# Stress Analyzes of a Didactic-Purpose Pressure Vessel

**Sofia Glyniadakis** [1], **Marcelo Sampaio Martins** [1], **José Elias Tomazini**[1]

[1] São Paulo State University (UNESP), School of Engineering, Guaratinguetá, Av. Ariberto Pereira da Cunha, 333 – Vila Paraíba – Guaratinguetá – SP – Brazil.

*Abstract: The present work describes a didactic device, a thin-walled pressure vessel developed for the demonstration of the two-dimensional state of stress. For this purpose, the equipment has been instrumented with strain gauges at several points. The strain/stress measured by the gauges were compared to values from classical mechanical engineering theories; they were also compared with analyzes suggested by the American Society of Mechanical Engineers (ASME) and the Finite Element Analysis (FEA). Furthermore, strain gauges near the nozzles, as well as other welded components were assessed for the investigation of the effects of any bending occurrences and compressive stress incidence. Divergences between both assessments were below 6%, which demonstrates the consistence of classical mechanics theoretical concepts. It was also possible to emphasize the importance of safety regarding project development as, in critical points, experimental and numerical analyses indicated higher stress values than the ones described in classical theories.*

*Keywords: Pressure Vessel, Strain Gauges, Didactic Device, Two-Dimensional Stress.*

## INTRODUCTION

According to Borse and Sharma (2012); and Yee and Kapper (2006), pressure vessels are fluid storage and transportation devices, usually made of steel, and widely used in risky industrial operations. They stand out for their design, which is aimed at operational safety.

Carbonari *et al.* (2011) reported different literature approaches to pressure vessels. The nozzles' and heads' location optimization have been independently considered. This can generate problems such as nozzle-head thickness variation, among others. In view of this, the pressure vessels' structural integrity and, consequently, their effective comprehension is of paramount importance. As reported by Neto and Paulo (2000); Lozev *et al.* (2005), ASME's section III [1] and section VIII [2] present the pressure vessel calculation parameters according to their application particularities.

To Dias (2017), the phenomena observed at the laboratory of Mechanics of Materials enabled students to better learn and understand how the vessel works. In view of this improvement, the Laboratory of Structures, part of the Department of Mechanics (DME), UNESP - Campus of Guaratinguetá (Brazil), has been innovating over the past few years and the present work makes a groundbreaking contribution by describing the whole design of a didactic device: the thin-wall pressure vessel.

The didactic approach consists of three main stages. The first is the theoretical analysis, which considers the theory of thin-wall pressure vessels and allows achieving strain/stress at specific points with the aid of material mechanics concepts. The next one is to collect actual strain data obtained by the strain gauges, and to correlate tensile stresses through the mechanical properties of the applied manufacture material. Finally, the ANSYS software (Canonsburg, Pennsylvania, USA) is used in the Finite Element Analysis (FEA), usually applied to Structural Engineering. The success of all three stages ensures device validation. In view of this, this study proposed to analyze actions for project reliability validation.

"Interdisciplinarity is characterized by the intensity of the exchanges between the specialists and the degree of real integration of the disciplines and, therefore, it is necessary to work together among the professors to accomplish this task" (SILVA *et al.*2013). Aiming at bringing about such integration, even though the main objective of the device is to analyze stresses and strains in the structure of the pressure vessel, this allows professors to present basic concepts of fluid mechanic systems, industrial instrumentation and acquisition of electrical measurements.

# Stress Analyzes of a Didactic-Purpose Pressure Vessel

## DEVICE ASSEMBLING

### The concept of a pressure vessel

Initially, a small-sized pressure vessel was designed so as to be used in an educational laboratory. In addition to its dimensions, a very important issue was the operating pressure, as it should also not be of such magnitude as to pose any danger to students or staff.

Regarding its assembly, a commercial steel tube SA 516 Gr. 60 (E = 210 GPa, ν = 0.32) was used as the vessel's cylindrical part, whose head was welded to each end. Along the pressure vessel, all main components of a commercial pressure vessel were used (manometer, safety valve, breath, drain, and fluid input nozzles) to faithfully represent the influences of such components under inherent stress. Its positioning and geometric characteristics are shown in Fig. 1. Figure 2 shows a device photo with all of its components.
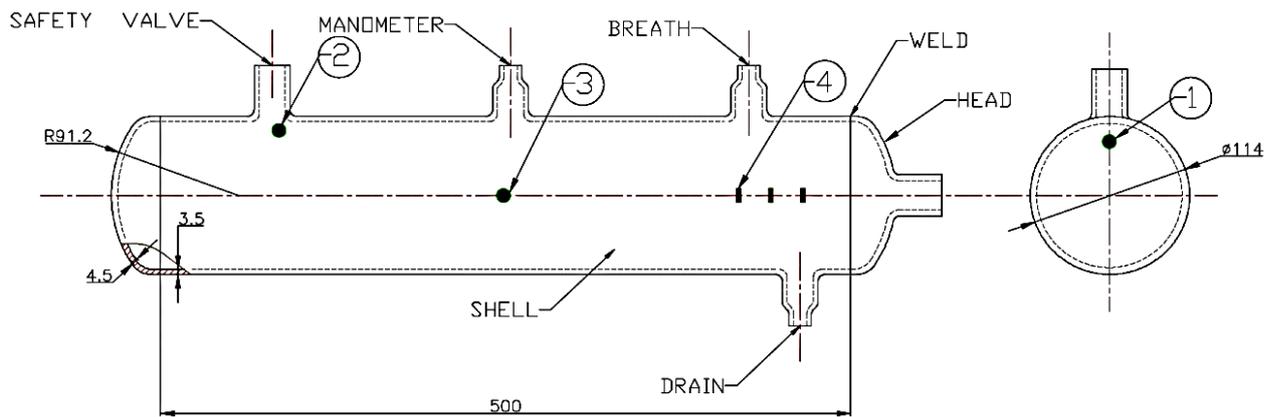


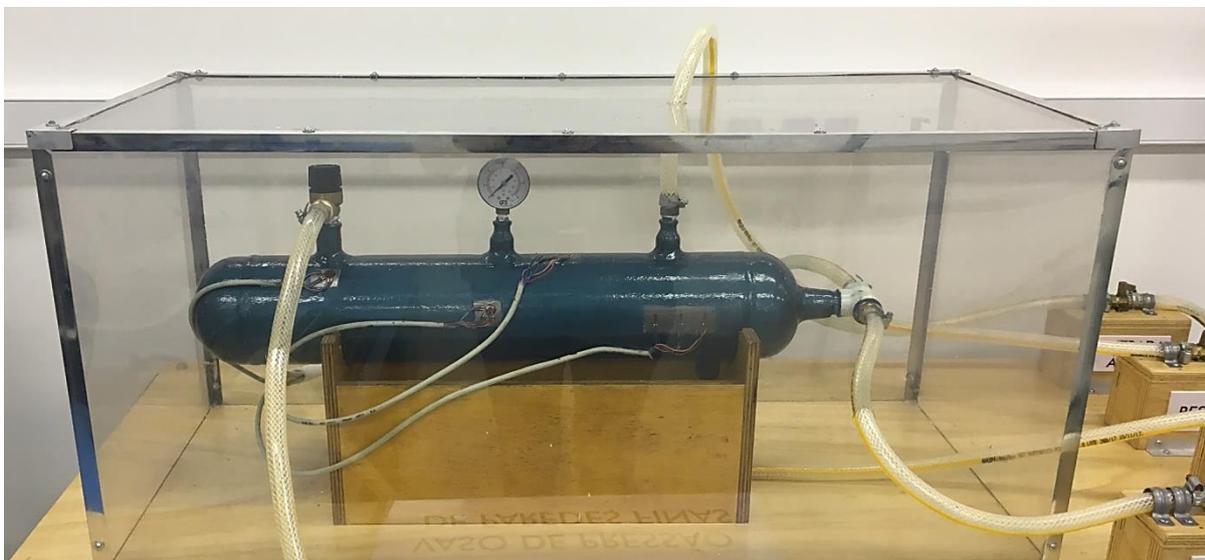**Figure 1 – Pressure vessel: components and strain gauges in position 1, 2, 3, and 4.**



**Figure 2 –The pressure vessel.**

Strain gauges were used to analyze stress on the pressure vessel, and the locations they were attached to are shown in Fig. 1. In these areas, represented by numbers 1, 2, and 3, rectangular rosettes were attached for main stress obtention, as showed in Fig. 1 and 2. Number 4 in Fig. 1 represents three unidirectional strain gauges, which were attached for elastic hoop stress analysis in a region near the head weld area and corresponding stress.

*Mathematical Modeling*

As the ratio between the thickness (t) and the inner radius (r) was 6.14% (less than 10%), the device was characterized as a thin-walled pressure vessel (AL-GAHTANI, 2019a). Thus, normal stress in the spherical head is given by Eq. (1) and normal stress in the cylindrical body by Eq. (1) and (2), longitudinal and hoop, respectively (GOLUBOVIĆ *et al.*, 2018) (SINCLAIR, HELMS; 2015) .

$$\sigma_l = \frac{pr}{2t} \tag{1}$$

$$\sigma_t = \frac{pr}{t} \tag{2}$$

In contrast, ASME (2015) somewhat incorporates a correction factor based on experimental data for the classical equations in literature as a pressure function, which are presented in Eq. (3), (4), and (5). Equation (3) makes it possible to obtain head stress, whereas (4) and (5) represent shell, hoop (4) and longitudinal (5) stress, respectively.

$$\sigma = \frac{pr}{2t} + 0.2\,p \tag{3}$$

$$\sigma_t = \frac{pr}{t} + 0.6\,p \tag{4}$$

$$\sigma_l = \frac{pr}{2t} + 0.6\,p \tag{5}$$

Equations (6) and (7) can be used to obtain the principal experimental stress of the head and of the cylindrical shell, as developed by Morita *et al.* (2019). In these expressions $\varepsilon_t$, $\varepsilon_l$, and $\varepsilon_{45°}$ are the hoop, longitudinal and 45° stress measured by the rosette strain gauges.

$$\sigma_1 = E\left[\frac{\varepsilon_t + \varepsilon_l}{2(1 - v)} + \frac{1}{2(1 + v)}\sqrt{(\varepsilon_t + \varepsilon_l)^2 + (2\,\varepsilon_{45°} - \varepsilon_t - \varepsilon_l)^2}\right] \tag{6}$$

$$\sigma_2 = E\left[\frac{\varepsilon_t + \varepsilon_l}{2(1 - v)} - \frac{1}{2(1 + v)}\sqrt{(\varepsilon_t + \varepsilon_l)^2 + (2\,\varepsilon_{45°} - \varepsilon_t - \varepsilon_l)^2}\right] \tag{7}$$

To analyze the rosette represented by number 2 in Fig.1, the mathematical modeling described by Eq. (6) and (7) may also be applied to obtain the experimental results, but these cannot be compared to analytical models due to nozzle interference, but a comparison is possible if made through FEA.

The experimental data collected by the unidirectional strain gauges positioned in region 4, as shown in Fig. 1, only allow an analysis of strains, which can be compared with the elastic hoop stress through Eq. (8) (SINCLAIR; HELMS, 2015)

$$\varepsilon_1 = \frac{pr}{Et}\left(1 - \frac{v}{2}\right) \tag{8}$$

For the numerical analysis, the commercial software ANSYS (student version) based on the Finite Element Analysis (FEA) was used. The numerical simulation was performed using Tetrahedral elements, 23,818 nodes, 11,620 elements, and 8-mm maximum element size.

## Experimental setup

A hydrostatic pressure test was performed with the device, as described by ASME (2015) with pressures ranging from 0.0 to 0.54 MPa. The strain gauges were connected to a QuantumX MX1615 HBM™ signal conditioning equipment through a ¼ Wheatstone bridge assembly and connected to a computer through which the Catman AP (HB™) software generated a spreadsheet with all collected pressure function strains.

# Stress Analyzes of a Didactic-Purpose Pressure Vessel

Initially, the pressure vessel was filled with water and, to reach final pressure, it was pressurized with air through a manual pneumatic pump. A manual air compressor not only decreased the device's final cost, but also ensured slower and safer pressurization in operational terms.

The strain data sample time was 2 Hz, since pressurization was a quasi-static process. Although pressure varied from 0.0 to 0.54 MPa, the strains were computed at 0.40, 0.45, 0.48, and 0.54 MPa. When each of these pressure values were reached, the pressurizing process was paused, and values were collected after 10 seconds, each time. For calculation purposes, the strain value corresponding to each pressure value was given by the average of the strain values collected in each interval.

## RESULTS AND DISCUSSIONS

The FEM simulations are shown in Fig. 3 and 4. Figure 3 shows the maximum principal stress values in the entire pressure vessel. Figure 4 shows the maximum principal stress values found only in areas where strain gauges are attached to the cylinder. These values will be compared with others obtained from analytical and experimental results. In both ways, cylindrical coordinates were used for the simulation, and the pressure vessel design required a global orthogonal coordinate.

Regarding the experimental results, the principal stresses ($\sigma_1$ and $\sigma_2$) were obtained using Eq. (6) and (7) for rosettes located in regions 1, 2 and 3, shown in Fig. 1. As for the unidirectional strain gauges located in region 4 represented in Fig. 1, it was not possible to associate the experimental strain values with stress values because there is a two-dimensional state of stress in these regions. However, it was possible to compare these experimental results with the strains obtained through a numerical analysis, whose results are expressed graphically by Fig. 5, 6, 7, and 8.
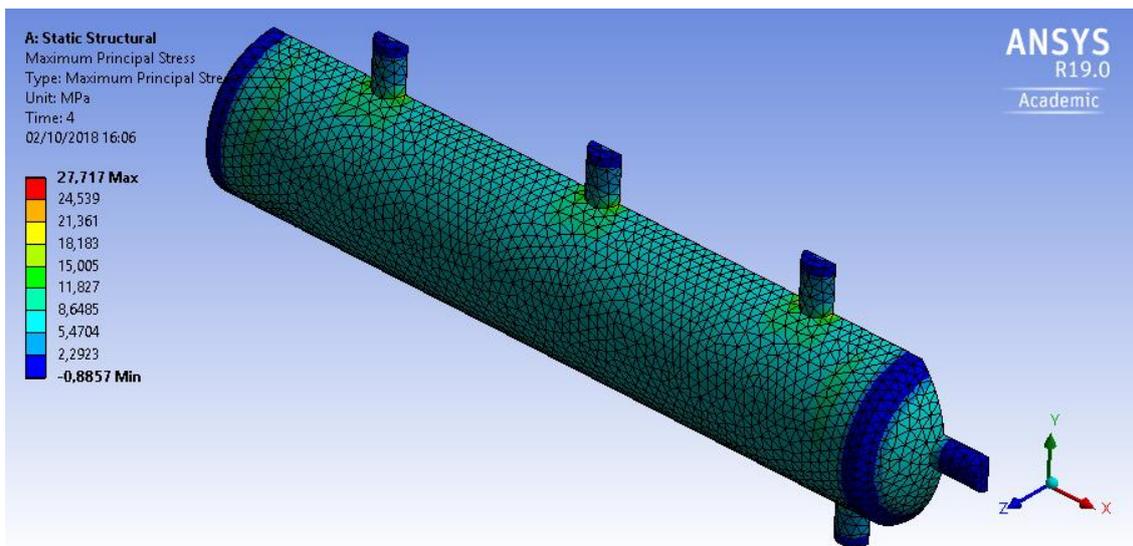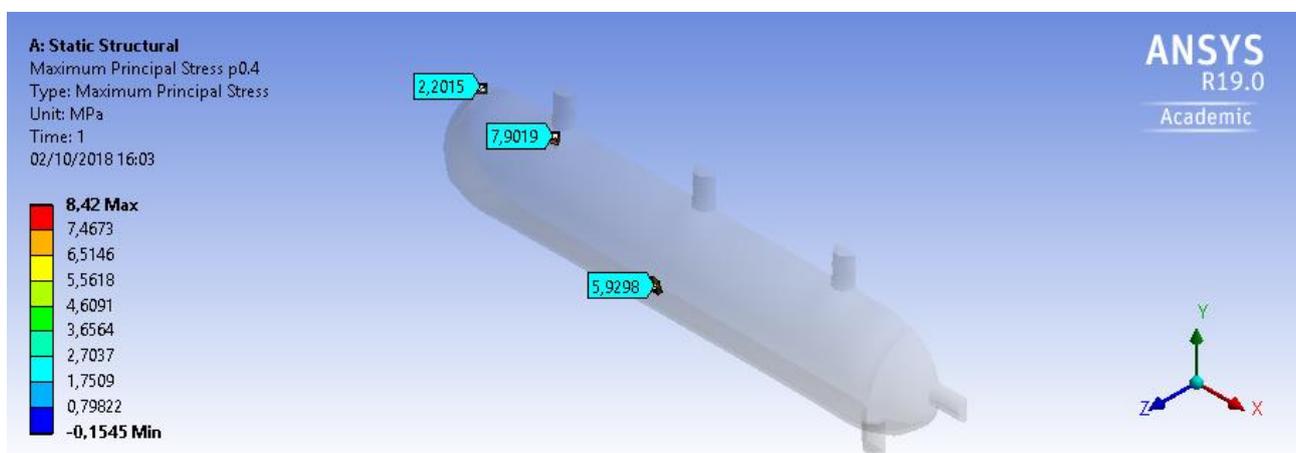


**Figure 3 – Global simulation.**
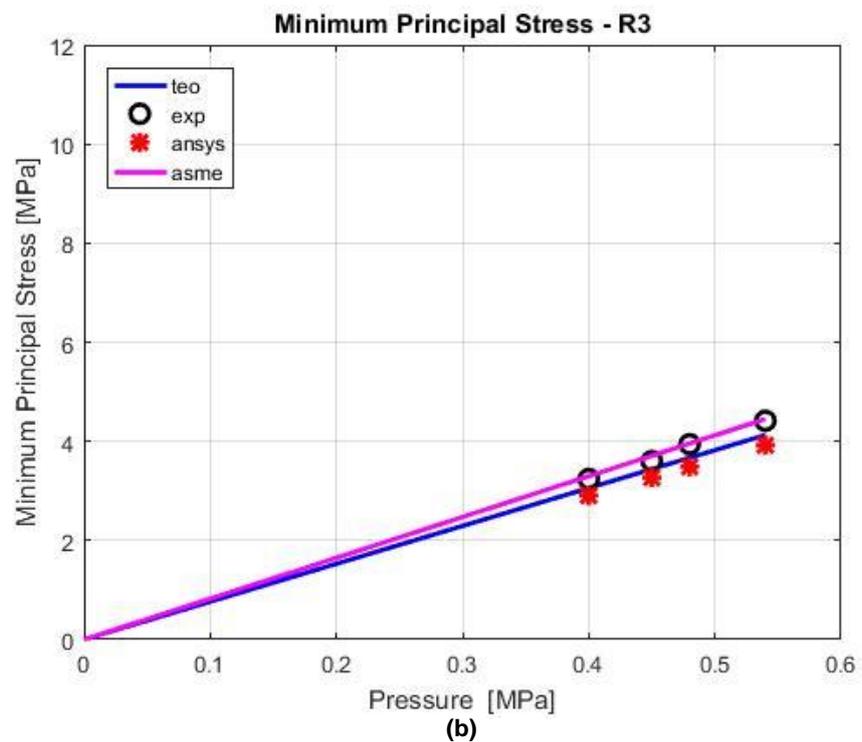


**Figure 4 – Rosettes' principal numerical stress values.**
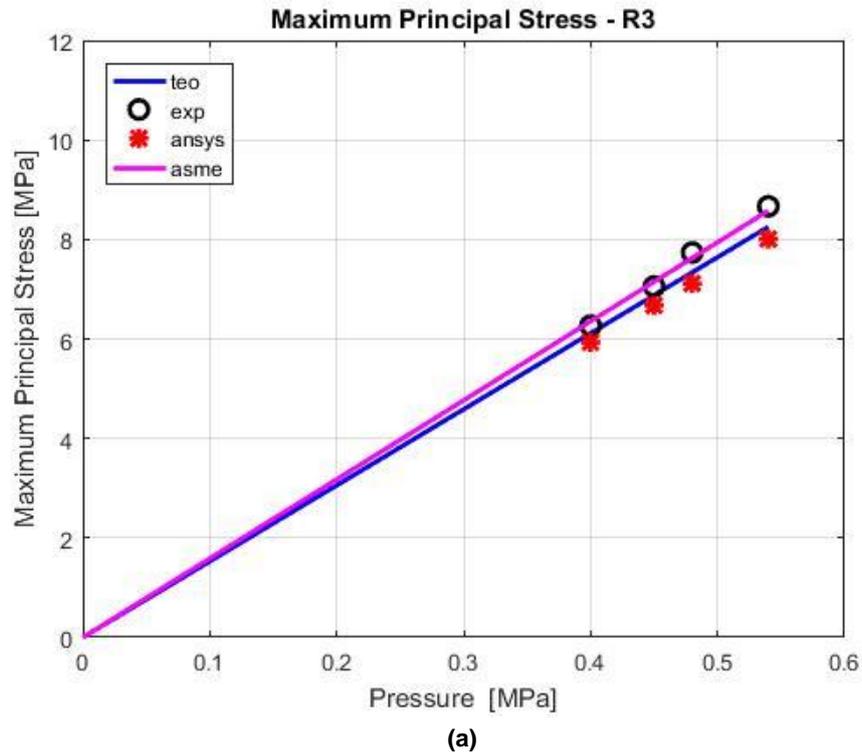
**(a)**



**(b)**

**Figure 5 – Rosette 3 (a) maximum; (b) minimum stress.**

According to the experimental results with rosette 3 area, shown in Fig. 5, obtained data were similar through the analytical and the numerical methods. By considering Eq. (4) and (5) used by ASME, better results were achieved with a 1.4%-maximum error. It should be emphasized that the region where this rosette was attached to is far from bend stress points, such as nozzles, welds, or junctions of different geometries.

# Stress Analyzes of a Didactic-Purpose Pressure Vessel
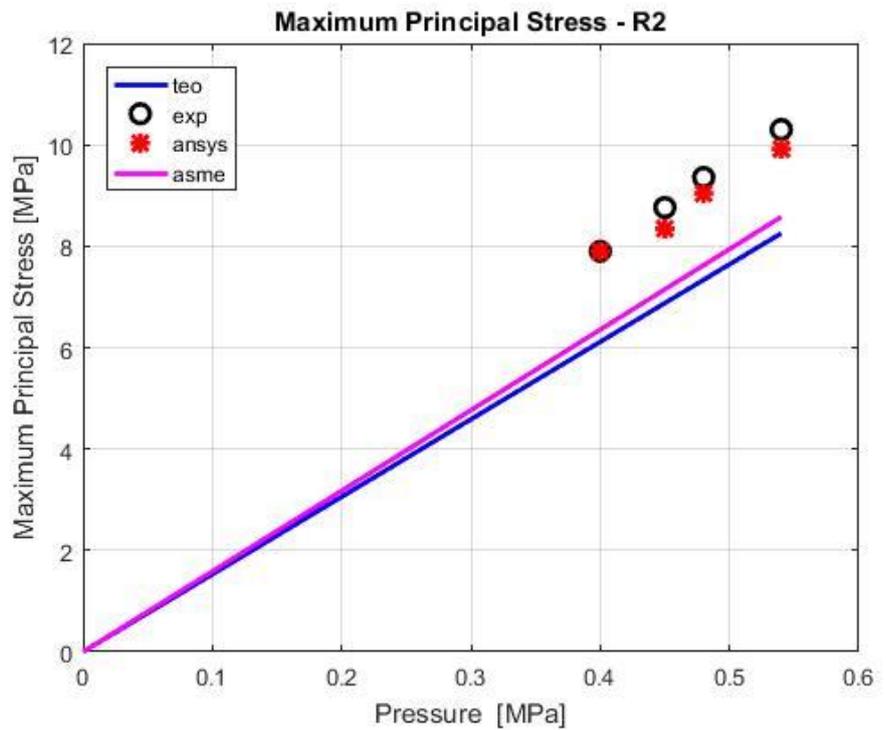


**(a)**
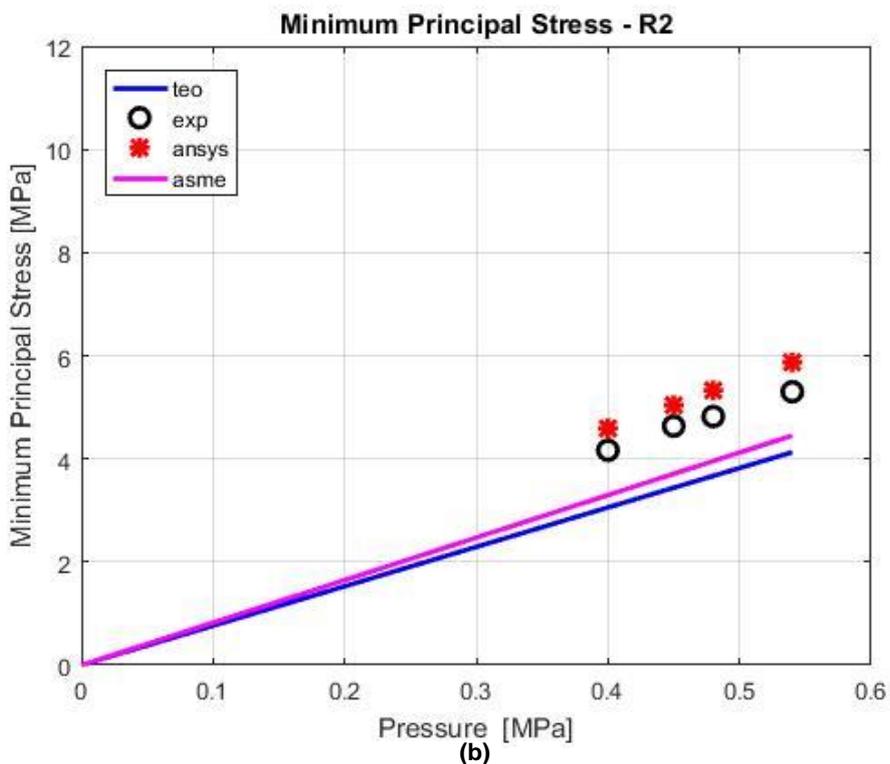


**(b)**

**Figure 6 – Rosette 2 (a) maximum; (b) minimum stress.**

Values of experimental principal stresses in area 2, shown in Fig. 6, differ from those calculated by the analytical methods, but are close to those numerically obtained. This divergence from the analytical methods is due to nozzle and weld influences, which lead to secondary stress. The graph in Fig. 4 shows that the differences between experimental and analytical stress values were nearly constant.
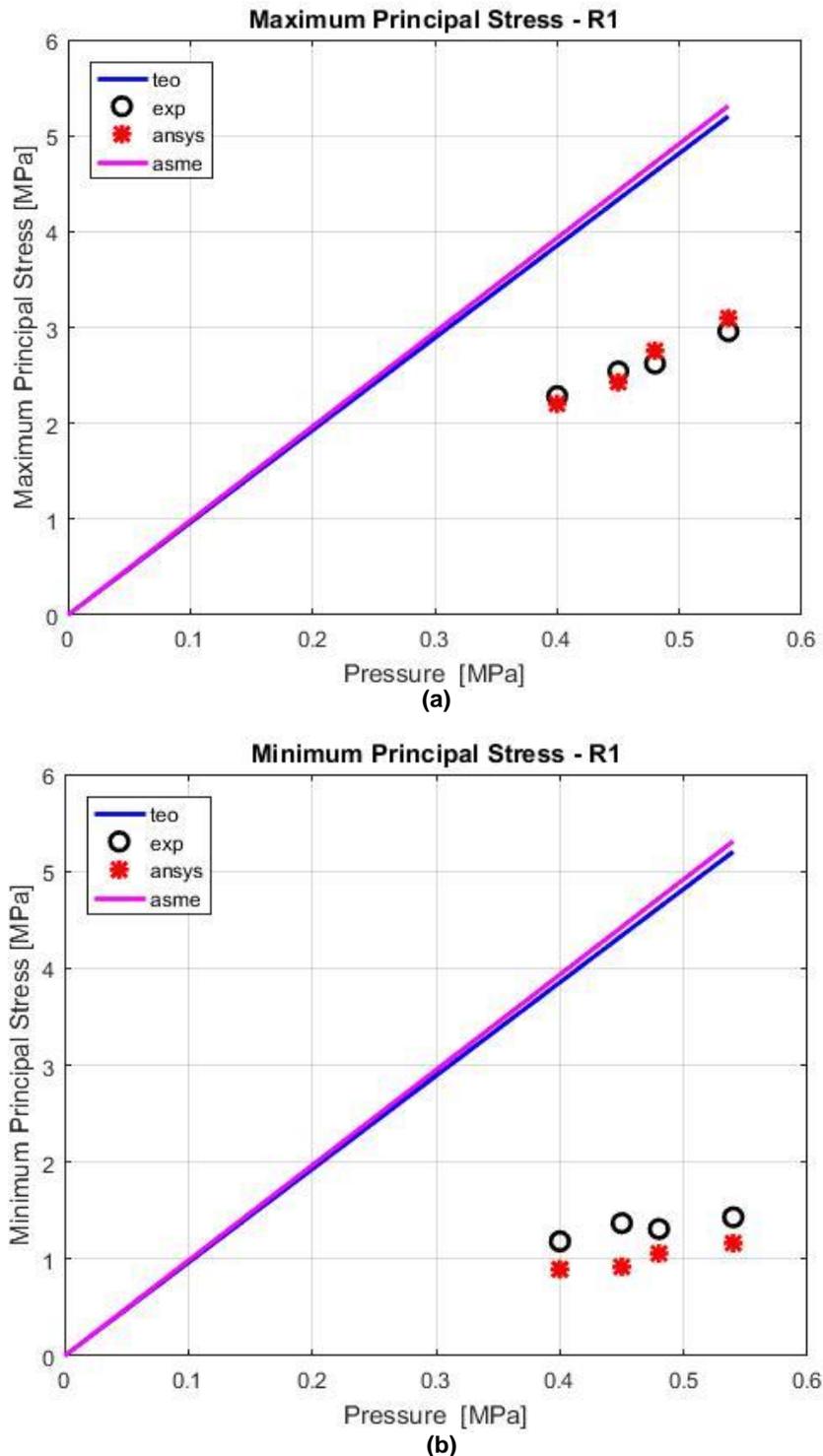
**Figure 7 – Rosette 3 (a) maximum; (b) minimum stress.**

The experimental principal stress in region 1 shown in Fig. 7 presented different values from those predicted by the analytical and numerical methods. This is possibly due to the rosette being close to the head-shell junction. According to Al-Gahtani (2019b), there were moments in which some bending occurred in the transition region of the two geometries, thus producing flexural stress. According to the classical theory, maximum and minimum principal stresses should be equal. However, different values were observed in this study. Therefore, it is possible to conclude with such instrumented region that the proximity of the attachment area between the head and the shell generates secondary stress, thus diverging from the theoretically predicted results in a way that those concentration factors are disregarded by Eq. (1) and (3). Regarding FEM, the proximity between this value and the experimental stress values can be verified below.

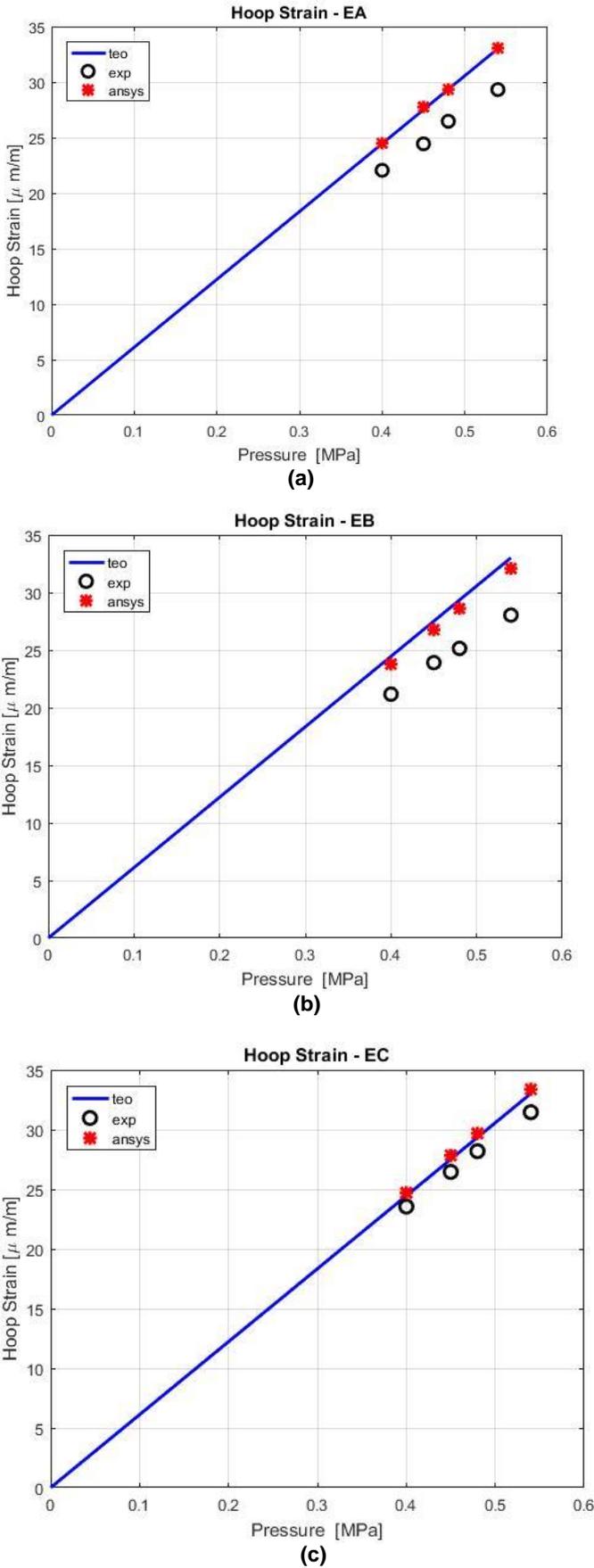# Stress Analyzes of a Didactic-Purpose Pressure Vessel



Figure 8 – Strains in region 4 (a) strain A; (b) strain B; (c) strain C.

The elastic hoop stress values measured by the unidirectional strain gauges located in region 4, shown in Fig. 8, were smaller than those analytically found. This may be due to the welded area being close to the device's instrumented region.

Considering that the range of strain results are in micrometer tenths, the difference among experimental, classical and FEM approaches is at expected levels. According to Walendziuk *et al.* (2016), a 4.85%-variation is considered normal, even if in micrometer hundredths. This device allows reporting material mechanics concepts validation to undergraduates. However, it is still possible to introduce ASME (2015) standards to students, as to regulate pressure vessels designing. As expected, ASME's (2015) stress calculation was closer to experimental results for its empirical basis. In addition to the mathematical modeling itself, it was possible to verify a convergence between standard-calculated experimental and theoretical results, hence demonstrating the importance of empirical correction factors.

## CONCLUSIONS

The experiment carried out with the pressure vessel allowed showing undergraduates several concepts, such as the theory of stress in thin-wall pressure vessels in the shell and head, the use of strain gauges and rosettes for the experimental determination of stresses, as well as the use of commercial software for stress analysis. It was verified in the geometry regions transition areas, and welding and nozzles areas, that stresses diverge from the analytical model and in these cases only an experimental and/or numerical analysis enable stress determination.

In addition to the aforementioned concepts, the importance of ASME's pressure vessels design standards to determine stress is of paramount relevance, as it as they were proved to be closer to previously reported experimental results.

## REREFENCES

AL-GAHTANI, H. J. Design-Focused Stress Analysis of Cylindrical Pressure Vessels Intersected by Small-Diameter Nozzles., v. 139, n. April 2017, p. 1–11, 2019a.

AL-GAHTANI, H. J. Simplified Formulation of Stress Concentration Factors for Spherical Pressure Vessel – Cylindrical Nozzle Juncture., v. 138, n. June 2016, p. 1–10, 2019b.

American Society of Mechanical Engeneers. **Boiler and pressure vessel code** Lynn: ASME 2015.

BORSE, Y.; SHARMA, A. K. Design analysis of pressure vessels at high stress zones using Pro/E v 4.0. **International Journal of Modern Engineering Research** , v. 2, n. 3, p. 991–995, 2012.

CARBONARI, R. C.; MUÑOZ-ROJAS, P. A.; ANDRADE, E. Q.; et al. Design of pressure vessels using shape optimization: An integrated approach. **International Journal of Pressure Vessels and Piping**, v. 88, n. 5–7, p. 198–212, 2011. Elsevier Ltd.

DIAS, R. A. A. **Dispositivo didático para aferição de deflexão horizontal em vigas curvas isostáticas**. 2017. 82f. Trabalho de Conclusão de Curso (Graduação em Engenharia Mecânica) - Faculdade de Engenharia do Campus de Guaratinguetá, Universidade Estadual Paulista, Guaratinguetá, 2017

GOLUBOVIĆ, T.; SEDMAK, A.; BRKIĆ, V. S.; KIRIN, S.; RAKONJAC, I. Novel Risk Based Assessment of Pressure Vessels Integrity. **ehnicki Vjesnik - Technical Gazette**, v. 3651, n. M, p. 803–807, 2018.

LOZEV, M. G.; SMITH, R. W.; GRIMMETT, B. B. Evaluation of Methods for Detecting and Monitoring of Corrosion Damage in Risers. **Journal of Pressure Vessel Technology**, v. 127, n. 3, p. 244, 2005.

MORITA, N.; HARADA, T.; NOGUCHI, H. Simple Method of Using Strain Gauges to Estimate Stress Fields Near a Notch in Structures Subject to Large Deflections Small and Large Deflections. **Journal of Testing and Evaluation**, v. 45, p. 1618–1629, 2019.

MATTAR NETO, M..; **Projeto por análise de vasos de pressão:** exemplos de utilização do ANSYS. São Paulo IPEN. 2000

SILVA, D. L. F.; SILVA, B. C.; AMORIM, G. F. Desenvolvimento de um módulo didático para o ensino multidisciplinar em engenharia de controle e automação. In: COBENGE - CONGRESSO BRASILEIRO DE EDUCAÇÃO EM ENGENHARIA. Gramado, 2013. **Anais...** Gramado, 2013.

SINCLAIR, G. B.; HELMS, J. E. A review of simple formulae for elastic hoop stresses in cylindrical and spherical pressure vessels : What can be used when. **International Journal of Pressure Vessels and Piping**, v. 128, p. 1–7, 2015. Elsevier Ltd.

WALENDZIUK, W.; GOLEBIOWSKI, J.; IDZKOWSKI, A. Comparative Evaluation of the Two Current Source Supplied Strain Gauge Bridge. **ELEKTRONIKA IR ELEKTROTECHNIKA**, v. 22, p. 33–38, 2016.

YEE, R. K.; KAPPER, M. A structural integrity assessment methodology for pressurized vessels. **Journal of Pressure Vessel Technology**, New York, v. 128, n. 4, p. 541, 2006.

# Stress Analyzes of a Didactic-Purpose Pressure Vessel

**RESPONSIBILITY NOTICE**

The authors are the sole responsible parties for the material included in this paper.