ANALISYS OF STRESS CAUSED BY THE INFLUENCE OF CIRCULARITY ERROR IN PRESSURE VESSEL

Ivair Carvalho de Souza

ivaircs@yahoo.br

André Fioravante

andre@tpaengenharia.com.br

José Rubens Gonçalves Carneiro

joserub@pucminas.br

João Pedro Santiago Carneiro

jsantiagocarneiro@gmail.com

Pontifícia Universidade Católica de Minas Gerais

Avenida Dom José Gaspar, 500, Coração Eucarístico, Belo Horizonte, Minas Gerais, Brazil, 30535-901

Abstract. *The quality of pressure vessels' manufacturing process in terms of their circularity error is a theme of discussion due to there is a difference between acceptance criteria of two most utilized standards in Brazil, that regulates manufacturing of this equipment. The stresses caused by circularity error in a pressure vessel were simulated using the finit element method, the software were COSMOSWorks and results obtained were compared to the acceptance criteria of each standard. The objective of this work is to evaluate the impact of pressure vessel's circularity error and compare the results to N-268 e ASME VIII D.1 standards.*

Keywords: *Circularity errors; Finite Element Methods; Pressure Vessels*

INTRODUCTION

Misinterpretation of Standards criteria contributes significantly in order to increase the number of reworks, therefore this tends to make the production costs higher. Guarantee the execution of processes acording to Standards reduce unnecessary expenses with reworks.

Regulatory Norm (NR13) (1978) defines pressure vessel as being equipments which contains fluids under intern and extern pressure. These vessels are used in food and petrochemichal industry.

Nowadays, pressure vessels manufactured to atend oil and natural gas unit of production are regulated by Standards PETROBRAS N268 (2012) and ASME (*The American Society of Mechanical Engineers*) VIII (2010).

The difference between the criteria of circularity error adopted by those 2 standards above is a theme of discussions during the manufacturing processes of pressure vessels and achieve the desired performance. Both Standards do not consider thickness of transversal section is another factor to create lots of questions related to this doubts.

Castro and Pierce's model, mentioned by Abrantes and Batalha (2003) suggests that material's thickness had a direct impact on stress as indicated in eq. 1.

$$\frac{\sigma_x}{\rho_x} + \frac{\sigma_y}{\rho_y} = \frac{P}{t} \tag{1}$$

Where, σ is the stress in the direction indicated, ρ is radii in the direction indicated, P is the pressure and t is the thickness of material.

Circularity error is the difference between the maximun and minimun value of intern diameters (circular section) as illustrated in fig.1

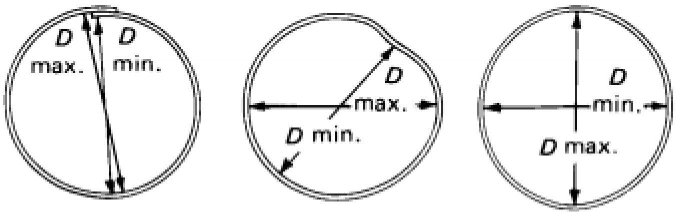

Figure 1: Illustration of circularity error (adapted from ASME VIII standards)

The admission criterion especified in norm ASME VIII for circularity error is 1% of nominal intern diameter of the vessel, which could be increased thickness value of hull plate when there is longitudinal weld. When the transversal section measured coincide with opening axis to allocate nozzles or manholes, this tolerance can be increased more two percent (2%) of opening diameter.

The Standard PETROBRÁS N-268 uses a criterion cited above and another one that creates a limit the circularity error in the regions indicated in itens 19 and 20 in fig.2.

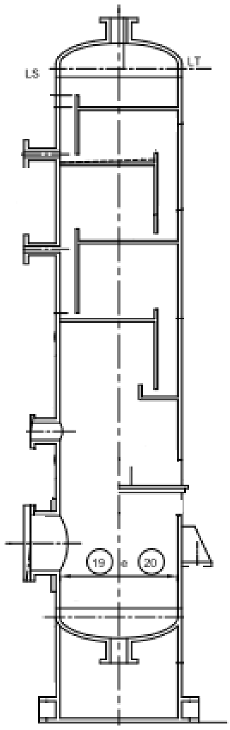

Figure 2: Admission criterion (adapted from PETROBRÁS N-268G standards (2012))

The circularity error is limited in 20 milimeters when intern diameter of vessel is less or equal to 4 meters, 30 milimeters when intern diameter of vessel is more than 4 meters and less than 6 meters and 35 milimeters when intern diameter of vessel is more than 6 meters.

METHODOLOGY

The vessel's parameters are specified in order to be considered in simulation.

Materials specifications

- Hull ASTM A516 Gr70 (Young's modulus 200GPa, Poisson coefficient:0,26, density: 7850 kgf/m^3
- Lid ASTM A516 Gr70 (Young's modulus 200GPa, Poisson coefficient:0,26, density: 7850 kgf/m^3
- The other parts of vessel wil not be considered.

Project data

- Project's pressure: 10,5 kgf/cm^2
- Project's temperature: 60 ^{o}C
- Nominal intern diameter: 3318 mm

- Nominal thickness of hull: 16.0 mm
- Nominal thickness of lids: 19.0 mm
- Minimum thickness of lids after forming: 13.5 mm
- Lengths between tangents (straigth part): 6510 mm
- Ellyptical lids: ASME 2:1
- Efficiency: lids: 0.85/ hull: 1.0
- Maximum allowable pressure: 10.59 kgf/cm^2

The Finit Element Analisys was adopted, using the software COSMOSworks in order to simulate the stresses caused by roundness error in shell (lids and hull) in a pressure vessel in two different conditons.

In the first condition, the stresses in a pressure vessel without circularity error were simulated (ideal condition) on the maximum internal pressure effect allowed in the project (severe working condition). Then, in the second condition, the stresses in a pressure vessel with a roundness error two times greater than the maximum allowed to ASME VIII division 1 code (which represents 66,36 mm of error in this case) with the same pressure in the first situation.

The two-dimensional method of analysis kind of "Planar Symmetry in Shell with Surface Mesh". This model allows to proceed with an analysis using just a quarter of model, as shown in fig. 3 due to its simmetry, improving results accuracy and reducing the computational time.

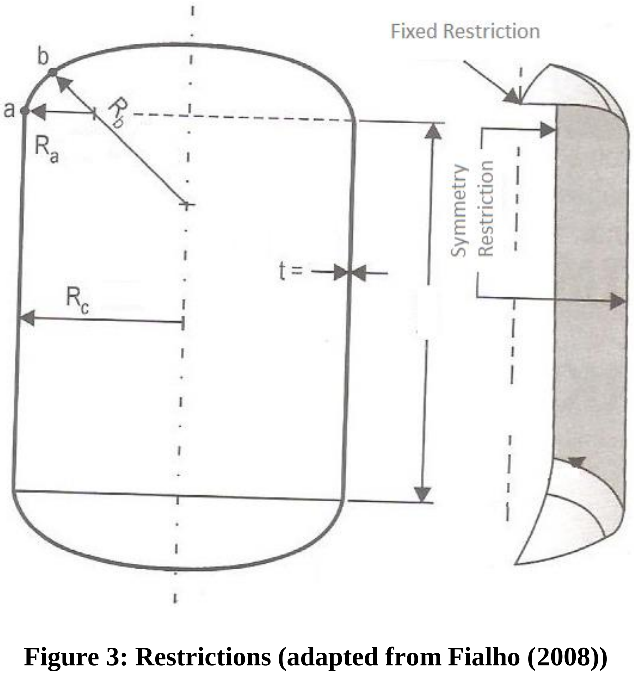

Figure 3: Restrictions (adapted from Fialho (2008))

Fixed restriction on the central axis of the vessel was used because the gauge pressure exerts the same force normal to the inner surface of the casing from the center (Telles, 1991) and symmetry constraint in ¼ edges of the vessel sidewall, as illustrated in Fig. 4 limiting the displacement to the longitudinal and radial direction. The mesh used has 6328 nodes and 3099 elements.

Simulation data was obtained and results were compared to the maximum allowed stress in table of ASME VII code. A graphic of stress in a function of roundness error was plotted, and from that graphic, an experimental equation was obtained. With this experimental equation, estimated the maximum deformation value to the vessel, which not exceeds the maximum permissible stress for the material.

RESULTS

Displacement results (elastic strain) in a vessel without circularity error is illustrated in fig. 4.

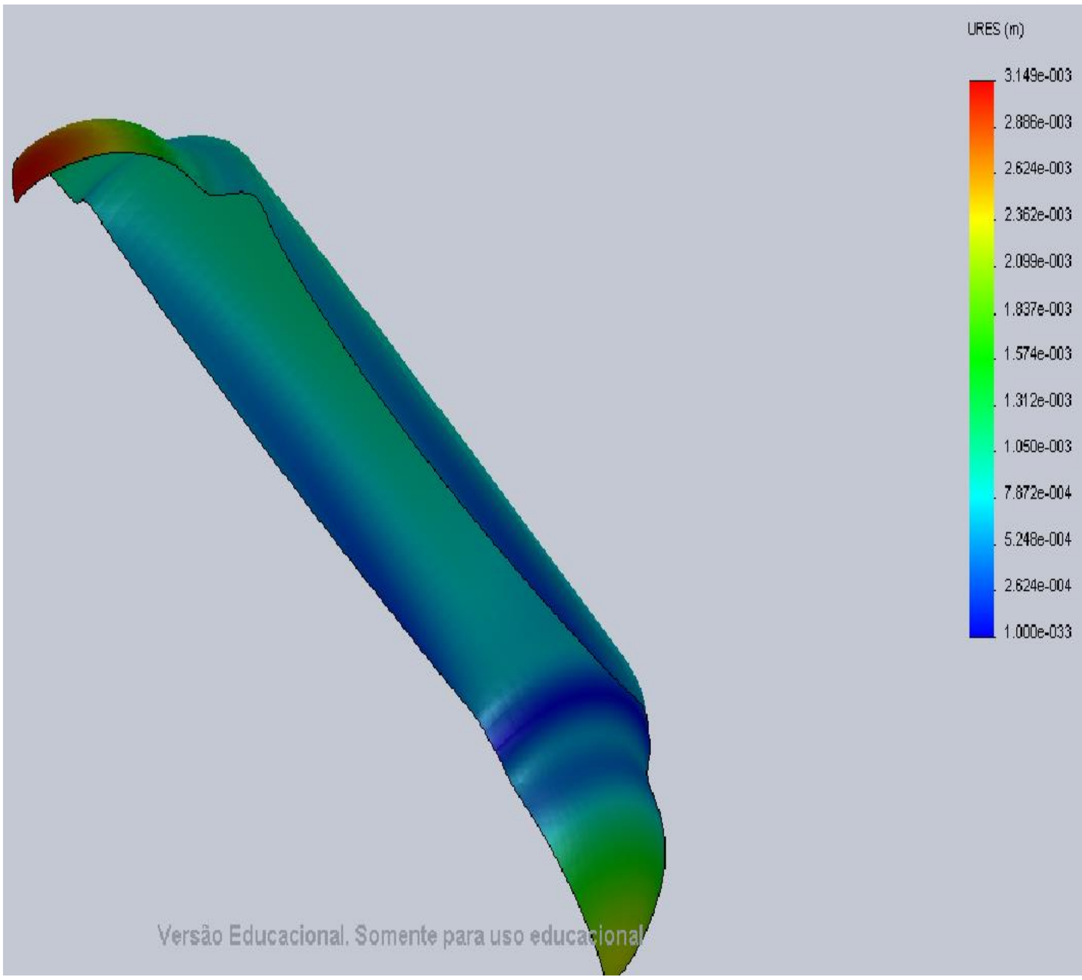

Figure 3: Strain in vessel without circularity error

The maximum displacement in this situation was 3,149 mm in longitudinal direction and close to 1.00 mm in transversal direction.

Displacement results (elastic strain) in a vessel with circularity error is illustrated in fig. 5.

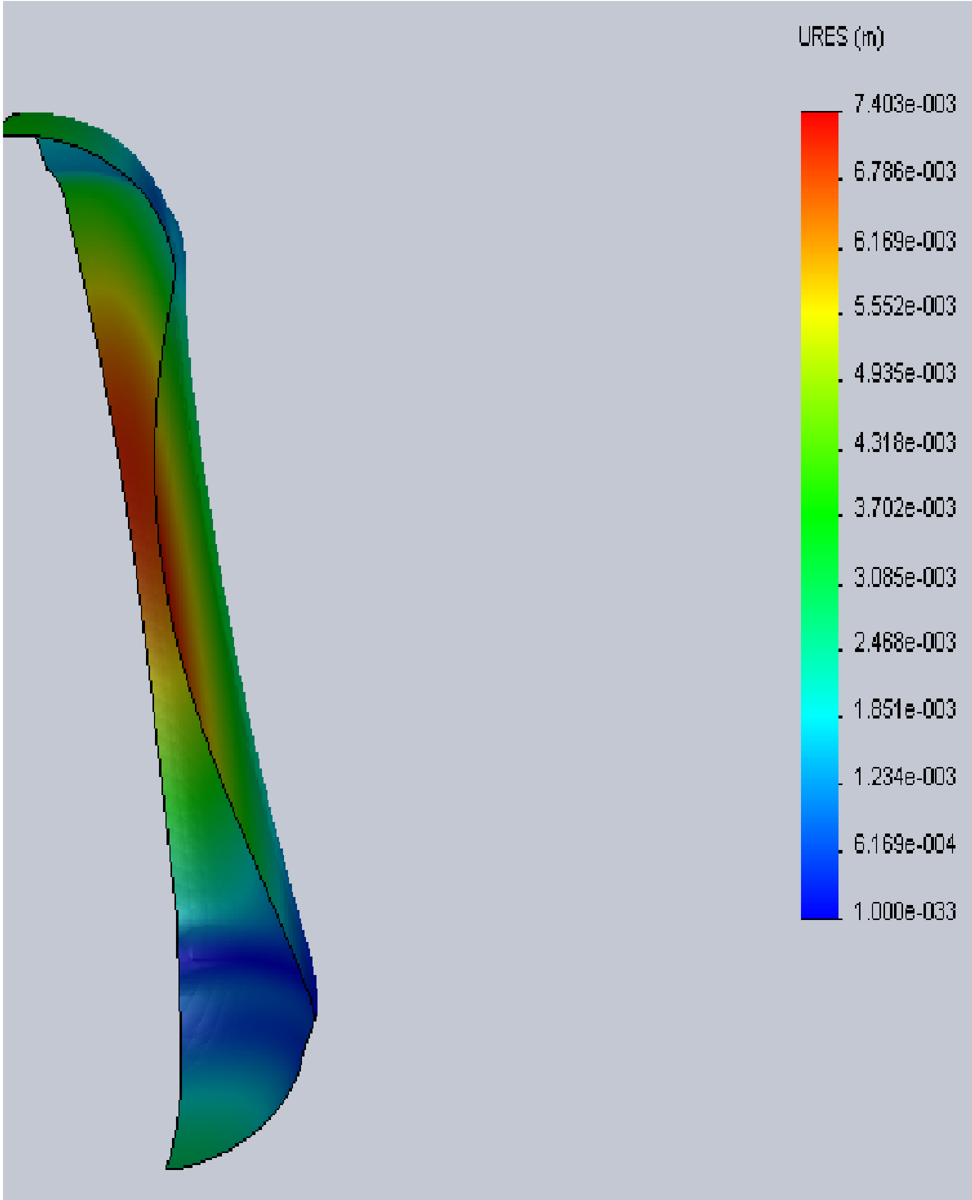

Figure 4: Strain in vessel with circularity error

In this situation, the circularity error reached 66.36, the maximum displacement was 3.085 mm in longitudinal direction and close to 7.403 mm in transversal direction.

Stress in a vessel without circularity error is illustrated in fig. 6.

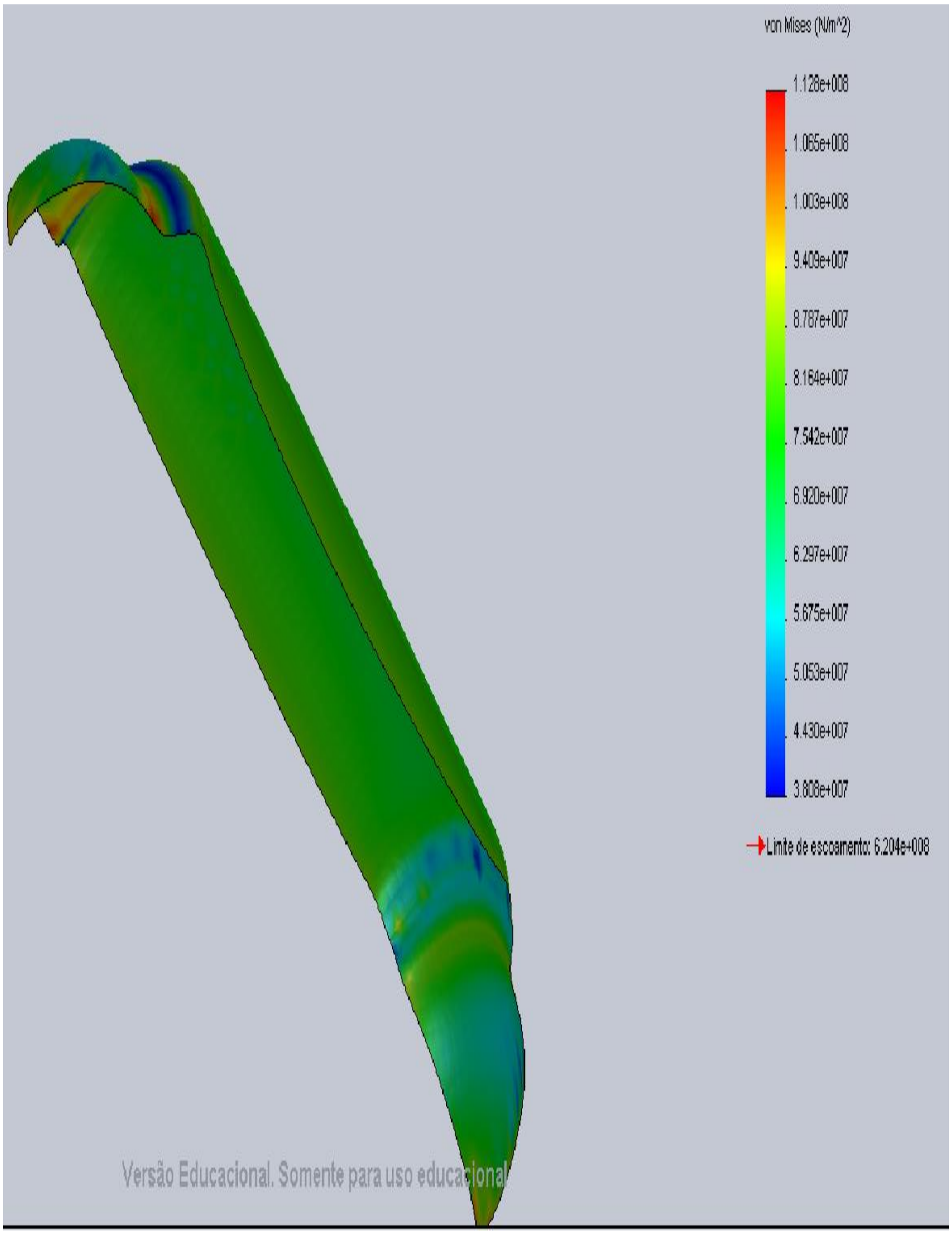

Figure 6: Stress in vessel without circularity error

In this situation, without circularity error, the maximum stress in the vessel was 112.8Mpa in a transition point between the greater and lower radii of lids. Stress did not reach the value of 81.64 Mpa in cilindrical part of vase

Stress in a vessel with circularity error is illustrated in fig. 7.

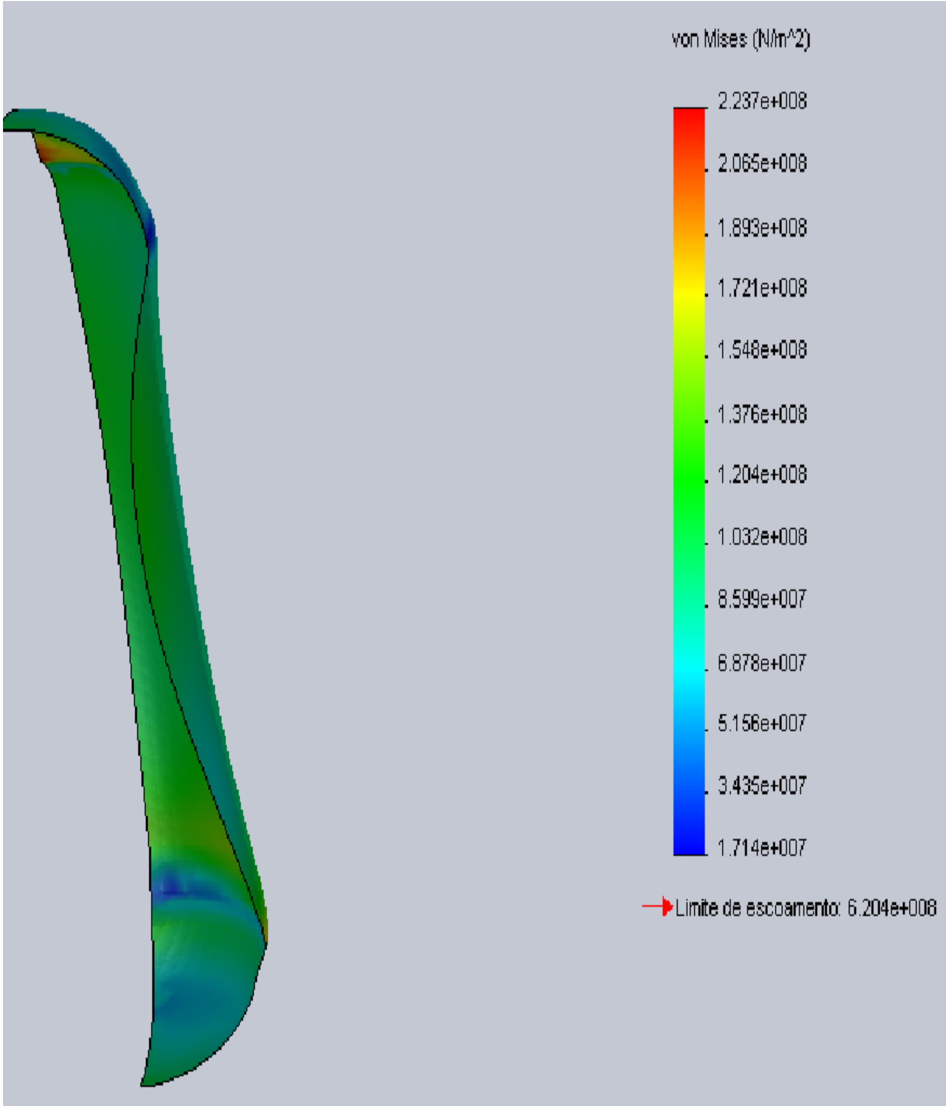

Figure 7: Stress in vessel with circularity error

In this situation, without circularity error, the maximum stress in the vessel was 223.7 Mpa in a transition point between the greater and lower radii of lids. Stress did not reach the value of 137.6 Mpa in cilindrical part of vase

Simulation shows that the stress is higher than 137,7 Mpa only in the transion between lower and high radii of lids. 137,7 MPa is the allowable stress accoding to ASME code in ASTM A516 GR70 in a range of temperatures of -30 and 90 ^{o}C.

The highest value of stress is 223,7 MPa, it is less than the yield stress (260 MPa) of the material utilized, therefore strain could be considered as a linear function of stress according to Cetlin and Helman (2012). Thereby, a graphic (fig. 8) was plotted in order to estimate the strain for stress above 137,7 MPa.

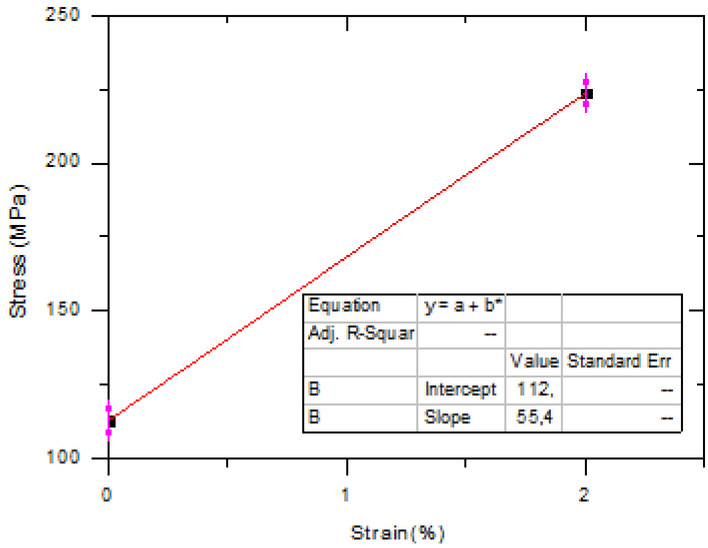

Figure 8: Stress vs. Strain curve

Using the stress of 137.7 MPa, the deformation would be 0.44%, which shows the highest circularity error is 14.6 mm approximately

CONCLUSION

Simulations have shown that stresses caused by circularity error tends to be greater in the point of transition between lower and higher radii of lids. It may happen due to high structural stiffness in that point.

PETROBRÁS N-268 is more conservative than ASME code, however, It is important to execute novel researchs about this subject. None of the standards cited takes account the thickness of material.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge PUC/Minas and FAPEMIG due to support.

REFERENCES

Abrantes, J. P.; Batalha, G. F.; Simulação Numérica Via Elementos Finitos da Conformação de Chapas Metálicas por Pressão Hidráulica (BULGE TEST) in: Congresso Brasileiro de Engenharia de Fabricação, 2003.

ASME VIII Divisão 1; 2007 Boiler and Pressure Vessel Code.;

Cetlin, Paulo Roberto e Helman, Horacio; 2012. Fundamentos da Conformação Mecânica. 2° Ed,;

Fialho, A. B; 2008. Cosmos. 1° Ed,;

Filho, J. S. P; Análise de Defeitos de Teste Hidrostáticos em Vasos de Pressão. *Masters degree thesis in mechanical engineering. Universidade Federal de Santa Catarina.*

Lotti, R. S. et al; 2006. Aplicabilidade Científica do Método dos Elementos Finitos. *Revista Dental Press Ortodon Ortop Facial*. Maringá, v. 11, n. 2, p. 35-43;

Meirelles, C. R. M. et al; 2009. Aplicação da Modelagem em Elementos finitos na Concepção das Cascas de Concreto Armado. CG Computação Gráfica 135.

N-253 K; Projeto de Vasos de Pressão. PETROBRÁS,2010;

N-268 G; Fabricação de Vasos de Pressão. PETROBRÀS, 2012;

NR 13; Norma Regulamentadora de Caldeiras e Vasos de Pressão, 1978.

Telles, P. C. S., 1996.Vasos de Pressão. 2° Ed;

Vlack, L. H. V.,1998. Princípio de Ciência e Tecnologia dos Materiais. 2° Ed;