We are IntechOpen, the world's leading publisher of Open Access books Built by scientists, for scientists

6,900
Open access books available

185,000
International authors and editors

200M
Downloads

Our authors are among the

154
Countries delivered to

TOP 1%
most cited scientists

12.2%
Contributors from top 500 universities



WEB OF SCIENCE™

Selection of our books indexed in the Book Citation Index in Web of Science™ Core Collection (BKCI)

Interested in publishing with us?
Contact book.department@intechopen.com

Numbers displayed above are based on latest data collected.
For more information visit www.intechopen.com



15

Reduction of Stresses in Cylindrical Pressure Vessels Using Finite Element Analysis

Farhad Nabhani*, Temilade Ladokun and Vahid Askari
Teesside University, School of Science and Engineering, Middlesbrough, TS1 3BA, UK

1. Introduction

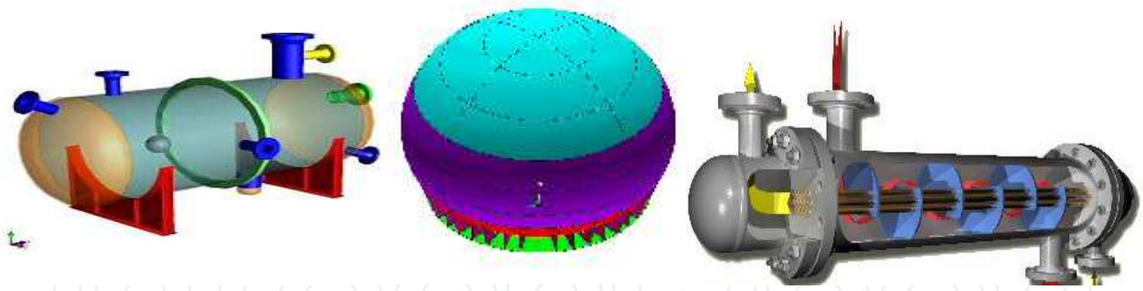
Due to the differential operating pressure of pressure vessels, they are potentially dangerous and accidents involving pressure vessels can be deadly and poses lethal dangers when vessels contents are flammable/explosive, toxic or reactive. Stress induced operating factors (e.g., process-upset, catalyst regeneration) and stress related defects (e.g., fatigue creep, embrittlement, stress corrosion cracking) accounts for approximately 24.4% of reoccurring catastrophic pressure vessels failures in process industries, many of which has resulted in loss of several lives, properties and in some cases preventive measures of evacuation of residents and community enforced (Sirosh & Niedzwiecki 2008). Pressure vessels store large amounts of energy; the higher the operating pressure - and the bigger the vessel, the more the energy released in the event of a rupture and consequently the higher the extent of damage or disaster or the danger it poses, (American Petroleum Institute 2001). To prevent stress related vessel rupture and catastrophic failure, it is necessary to identify the main factors that contribute extensively to stress development in pressure vessels and how they can be mitigated. This work presents critical design analysis of stress development using 3D CAD models of cylindrical pressure vessels assembly and finite element engineering simulation of various stress and deformation tests at high temperature and pressure.

2. Applications of pressure vessels

Pressure vessels are air-tight containers used mostly in process industry, refinery and petrochemical plant to carry or hold liquid, gases or process fluids. The commonly used types of pressure vessels in the industry are heat exchangers, tanks, towers, boilers, drums, condensers, reactors, columns, air cool exchangers and the usual shape employed in their design are cylinders, cone and spheres as shown in figures below.

Any pressure vessel in-service poses extreme potential danger due to the high pressure and varying operating temperature, hence there should be no complacency about the risks. Unfortunately, pressure vessels accidents happen much more than they should.

* Corresponding Author



Subfigure 1: 3D Model of a Boiler (b) Subfigure 2: 3D Model of a Drum (c) Subfigure 3: 3D Model of a Heat Exchanger
Source: PM Projects & Services Pvt Ltd.

Fig. 1. Subfigure with three images

2. Rate of pressure vessels accidents

Bulk Transporter (2009) reported that the National Board of Boiler and Pressure Vessel Inspectors in the US recorded the number of accidents involving pressure vessels at an increase of 24% over the course of a year between 1999 and 2000. These statistics include power boilers, steam heating boilers, water heating boilers, and unfired pressure vessels. However, the increased number of accidents was not reflected through to the number of fatalities, as these actually dropped by 33% over this period. By broadening this search, it can be seen that the reporting period of 1992 to 2001 saw a total of 23,338 pressure vessel related accidents which averages at 2,334 accidents per year. Reporting year 2000 saw the highest number of accidents at 2,686 with the lowest at 2,011 in 1998 (National Board of Boiler and Pressure Vessel Inspectors, 2002).

The number of fatalities as a direct result of boiler and pressure vessel accidents has been recorded as 127 over the past 10 years (Air-conditioning, Heating, Refrigeration- The News (2002). During the reported period between 2001 and 2008, the statistics show that the rate of accidents that were directly linked to pressure vessels is not yet on the decline.

3. Causes of pressure vessel failures

The main causes of failure of a pressure vessel are as follows:

- Stress
- Faulty Design
- Operator error or poor maintenance
- Operation above max allowable working pressures
- Change of service condition
- Over temperature
- Safety valve
- Improper installation
- Corrosion
- Cracking
- Welding problems
- Erosion

- Fatigue
- Improper selection of materials or defects
- Low –water condition
- Improper repair of leakage
- Burner failure
- Improper installation Fabrication error
- Over pressurisation
- Failure to inspect frequently enough
- Erosion
- Creep
- Embrittlement
- Unsafe modifications or alteration
- Unknown or under investigation

4. Stresses in pressure vessels

Stress is the internal resistance or counterforce of a material to the distorting effects of an external force or load, which depends on the direction of applied load as well as on the plane it acts. At a given plane, there are both normal and shear stresses (Engineers Edge 2010). However, there are planes within a structural component subjected to mechanical or thermal loads that contain no shear stress. Such planes are principal planes, the directions normal to those planes are principal directions and the stresses are principal stresses. For a general three-dimensional stress state there are always three principal planes along which the principal stresses. (Spence. J & Tooth.A.S. (1994).

Different types of stresses as stated in Chattopadhyay. S. (2004) are as follows:

i. Pressure stresses
ii. Thermal stresses
iii. Fatigue stresses
iv. Local stresses
v. External stresses
vi. Compressive stresses
vii. Bending stresses
viii. Normal stress
ix. Circumferential stresses
x. Longitudinal stresses
xi. Radial stresses
xii. Tangential stresses
xiii. Tensile stresses
xiv. Shear stress
xv. Bending stress
xvi. Principal stress

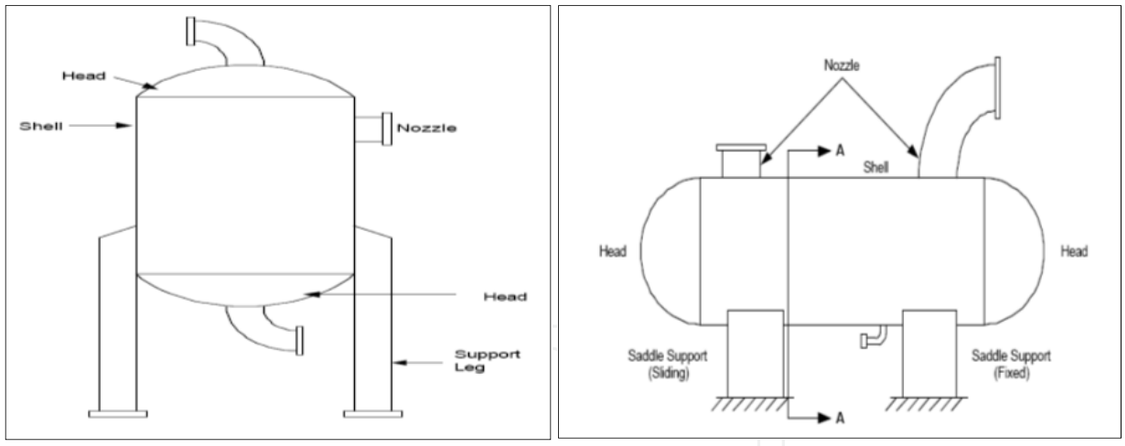
According to D. R. Moss (2004) stresses are generally categorised as primary, secondary or peak stresses. Primary stresses are stresses due to pressure (internal or external), mechanical loads and wind which can result in the rupture or total collapse of a pressure vessel, they are the most hazardous. Secondary stresses on the other hand are strain-induced stresses,

and can be developed at the junction of major components of a pressure vessel (e.g. radial loads on nozzles) because of stresses caused by relenting load or differential thermal expansion. While Peak stresses are the maximum stress concentration point in addition to the primary and secondary stresses present in a region. Peak stresses are only significant in fatigue conditions and are the sources of fatigue cracks, which are applicable to membrane, bending and shear stresses (Rao. K. R. 2002).

When a thin-walled cylinder is subjected to internal pressure, three mutually perpendicular principal stresses will be set up in the cylinder material, namely the circumferential or hoop stress, the radial stress and the *longitudinal* stress, (Sharma.S.C .2010). Provided that the ratio of thickness to inside diameter of the cylinder is less than 1/20**,** it is reasonably accurate to assume that the hoop and longitudinal stresses are constant across the wall thickness, and that the magnitude of the radial stress set up is so small in comparison with the hoop and longitudinal stresses that it can be neglected. This is obviously an approximation since, in practice, it will vary from zero at the outside surface to a value equal to the internal pressure at the inside surface (Hearn.E.J.1998).

5. Vessel description

The arrangement of a typical pressure vessel is shown in Fig. 2. A typical pressure vessel consists of shell (body of the vessel), closure heads, openings for inspection and instrumentations, and a combination of nozzles for pressure relief or other purpose and supports (Syed U. A. (2009).



Subfigure 1: Typical horizontal vessel (b) Subfigure 2: Typical vertical arrangement
Source: Megyesy. E.F (2001)

Fig. 2. Subfigure with two images

6. Finite element model of pressure vessels

In order to proceed with the analysis, three Finite element models were designed and denoted as design case one, two and three. A finite element model consists of boundary conditions, mesh of elements and nodes. Each component of pressure vessel analysed for stress and deformation at the design conditions of 137 MPa and 400^{o}C for all the cases

considered. Wind loads and seismic loading were also taken into account. All finite element analyses were run using ANSYS 10.0.

The design parameters taken are as follows:

- Design Code ASME BVPC 2007
- Shell Material SA-723, grade 1, UNS No k23550, class 3
- Design Pressure 137 MPa
- Design Temperature 400 ^{o}C
- Tensile Strength of Material 965 MPa
- Material Rating at design temperature 806 MPa
- The shell to be used is fully radiographed, hence E is 100%
- Monobloc shell assumed.

6.1 Model design case one

The pressure vessel model assumed a cylinder with semi ellipsoidal top head and hemi-spherical bottom head capped. Considering the shell to be monobloc and the design rule that the design pressure P_D should not exceed the limit set by article KD-251.1 (division 3 of ASME BPVC section 8) given as:

$$P_D = \frac{1}{1.732}(Sy)\ \ln(Y) \tag{1}$$

$$137 = \frac{1}{1.732}(965)\ \ln(Y)$$

$$Y = 1.28$$

Where Y is the ratio of outer diameter (D_o) to the inner diameter (D_i) of the shell, $Y = D_o/D_i$, and ratio of 2 is assumed for safety. Assumed D_o =3048 mm, D_i=1524 mm, Length = 3 x D_o and flange thickness = 254 mm. The ellipsoidal top head is considered to be fully radiographed like shell, hence E = 1. Then using given equation:

$$t = \frac{PDk}{2SE - 0.2P} \tag{2}$$

Where K is the stress intensity factor by the equation below:

$$K = \frac{1}{6}\left[2 + \left(\frac{a}{b}\right)^2\right] \tag{3}$$

$$t = \frac{137 * 3048 * 1}{2 * 965 * 1 - 0.2 * 137}$$

Thickness = 220 mm

The required thickness of the bottom hemispherical head is normally one-half the thickness of an elliptical or torispherical head for the same design conditions, material, and diameter which will be 110 mm in this case.

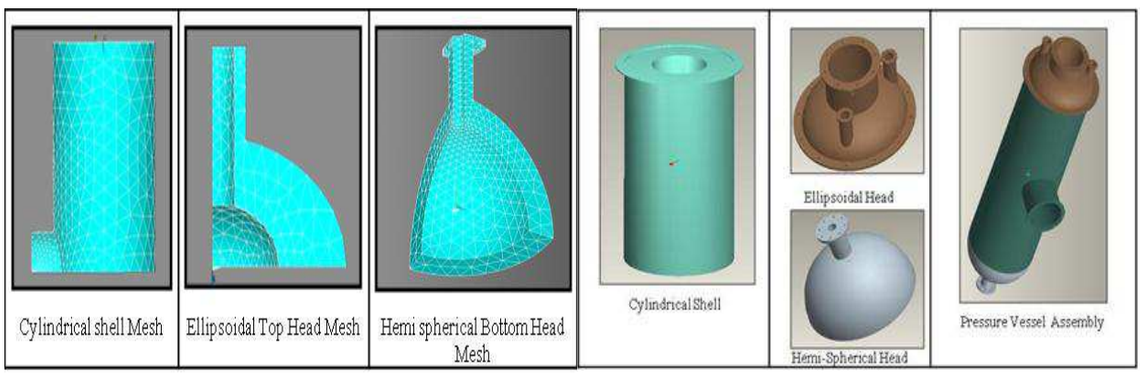
For the openings and closures, which are an important requirement for safety, is given as:

$$t_{rn} = \frac{\Pr}{SE - 0.6P} \tag{4}$$

$$t_{rn} = \frac{137 * 1524}{965 * 1 - 0.6 * 137}$$

Where r is the inside radius of nozzle, assuming a 1524 mm nozzle, the required thickness comes out to be 254 mm.

The CAD model of the pressure vessel is shown below:



Subfigure 1: Design model for case one

Subfigure 2: Meshing of pressure vessel components for case one

Fig. 3. Subfigure with two images

6.1.1 Element type

Each model part was meshed with SOLID 45 element with a higher concentration of elements around the nozzles and stress concentration areas. Eight nodes having three degrees of freedom at each node define the element translations in the nodal x, y, and z directions. The element has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities. Subfigure 2 of Fig.3 above shows the meshing of the model parts.

6.1.2 Boundary conditions

Axis symmetric displacement boundary conditions was applied to the two axes to ensure that the body, bottom end of the shell and the head are fully constrained with internal pressure loadings of 137 MPa and temperature of 400 oC applied. The finite analysis results are shown in tables 1 and 2 below.

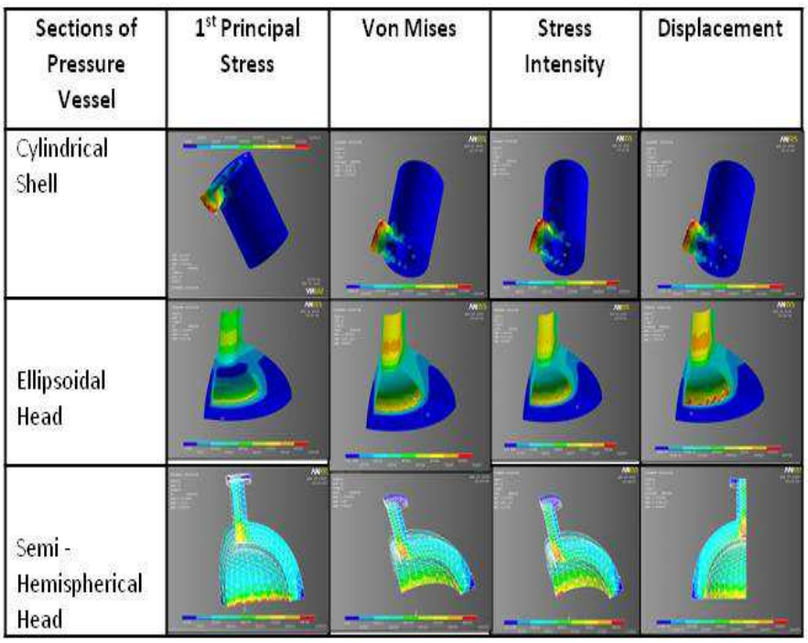


Table 1. Result of finite analysis for design case one

Sections	1st Principal Stress (MPa)	Von Mises (MPa)	Stress Intensity (MPa)	Displacement (mm)
Cylindrical Shell	2562.59	2541.21	2552.94	0.38605
Ellipsoidal Head	704.35	623.94	720.15	0.09479
Semi Hemispherical Head	726.73	744.73	770.86	0.11313

Table 2. Summary of finite analysis result for design

6.2 Case two design model

$$P_D = \frac{1}{1.732}(Sy)\ \ln(Y)$$

After the finite element analysis of design case one, large stress was found around the nozzle junction of the cylindrical shell and also found stress developed at the bottom and top head. Ways of mitigating the stress is addressed in the model design case two. The design parameters are kept the same as in Design case one except for the temperature rating tensile strength of the material that is 806 MPa. The diametric ratio Y assumed is 1.5 derived from equation (1):

Outer and inner diameter assumed were 3048 mm and 2030 mm respectively. Reconsidering the nozzle thickness in design case two, the value of S is taken as 806 MPa using equation (4):

$$t_{rn} = \frac{\mathrm{Pr}}{SE - 0.6P}$$

Thickness = 152 mm

To mitigate the stress discovered during the finite element analysis of design case one, reinforcement pad is introduced to lower the stresses around the junction of shell and nozzle of the pressure vessel model. The total cross-sectional area of reinforcement A required in any given plane for a vessel under internal pressure should be not less than:

$$A = dt_r F \tag{5}$$

Where

d = diameter in the given plane of the finished opening, (mm).

t_r = minimum thickness which meets the requirements of KD-230 in the absence of the opening (mm).

And F is calculated from the graph in non-mandatory appendix H of section 8 division 3 of ASME BPVC to be 1.

Hence,

$$A = 1524 * 152 * 1$$
$$A = 9144\, mm^2$$

Using the formula for the area of circular disc given as,

$$A_{disk} = \pi(r_o^2 - r_i^2) \tag{6}$$

The outer diameter of the reinforcement pad = 1625 mm, while thickness = 101 mm. The top head is 2:1 ellipsoidal with thickness assumed calculated as 279 mm using equation (2):

$$t = \frac{PDk}{2SE - 0.2P}$$

Thick head with pressure on the concave side is used for the bottom hemispherical with given equation:

$$t = \frac{5 * 137 * 1524}{6 * 806} \tag{7}$$

$$t = \frac{5PL}{6S}$$

$$t = 216.9\ mm$$

6.2.1 Element type

SOLID 92 elements was used to mesh each model part in design case two, due to its quadratic displacement behaviour, its suitability to model irregular meshes and accuracy. Ten nodes having three degrees of freedom at each node define the elements: translations in

the nodal x, y, and z directions. In addition, symmetric boundary condition was used; hence, half of the cylindrical shell and quarter of heads model were used for the analysis. The result of the finite simulation is shown in table 3 and summarised in table 4 below:

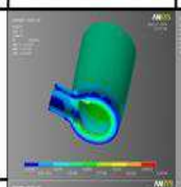
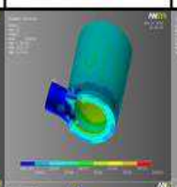
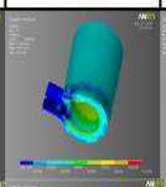
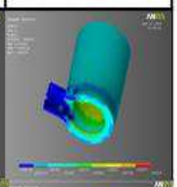
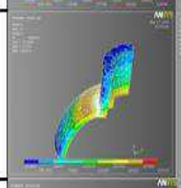
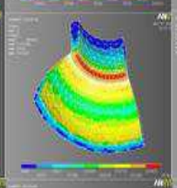
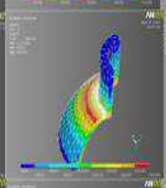
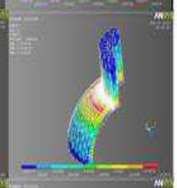
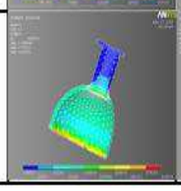
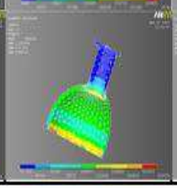
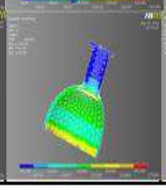
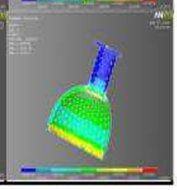
Sections of Pressure Vessel	1st Principal Stress	Von Mises	Stress Intensity	Strain
Cylindrical Shell				
Ellipsoidal Head				
Semi Hemispherical Head				

Table 3. Result of finite analysis for design case two

Summary of FEA Results	1st Principal Stress (MPa)	Von Mises (MPa)	Stress Intensity (MPa)	Displacement (mm)
Cylindrical Shell	792.31	695.74	780.40	0.15532
Ellipsoidal Head	2106.58	1777.62	1946.97	0.38747
Semi Hemispherical Head	2171.67	1653.04	1902.59	0.37863

Table 4. Summary of finite element results

6.3 Case three design model

In design case 3, the shortcomings of both design cases 1 and 2 were considered from design point of view to lower the stresses to an acceptable range. Reinforcement pad is used along with the use of diametric ratio of 2.0, i.e. Y = 2 as calculated in design case 1. Outer diameter of 3048 mm and the inner diameter of 1524 mm with length of cylindrical shell being three times the outer diameter for structure stability. Nozzle thickness kept same as design case 2.

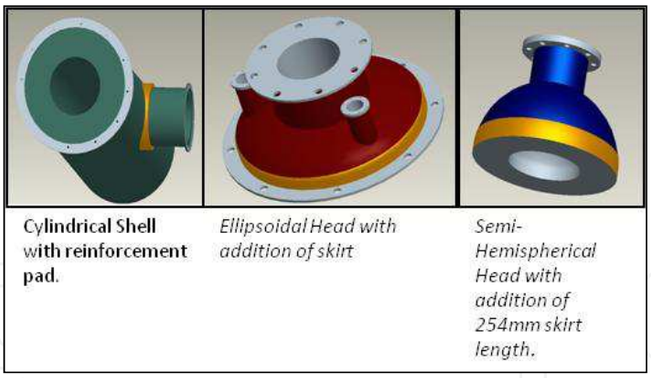


Fig. 4. Design model for case three

6.3.1 Element type

Each component of the pressure vessel is simplified using symmetry. Solid 3D Element was used to mesh the models with fine mesh around the nozzle and shell junction. Pressure and temperature of 137 MPa and 400 ^{o}C applied on the internal surface of the components. The results obtained from finite analysis are given below:

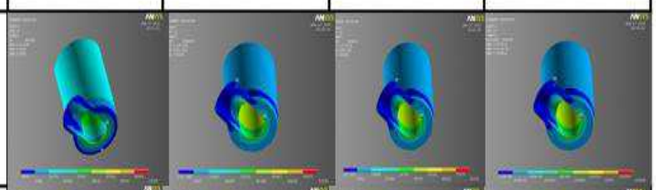
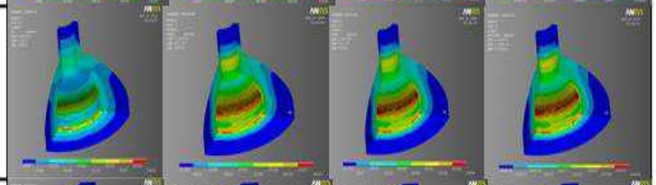
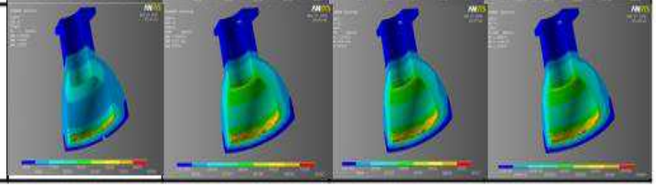
Sections of Pressure Vessel	1st Principal Stress	Von Mises	Stress Intensity	Strain
Cylindrical Shell				
Ellipsoidal Head				
Semi Hemispherical Head				

Table 5. Result of finite analysis for design case three

Sections of Pressure Vessel	1st Principal Stress (MPa)	Von Mises (MPa)	Stress Intensity (MPa)	Strain (mm)
Cylindrical Shell	509.99	499.92	568.98	0.1132
Ellipsoidal Head	475.91	400.37	449.9	0.0895
Semi Hemispherical Head	659.11	567.5	567.5	0.1129

Table 6. Summary of finite element analysis result

7. Discussion

Finite element analysis result in design case one was compared with the allowable stress intensities of ASME BVPC Division 3 and it was found that the stresses in the cylindrical shell exceeded the value of the allowable stress with almost 70 %. Also, the result shows higher stress development along the nozzle length away from the shell and at the bottom semi spherical head. However, the stresses developed in ellipsoidal enclosures head has a very less margin than the allowable stress of about 4%, which can be improved by using better thickness to diameter ratio.

In the design case two model, the shell thickness was reduced and thickness of the nozzle increased as well as the addition of reinforcement pad of high alloy steel SA-705 grade XM-13 around the nozzle to shell junction. The finite element analysis shows that less stresses were developed in the shell wall than the allowable and there was a reduction of stress from 2552 MPa in design case one to 780 MPa in design case two; about 70% drop in stresses. However, despite the decline in the stresses to 780 MPa, it is still higher than the allowable stress by the ASME Code. Finally in the design case three, the cylindrical shell was modified and the reinforcement pad introduced in design case two retained to reduce stresses around the nozzle to shell junction and a skirt length of 254 mm was added around the enclosure heads. This is necessary due to the result of the analysis obtained in the design cases one and two which showed that highest stresses were concentrated on the lower end of the heads where there was formation of joins to the shell. The finite element analysis shows that with the addition of the skirt, there was lesser development of stresses in the lower ends of the heads. The table below shows the comparison of the stresses obtained from finite element analysis and the stresses allowed by the ASME Code.

Sections	Stress Intensity MPa		Allowable Stress MPa
	Min	Max	
Cylindrical Shell	1.42	568.9	747
Ellipsoidal Head	0.62	449.9	747
Semi Hemispherical Head	0.951	567.5	747

Table 7. Stress Intensity comparison of design case three

8. Conclusion

Three main factors are seen to contribute extensively to the development of stresses in pressure vessels. These are thickness, nozzle positions and the joints of the enclosure heads. From the model design cases used in this research, it could be seen that as the thickness of pressure vessel increases, the stresses decreases, however this is not a viable solution due to cost. Nozzles though are safety relief devices and important component of pressure vessels comes with its own disadvantages of increasing weak areas and stress concentration. However, this was mitigated by use of high alloy reinforcement pads as applied in the design case two and three of this work. The high strength reinforcement pad used has a chemical composition of titanium 0.4 to 1.20%; hollow disc shaped with rectangular section can also reduce the stresses concentration around the nozzle.

Finally, the joints of enclosure heads either welded or bolted were identified as areas with the highest concentration of stresses i.e. with peak stress. Addition of 254 mm skirt length at the end of enclosure heads provide more room for the stresses to develop slowly in the wall of the head regions, thus making the pressure vessels more resistant to the loadings.

9. References

AirConditioning,Heating,Refridgeration - The News (2002) . Boiler Accident Report : "To Err Is Human" .URL:http://www.achrnews.com

American Petroleum Institute (2001). Inspection of Pressure Vessels (Towers, Drums, Reactors, Heat Exchangers, and Condensers), Recommended Practice 572, Second Edition.

Bulk Transporter (2009) . "Boiler, Pressure Vessel Accidents Increase" . URL: http://bulktransporter.com

Chattopadhyay. S (2004): Overview of Pressure Vessels. Pressure Vessels Design and Practice. 3rd edition: CRC Press, pp. 1-10. United States of America.

D. R. Moss (2004). Pressure Vessel Design Manual – Illustrated Procedures for Solving Major Pressure Vessel Design Problems. 3rd Edition, Gulf Professional Publishing, Oxford, United Kingdom

Engineers Edge (2010) – Design for Manufacturability Reference and Training Book. Stress-Strength of Materials.
URL: http://www.engineersedge.com.

Hearn.E.J. (1998). Mechanics of Materials, 2nd edition, Cambridge University Press, United Kingdom.

Megyesy. E.F (2001). Design of Tall Towers. Pressure Vessel Handbook. Pressure Vessel Publishing, Inc. United States of America.

Rao. K. R.(2002). Companion guide to the ASME boiler & pressure vessel code, ASME Press. United States of America.

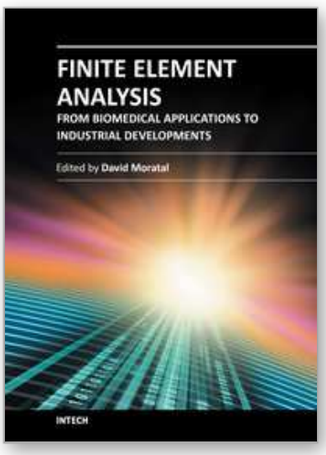
Sharma. S. C. 2010. Strength of Materials. Department of Mechanical & Industrial Engineering. Indian Institute of Technology Roorkee. URL: http://nptel.iitm.ac.in/courses/Webcourse.

Sirosh. N & Niedzwiecki. A. (2008). Hydrogen Technology: Development of Storage Tanks - High Pressure Vessels, Springer- Verlag, Berlin Heidelberg.

Spence. J & Tooth.A.S. (1994). Pressure Vessel Design: Concepts and Principles. Taylor & Francis; 1st edition, London.

Syed U. A. (2009). Design and Analysis of Pressure Vessels under Elevated Temperature and Pressure. MSc Thesis, University of Teesside, United Kingdom.

The National Board of Boiler and Pressure Vessel Inspectors (2002). National Board Bulletin. URL: http://www.nationalboard.org



Finite Element Analysis - From Biomedical Applications to Industrial Developments

Edited by Dr. David Moratal

ISBN 978-953-51-0474-2
Hard cover, 496 pages
Publisher InTech
Published online 30, March, 2012
Published in print edition March, 2012

Finite Element Analysis represents a numerical technique for finding approximate solutions to partial differential equations as well as integral equations, permitting the numerical analysis of complex structures based on their material properties. This book presents 20 different chapters in the application of Finite Elements, ranging from Biomedical Engineering to Manufacturing Industry and Industrial Developments. It has been written at a level suitable for use in a graduate course on applications of finite element modelling and analysis (mechanical, civil and biomedical engineering studies, for instance), without excluding its use by researchers or professional engineers interested in the field, seeking to gain a deeper understanding concerning Finite Element Analysis.

How to reference

In order to correctly reference this scholarly work, feel free to copy and paste the following:

Farhad Nabhani, Temilade Ladokun and Vahid Askari (2012). Reduction of Stresses in Cylindrical Pressure Vessels Using Finite Element Analysis, Finite Element Analysis - From Biomedical Applications to Industrial Developments, Dr. David Moratal (Ed.), ISBN: 978-953-51-0474-2, InTech, Available from: http://www.intechopen.com/books/finite-element-analysis-from-biomedical-applications-to-industrial-developments/reduction-of-stresses-in-cylindrical-pressure-vessels

INTECH
open science | open minds

InTech Europe
University Campus STeP Ri
Slavka Krautzeka 83/A
51000 Rijeka, Croatia
Phone: +385 (51) 770 447
Fax: +385 (51) 686 166
www.intechopen.com

InTech China
Unit 405, Office Block, Hotel Equatorial Shanghai
No.65, Yan An Road (West), Shanghai, 200040, China
中国上海市延安西路65号上海国际贵都大饭店办公楼405单元
Phone: +86-21-62489820
Fax: +86-21-62489821

© 2012 The Author(s). Licensee IntechOpen. This is an open access article distributed under the terms of the Creative Commons Attribution 3.0 License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

IntechOpen

IntechOpen