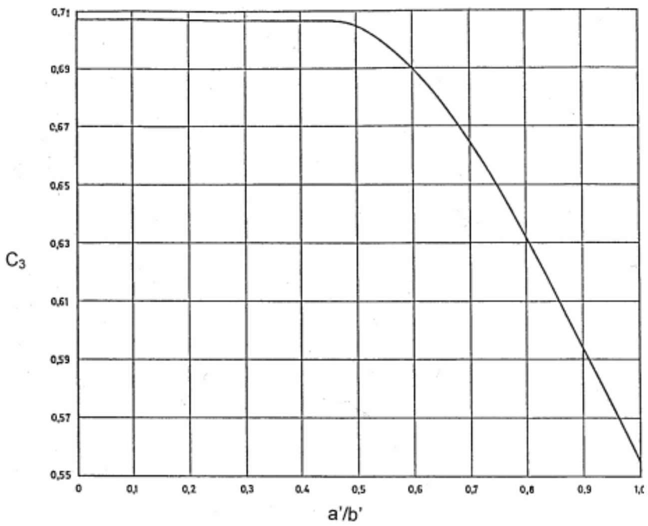
Body Wall Thickness

$$e = C_3 * \alpha' \sqrt{\frac{P}{f}} \tag{1}$$

$$\frac{a'}{b'} = \frac{2000}{6000} \tag{2}$$

$$\frac{a'}{b'} = 0,33$$

Table 2 *Shape factor C3 for welded non-circular flat ends*



$$C_3 = 0,71 (\text{obtained from Table 2})$$

$$e_{test} = 0,71 * 2000\,mm \sqrt{\frac{0,825\,N/mm^2}{300\,N/mm^2}}$$

$$e_{test} = 74,46\,mm$$

$$e_{test} = 0,71 * 2000\,mm \sqrt{\frac{0,55\,N/mm^2}{195,8\,N/mm^2}}$$

$$e_n = 75,25\,mm + 3\,mm (\text{measure for corrosion})$$

$$e_n = 78,25\,mm \cong 80\,mm \text{ was selected}$$

Membrane Stresses

At C point

$$(\sigma m)_C = P * \frac{a+L}{e} \tag{3}$$

$$(\sigma m)_C = 10,31\,N/mm^2$$

$$(\sigma m)_C = (\sigma m)_D = (\sigma m)_B = (\sigma m)_A = 10,31\,N/mm^2$$

At corner

$$(\sigma m)_{B-C} = \frac{P}{e} * \left\{ a + \sqrt{\left(L^2 + \ell_1^2\right)} \right\} \tag{4}$$

$$(\sigma m)_{B-C} = 1595,13\,N/mm^2$$

$$I_1 = I_2 = \frac{e^3}{12} \tag{5}$$

$$I_1 = I_2 = 42666,7\,mm^3$$

Bending Stresses

$$\alpha_3 = \frac{L}{\ell_1} \tag{6}$$

$$\alpha_3 = 1$$

$$Ø = \frac{a}{\ell_1} \tag{7}$$

$$Ø = 0,142$$

$$K_3 = \frac{l^{12}\left(6Ø^2\alpha_3 - 3\pi Ø^2 + 6Ø^2 + a_3^3 + 3a_3^3 - 6Ø - 2 + 1,5\pi a_3^2 Ø + 6Ø a_3\right)}{3\left(2a_3 + \pi Ø + 2\right)} \tag{8}$$

$$K_3 = -16138,5818\,mm^2$$

$$M_A = P * (-K_3) \tag{9}$$

$$M_A = 13315,15\,N$$

At C point

$$(\sigma b)_C = \pm \frac{e}{4I_1} * \left\| 2M_A + P\left(2aL - 2a\ell_1 + L^2\right) \right\| \tag{10}$$

$$(\sigma b)_C = 308,56\,N/mm^2$$

At D point

$$(\sigma b)_D = \pm \frac{e}{4I_1} * \left\| 2M_A + P\left(2aL - 2a\ell_1 + L^2 - \ell_1\right) \right\| \tag{11}$$

$$(\sigma b)_D = 12,48\,N/mm^2$$

At A point

$$(\sigma b)_A = \pm \frac{M_A * e}{2I_1} \tag{12}$$

$$(\sigma b)_A = 12,48\,N/mm^2$$

At B point

$$(\sigma b)_B = \pm \frac{e}{4I_1} * \left[2M_A + PL^2 \right] \tag{13}$$

$$(\sigma b)_B = 308,56\,N/mm^2$$

At corner

$$(\sigma b)_B = \pm \frac{e}{4I_1} * \left[2M_A + P\left\{2a\left(L\cos Ø - \ell_1(1 - \sin Ø)\right) + L^2\right\} \right]$$

$$(\sigma b)_B = 406,41\,N/mm^2$$

Mirror Wall Tickness

$$e = C_3 * \alpha' \sqrt{\frac{P}{f}} \tag{14}$$

$$\frac{a'}{b'} = 1 \tag{15}$$

$$C_3 = 0.56 (\text{obtained from Table 2})$$

$$e_t = 0,56 * 2000\,mm * \sqrt{\frac{0,825\,N/mm^2}{300\,N/mm^2}}$$

$$e_t = 58,73\,mm$$

$$e_n = 0,56 * 2000\,mm * \sqrt{\frac{0,55\,N/mm^2}{195,8\,N/mm^2}}$$

$$e_n = 59,35\,mm + 3\,mm (\text{measure for corrosion})$$

$$e_n = 62,35 \cong 65\,mm \text{ was selected}$$

Door Wall Thickness

$$e = C_3 * \alpha' * \sqrt{\frac{P}{f}} \tag{16}$$

$$\frac{a'}{b'} = 0,5 \tag{17}$$

$$C_3 = 0.83 (\text{obtained from Table 3})$$

$$e_t = 0{,}83 * 800\,mm * \sqrt{\frac{0{,}825\,N/mm^2}{300\,N/mm^2}}$$

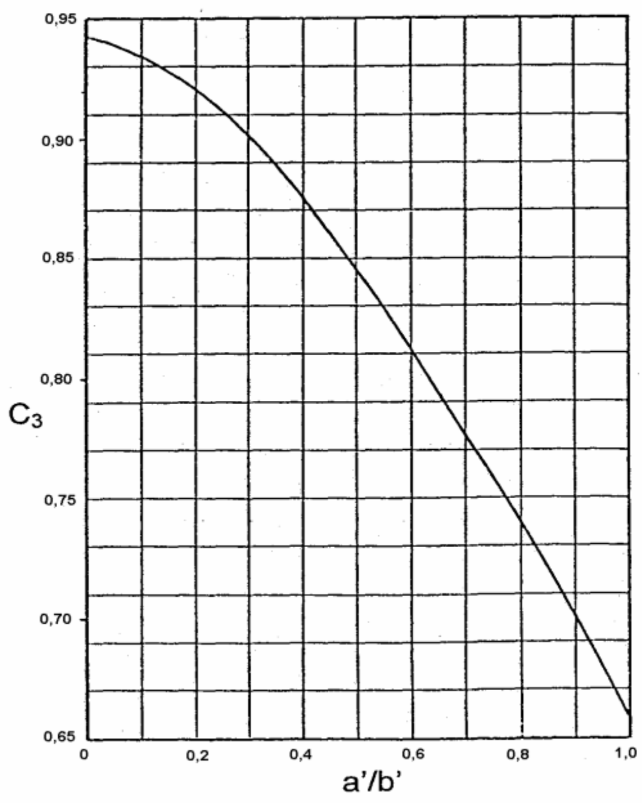
$$e_t = 34{,}82\,mm$$

$$e_n = 0{,}83 * 800\,mm * \sqrt{\frac{0{,}55\,N/mm^2}{195{,}8\,N/mm^2}}$$

$$e_n = 35{,}19\,mm + 3\,mm (\,measure\ for\ corrosion\,)$$

$$e_n = 38{,}19 \cong 40\,mm\ was\ selected$$

Table 3 *Shape factor C3 for bolted rectangular flat end with full-face gasket*



<u>**Necessary Nomenclature**</u>

C_3 are the shape factors for calculation of flat ends of non-circular shape

a' is the smaller width dimension in a rectangular, elliptical or obround end

b' is the greater width dimension in a rectangular, elliptical or obround end. M_A is the bending moment at the middle of the long side, it is positive when the outside of the vessel is put into compression. It is expressed as bending moment per unit length (in N. mm/mm)

σ_b is the bending stress

σ_m is the membrane stress

a is the inside corner radius

I_1, I_2 is the second moment of area per unit width of a strip of thickness e.

K_3 is a factor for unreinforced vessel

L_1, l_x, L, L_y are the dimensions of the vessel

$\emptyset$ is a factor, see equation at EN 13445 (15.5.1.2-15)

α_3 is a factor, see equation at EN 13445 (15.5.1.2-14)

e required thickness

e_n nominal thickness

f nominal design stress

f_d maximum value of the nominal design stress for normal operating load cases

f_{test} maximum value of the nominal design stress for testing load cases

$R_{p_{0,2/t}}$ minimum 0,2 % proof strength at temperature t in °C

$R_{m/t}$ minimum tensile strength at temperature t in °C

3. CONCLUSION

In this study, the body, mirror and door wall thicknesses were calculated for the prismatic pressure vessel designed in accordance with TS 13445-3 standard by using the formulas in the standard in accordance with DBF (design by formula) methodology. Then, the body, mirror and door wall thicknesses calculated with the formulas specified in the relevant standards were modelled with SolidWorks software. During the analysis, the test pressure was entered into the programmed as 8.25 bar, which is 1.5 times the working pressure of 5.5 bar.

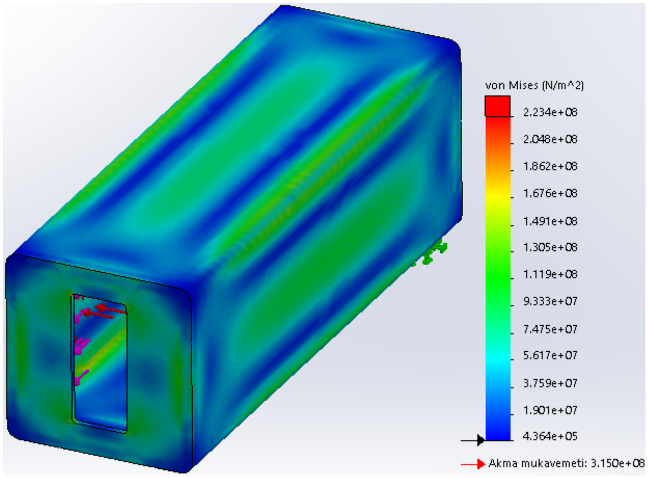


Fig. 3. Vonmises Stress Analysis Result

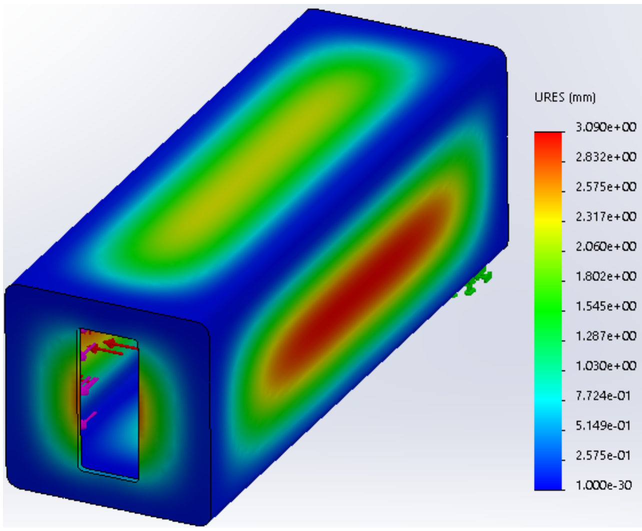


Fig. 4. Displacement Analysis Result

Fig. 3, the maximum Von Mises equivalent stress is 223 MPa, which is approximately 70% of the yield stress of the cabin material. Thus, it can be said that the cabin is 2 times safer cumulatively. This result is an acceptable value in terms of stress distribution in practical applications. However, as a result of the displacement analysis, the maximum displacement value was determined as 3,090 mm according to the programmed. Although this amount of displacement is within acceptable limits in terms of structural engineering, the situation should be evaluated in terms of long-term performance and functionality.

It is possible to minimize this displacement and reduce the equivalent stress by designing a reinforced cabin. A reinforced design of the cabin has the potential to reduce costs by optimizing the use of materials, while at the same time increasing structural safety. The use of reinforcing elements improves the load distribution in high stress regions, increasing the overall durability and minimizing possible deformations. Thus, a balanced relationship between cost effectiveness and reliability of the design will be achieved.

In conclusion, the analysis for the unreinforced cabin show that the design meets the general safety standards. However, in the light of the Von Mises equivalent stress value corresponding to 70% of the yield strength of the material and the amount of displacement observed, it is recommended to optimize the design and integrate reinforced structures.

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