NIGERIAN INSTITUTE OF INDUSTRIAL ENGINEERS

2011 INTERNATIONAL CONFERENCE

THEME: Infrastructural Development for Industrialization: Challenges and Prospects

DATE: August 4 - 6, 2011.

VENUE**:** CONFERENCE CENTRE, UNIVERSITY OF IBADAN, IBADAN, NIGERIA.

N.I.I.E. 2011 Conference Proceedings

PREFACE

The theme of this year's Conference; Infrastructural Development for Industrialization: Challenges and Prospects is apt. We are all witnesses to the weakening and bleeding of the economy due to lack of supporting infrastructure. Added to this is low productivity. Improved productivity is necessary to establish a "comfortable" STANDARD OF LIVING for ALL. There is also no doubt that Industrialization would remain a pipe dream in our circumstances; except we begin to do 'things differently'.

The response to the Call for Papers for the Conference was fairly good. Several papers (30) were received and referred. A good number of submitted papers needed changes which could be classified as "major". Only the accepted papers and corrected papers (21) were selected for the Conference Proceedings. As part of the review process, papers were reviewed by at least two referees.

During preparations for the Conference, the Interim President, Professor A. F Akinbinu was called to eternal rest. We appreciate and applaud his leadership. May his gentle soul rest in peace. Amen.

We would like to thank various persons who have contributed to the preparation of this proceeding, book of abstracts as well as organization of the Conference. Professor D. E Osifo deserves special mention; being the engine that propelled the organization of the conference through several advisories.

We also thank all authors, and wish everyone a successful Conference.

Professor A. E Oluleye
Chairman, Technical Committee

Dr. O. G Akanbi
Secretary, Interim EXCO

August 4, 2011

NIIE 2011 INTERNATIONAL CONFERENCE PROGRAMME

Date: Friday 05 August 2011

Time: 11.30 AM – 1.00 PM

Co-Chairs: Professor Nick Damachi / Professor Godwin Ovuworie

Session 1

Date: Friday 05 August 2011
Time: 2.00 – 4.00 PM

Co-Chairs: Professor Festus Olorunniwo / Engr. Sunny Etok

Session 2

Friday 05 August 2011 2.00 – 4.00 PM

Co-Chairs: Dr. Christian Bolu / Dr. Sociis Okolie

Session 3

FINITE ELEMENT MODELING OF STRESS DISTRIBUTION IN SPHERICAL LIQUIFIED NATURAL GAS (LNG) PRESSURE VESSELS

Adeyefa. O. A and *Oluwole. O. O
Mechanical Engineering Department
University of Ibadan, Nigeria.

ABSTRACT

This work investigated the distribution of Von Misses stress in LNG Spherical Carbon Steel Storage tanks. Using the Finite Element Method and equations of elasticity, constant thickness carbon steel spherical storage tanks of 40 in. dia. 70in. dia of 1 in. shell thickness were subjected to different loading conditions from 500 to 4000Psi in incrementals of 500 Psi. Spherical triangular elements based on shallow shell formulation were used for the model. The element has five degrees of freedom at each corner node, which are the essential external degrees of freedom without the degree of freedom associated with the in-plane shell rotation. The displacement fields of the element satisfy the exact requirement of rigid body modes of motion. The FORTRAN 90 coding was developed to obtain maximum Von Misses stress distribution with the tank subjected to different internal pressure and wind loadings. The results were then compared with the yield stress of the material of the tank. Von Misses stress is used as yield criteria whether to change tank material or increase the shell material thickness if yield stress is higher than the Von-Misses Stress. Results showed Von- Mises stresses for a 40 in dia. Spherical shell with 1 in shell thickness able to withstand internal pressure loading alone up to 3500 Psi after which the shell thickness will no longer be able to withstand the loading. The 70in. dia. Vessel could only withstand internal pressure loading up to 2000 Psi. Validation of Finite Element modeling was done using ASME Section VIII Div 1 standard. Modeled results were observed not to be significantly different from ASME values (P>0.05). External wind effects alone on small dia. vessels was seen to be constant for all sides of the pressure vessel.

Keywords: Von Misses Stress, FE Modeling, LNG, Spherical Storage Tank, ASME, ASCE

- **Corresponding author:** oluwoleo2@asme.org;+234(0)8033899701

1. Introduction

Spherical pressure vessels are the most efficient pressure vessels because they offer maximum volume for the least surface area and the required thickness of a sphere is one-half the thickness of a cylinder of the same diameter. When compared with a cylindrical vessel, for a given volume, a sphere would weigh approximately only half as much. However, they are more expensive to fabricate, so they are not used extensively until the need for large-sized pressure vessels arise. They are typically utilized as “storage” vessels rather than “process” vessels and are economical for the storage of volatile liquids and gases under pressure. Consequently, the need for reduced weight for long term storage provides a new challenge for LNG storage tank design. These new designs provide an opportunity to apply advanced materials and finite Element Method concepts in an effort to reduce the overall weight of the tank and keep both the stress distribution and the volume at an acceptable and practical level.

Stress analysis problems of pressure vessels entail two types of considerations, i.e., applied loads and the corresponding response. A realistic way to tackle the first consideration is to design the loads, which have a high probability of occurring during a rigorously specified usage of the vessel. To tackle the second consideration - predicting the response of the structure to specified loads is through the solution of the vessels governing equations. Elastic problems are governed by linear differential equations which can be translated to set of simultaneous equations using finite element method.

Ji-hoon et al (2006) considered the safety evaluation for the general structure against local earthquake loading properties and the characteristics of site where 200,000 kilolitres capacity of above-ground LNG cylindrical storage tank would be constructed. The paper introduced the basic design concept and features of the large capacity 9% nickel full containment LNG storage tank developed by KOGAS. Korea Gas Corporation (KOGAS) developed the world’s largest above-ground full containment cylindrical LNG storage tank with a gross capacity of 200,000m^3. The main objective of the development of the large capacity LNG storage tank was to reduce the construction cost and the boil-off gas and use the tank building site more effectively. Eduardo (2005), in his Doctor of Philosophy thesis evaluated the stability of cylindrical above-ground steel tanks under imposed support settlements and wind pressures. The behavior of these tanks was evaluated by means of computational experiments performed using finite element models developed with ABAQUS.

Dong et al (2005) in their paper used the FEM code, MARC, to simulate the hydrobulging process of a single-curvature polyhedron, including loading and offloading. The distributions of stress and strain were acquired as well as other important data. Pascal (1999) in his study on seismic post elastic behavior was performed for an existing equipment with a volume of 1000 m^3 containing 85% of LPG. The middle diameter of the sphere is 12.50 m, its thickness varies from 36.2 mm to 36.8 mm taking into account the corroded thickness. In the dynamic nonlinear analysis, the plastic strain levels obtained in the columns are small in a general manner in all simulations. The maximum values are about 6%, under the failure limits of the steel (between 21% and 26%). No plastic strain is observed in the sphere in all cases. The equipment resists very well to the applied seismic loads by generating small plastic strains in local regions. On the other hand, the modal spectral analysis shows that a reduction factor of 2.3 should be used to find acceptable stresses in columns (2.7 for regulatory spectrum). To have acceptable stresses in the sphere, a value of 2.6 is necessary (3.0 for regulatory spectrum). For braces, necessary values of reduction factor are 3.3 or 4.0.

Dong (2007) used FEM to carry out a free vibration analysis, and a nonlinear collapse analysis for simulated seismic loadings in two different set of tanks. Step changes in thickness were accounted for in the first set of tanks, and the presence of a plate or conical roof or stiffening girder was considered for the second set of tanks. The procedures of the study were extensively validated by making comparisons with available results in the literature. They provided results for the frequencies of the tanks in empty state conditions, and for the collapse loads for geometries without imperfections.

Finite Element Theory

Displacement Field Requirements; - It is evident that the accuracy which may be obtained by the finite element method depends directly on the accuracy with which the deformation patterns are selected. The assumed deformation patterns should, as closely as possible, reproduce the distortions actually developed within the element. If the deformation patterns are not properly chosen, the deformations will not necessarily converge to correct values when the mesh size is decreased. On the other hand, very good results may be obtained with a very coarse mesh if the element deformation patterns selected closely correspond to the actual patterns. Thus, the most critical factor in the entire finite element analysis is the proper selection of the element displacement field. The usual method of representing the displacement field is by selecting interpolation functions and generalized displacements at a finite number of nodal points of each element. To fulfill the conditions of the principle of minimum potential energy, the interpolation functions must be such that the displacements along the inter-element boundaries are compatible.

2. Methodology

The assumed displacement method was employed in this work to develop a shallow triangular spherical shell element without an in-plane rotation as a sixth degree of freedom. A shallow shell formulation was used to obtain the displacement fields. The element has five degrees of freedom at each of the three corners. Therefore, there would be fifteen degrees of freedom per element.

Assumptions

- Uniform Pressure is assumed for the LNG storage tank
- According to ASCE 7 - 95, wind load varies with tank height. It is assumed that wind load is uniform which is acceptable for a small storage tanks

Displacement Functions

The assumed displacement relationships for the proposed triangular shallow shell are expressed in curvilinear coordinates (Fig.1). Polynomial displacement functions were used as interpolation functions; the polynomial being of the highest order that will permit evaluation of the coefficients. Since displacements u and v are known at three points, nodes 1, 2 and 3, the highest-order expressions which can be assumed for u and v are

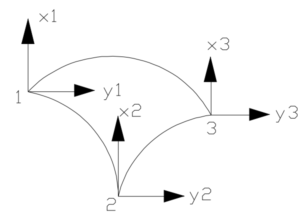

Fig. 1:- Shallow Triangular Element

$$u(x, y) = a_1 + a_2 x + a_3 y \tag{1}$$

$$v(x, y) = a_4 + a_5 x + a_6 y \tag{2}$$

The displacement w with its derivates, θ_x , θ_y has nine known values; hence, it may be assumed that

$$w(x, y) = a_7 + a_8 x + a_9 y + a_{10} x^2 + a_{11} xy + a_{12} y^2 + a_{13} x^3 + a_{14} xy^2 + a_{15} y^3 \tag{3}$$

from which it follows that

$$\theta_x (x, y) = \frac{\partial w}{\partial y} = a_9 + a_{11} x + 2a_{12} y + 2a_{14} xy + 3a_{15} y^2 \tag{4}$$

$$\theta_y (x, y) = -\frac{\partial w}{\partial x} = -a_8 - 2a_{10} x - a_{11} y - 3a_{13} x^2 + 2a_{14} y^2 \tag{5}$$

to determine constants as, known displacements at nodes are substituted and the equations become

$$[a] = [A^{-1}][\delta] \tag{6}$$

Where [δ] is the nodal degrees of freedom, [A^{-1}] is inverse of transformation matrix and [a] is vector of independent constants.

Strain Displacement Relationship

The strain-displacement relationships for thin shells are simplified for the shallow shell and expressed as follows in curvilinear coordinates (Reissner, 1960).

$$\varepsilon_x = \frac{\partial u}{\partial x} + \frac{w}{r},\ \varepsilon_y = \frac{\partial v}{\partial y} + \frac{w}{r},\ \varepsilon_{xy} = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x},\ k_x = -\frac{\partial^2 w}{\partial x^2},\ k_x = -\frac{\partial^2 w}{\partial y^2} \qquad (7)$$

The strain equations can be written in matrix form after necessary substitutions of u, v and w into the above strain equations.

Stress in a Triangular Element

Stress varies from point to point along the shell profile and also through the thickness of the shell. It is thus in reality an unknown function of two variables, therefore leads us to the equations below:

$$\sigma_b = \frac{6M}{t^2} \qquad \sigma_m = \frac{N}{t} \qquad (8)$$

Where: M is the moment per unit length, M and σ_b is the bending stress at the surface.

N is to be force per unit length and σ_m which is membrane stress

Strain Energy

The strain energy of an isotropic linear shell is given by [8]

$$U = \iint_A \int_{-\frac{t}{2}}^{\frac{t}{2}} \frac{E}{2(1-v^2)} \left[\varepsilon_x^2 + \varepsilon_y^2 + 2v\varepsilon_x\varepsilon_y + \frac{1}{2}(1-v)\gamma_{xy}^2 \right] d\varsigma dx dy \qquad (9)$$

Where, t = thickness of the shell, v = Poisson's ratio and E = Modulus of elasticity

After substitution for strains in the above expression and integration with respect to ς, the strain energy can be separated into the membrane energy U_m and the bending energy U_b.

$$U = U_m + U_b \qquad (10)$$

$$U_m = \frac{Et}{2(1-v^2)} \iint_A \left[e_x^2 + e_y^2 + 2ve_x e_y + \frac{1}{2}(1-v)e_{xy}^2 \right] dx dy \qquad (11)$$

$$U_b = \frac{Et^3}{24(1-v^2)} \int\int_A \left[k_x^2 + k_y^2 + 2vk_x k_y + \frac{1}{2}(1-v)k_{xy}^2 \right] dxdy \qquad (12)$$

The potential energy is then written as: $\Phi = U - W$ where W represents the work done by the external load on the system. In the finite element method, the potential energy of a shell is expressed as:

$$\Phi = \sum_{k=1}^{n} \phi_k \qquad (13)$$

where ϕ_k is the potential energy of the k^th^ element.

Stiffness Matrix

By writing strain energy equations in terms of displacements, element stiffness matrix can be determined in the usual manner**,**

$$k_m = t[A^{-1}]^T \iint_A B_m^T D_m B dxdy [A^{-1}] \qquad (14)$$

$$k_b = t[A^{-1}]^T \iint_A B_b^T D_b B dxdy [A^{-1}] \qquad (15)$$

k_m and k_b are element stiffness matrices due to membrane and bending stresses respectively

D_m and D_b are elasticity matrices for membrane and bending stresses respectively

B_m and B_b are strain matrices for membrane and bending stresses respectively

Therefore, element total stiffness matrix is

$$k = k_b + k_m \qquad (16)$$

Element stiffness matrix is then combined to give system stiffness matrix.

Consistent Load Vector

The simplest method to establish an equivalent set of nodal forces is the lumping process. An alternative and more accurate approach for dealing with distributed loads is the use of a consistent load vector which is derived by equating the work done by the distributed load through the displacement of the element to the work done by the nodal generalized loads through the nodal displacements. If a triangular shell element is acted upon by a distributed load q per unit area in the direction of w, the work done by this load is given by:

$$P_1 = \int_A qwdxdy \qquad (17)$$

If w is taken to be represented by:

$$\{w\} = [C^T]\{a\} = [C^T][A^{-1}]\{\delta\} \tag{18}$$

where $[C^T]$ for the present element is given by

$$C^T = [0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 1 \quad x \quad y \quad x^2 \quad xy \quad y^2 \quad x^3 \quad xy^2 \quad y^3] \tag{19}$$

The work done by the consistent nodal generalized force through the nodal displacements $\{\delta\}$ is given by:

$$P_2 = \{F^T\}\{\delta\} \tag{20}$$

Hence, from equations (17 -20), the nodal forces are obtained

$$F = [A^{-1}]\iint [C^T][q]dxdy \tag{21}$$

Equation (21) gives the nodal forces for a single element; and the nodal forces for the whole structure are obtained by assembling the elements' nodal forces.

2.8 Boundary Conditions

In order to reduce computing time, symmetrical nature of the storage tank was considered. Shown below is the typical shallow spherical triangular shell mesh for finite element modeling of LNG spherical storage tanks. This mesh in the fig. 2 below has six shallow spherical triangular shell elements with eight nodes.

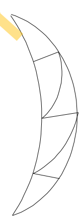

Fig. 2: Typical Spherical Shell Mesh

Each node has five degree of freedoms; therefore the mesh in figure 2 has forty degrees of freedom. In considering known displacements, all displacements at given node were given zero values with the exception of radial displacements, w.

2.9 Considerations

The direct stresses, Nx, N_y and the bending moment, Mx and M_y were computed along x and y axes respectively. It was necessary to determine also equivalent stresses using Von Misses yield criterion. The maximum values of Von Misses stresses were then compared with the yield stress of the tank material. In this work, the yield stress used for the tank shell material is allowable stress of the tank shell as it is given by ASME Section VIII Div 1. Symmetrical nature of the storage tank was considered in the coding thereby reducing computing time.

The spherical storage tanks considered have the following properties:-

First tank properties:

Radius = 20 in= 40 in. dia. vessel

Thickness = 1 in

Young Modulus of Elasticity = 30 * 10^6 Psi

Poisson Ratio = 0.3

Shell Material = A516M Grade 70

Shell Material Minimum Yield Stress = 38*10^3 psi

No of elements used in modeling = 23

Second tank properties:

Radius = 35 in= 70 in. dia. vessel

Thickness = 1 in

Young Modulus of Elasticity = 30 * 10^6 Psi

Poisson Ratio = 0.3

Shell Material = A516M Grade 70

Shell Material Minimum Yield Stress = 38*10^3 psi

No of elements that gave best results = 21

Sector angle = 50 degrees

3. Results and Discussions

Tables 1 shows the results obtained from the finite element modeling of internal stresses and the resulting von-mises stresses on the steel shells for 40in and 70 in . dia. Sphereical storage tanks.

Table 2 shows the effect of wind pressures alone and the corresponding Von-mises stress on both vessels.

Figures 3a and 4a show the variations in the calculated values of Von Misses given by Finite Element modeling and ASME Section VIII Div 1, Part UG for 40 in.dia. and 70in. dia. Pressure vessels.

The Von Misses values plotted on the graph were the maximum at the given loading conditions.

Figures 3b and 4b show equivalent maximum Von mises stresses were well within the limits of shell material minimum yield stress for internal pressure and wind loads between 500psi and 3500psi. for the 40in. dia. pressure vessel and between 500 Psi. and 2000Psi for the 70in. dia. Vessel.

At internal pressures of 3800 Psi and above, the equivalent maximum Von mises yield stresses exceeded the shell material minimum yield stress for the 40in.dia. pressure vessel. For the 70in. dia. Vessel, the material yields strtength was already exceeded at 2500Psi. Thus, the need to correctly design for spherical tank pressure vessels.

To prevent failure of LNG storage tank at above these limiting pressures, one of the followings has to be done.

- The shell thickness has to be increased accordingly so that equivalent maximum Von mises stress will be lower than the shell material minimum specified yield stress.
- Select shell material with higher minimum yield value so that the equivalent maximum Von mises stress calculated will be less than the shell material minimum specified yield stress.
- Decrease the working internal pressure of the spherical storage tank so that the equivalent maximum Von mises stress will fall below the shell material minimum specified yield stress.

Figures 3c and 4c show the effect of external wind pressure loads on the spherical shells. At external wind loads from 3800 Psi and above, the equivalent maximum Von mises yield stresses exceeded the shell material minimum yield stress for the 40in.dia. pressure vessel whilst, for the 70in. dia. Vessel, the material yields strength was already exceeded at 2500Psi.

Comparison of the results with ASME showed no significant difference (P>0.05).

Table 1: Internal Pressures only with the Equivalent Max Von Misses Stresses

Case No	Internal Pressure (Psi)	Max Von Misses Stress (Psi) for 40in. Dia. vessel	Max Von Misses Stress (Psi) for 70in. Dia. vessel
1	500	5003	8864
2	1000	10012	17729
3	1500	15042	26593
4	2000	20183	35457
5	2500	25471	44321
6	3000	30421	53186
7	3500	35511	62050
8	4000	40512	70914

Table 2: Wind load Pressure (Psi) only with the Equivalent Max Von Misses Stresses Given

Case No	Wind Load Pressure(Psi)	Max Von Misses Stress (Psi) for 40in. Dia. vessel	Max Von Misses Stress (Psi) for 70in. Dia. vessel
1	500	5003	8864
2	1000	10012	17729
3	1500	15042	26593
4	2000	20183	35457
5	2500	25471	44321
6	3000	30421	53186
7	3500	35511	62050
8	4000	40512	70914

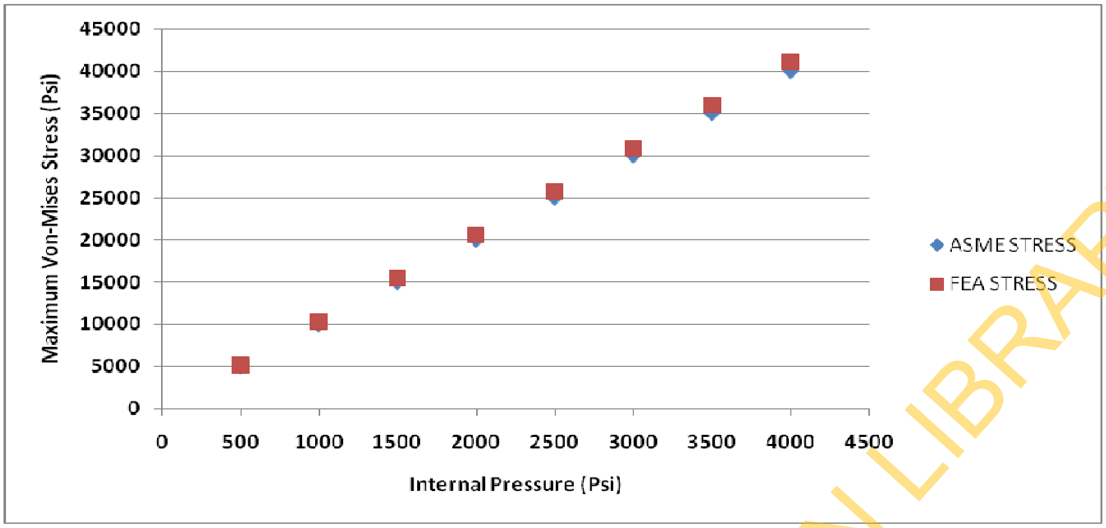

Fig. 3a: Internal Pressure Loading against Von Misses Stresses for 40 in. dia Spherical vessel and Sector Angle = 90 Degrees.

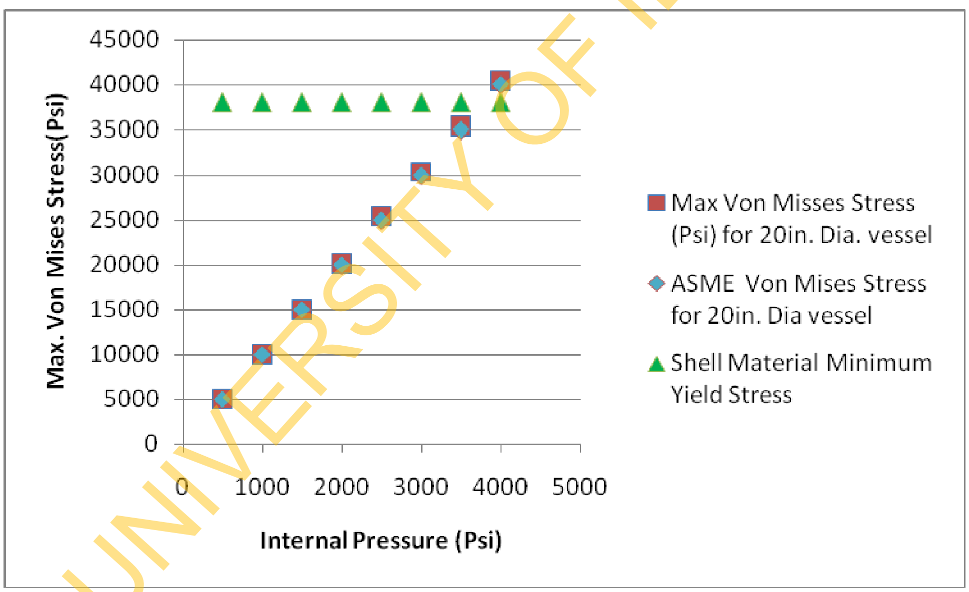

Fig. 3b: Internal Pressure Loading against Von Misses Stresses for 40 in. dia. Vessel showing the material yield point acting as limiting value for loading in spherical vessels.

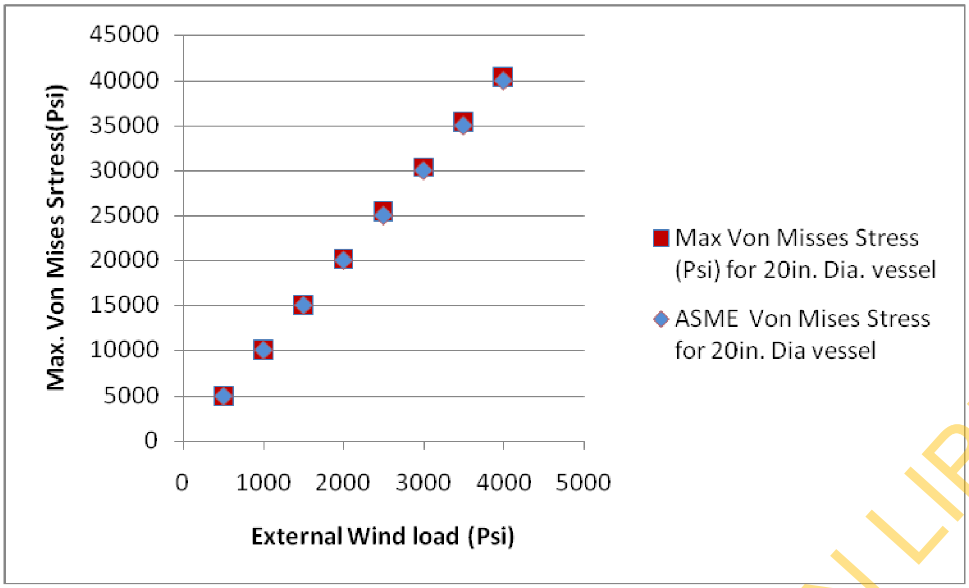

Fig. 3c: External wind Loading against Von Misses Stresses for 40 in. dia Spherical Vessel.

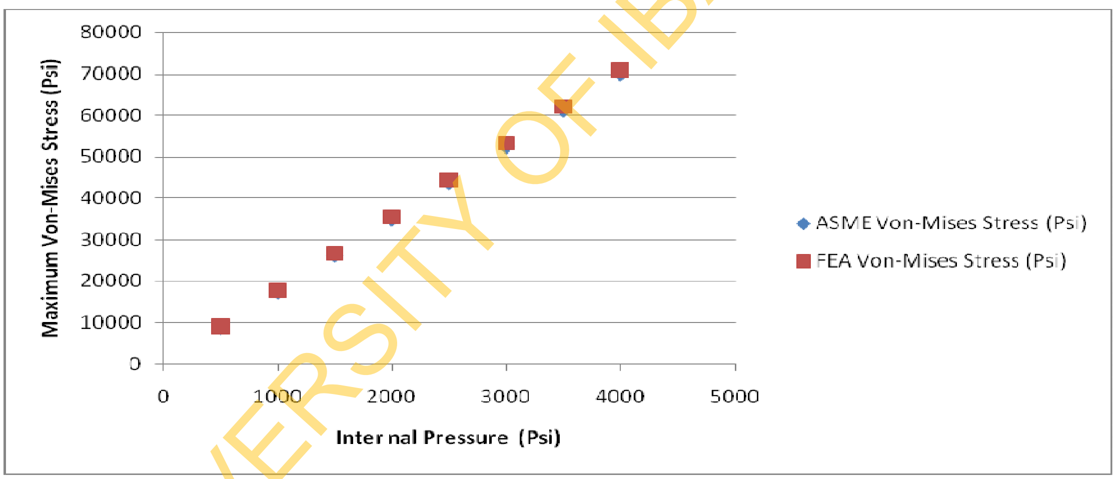

Fig. 4a: Maximum Von-Mises Stress (Psi) Against Internal Pressure Loading (Psi) for 70 in. dia Spherical vessel and Sector Angle = 50 Degrees.

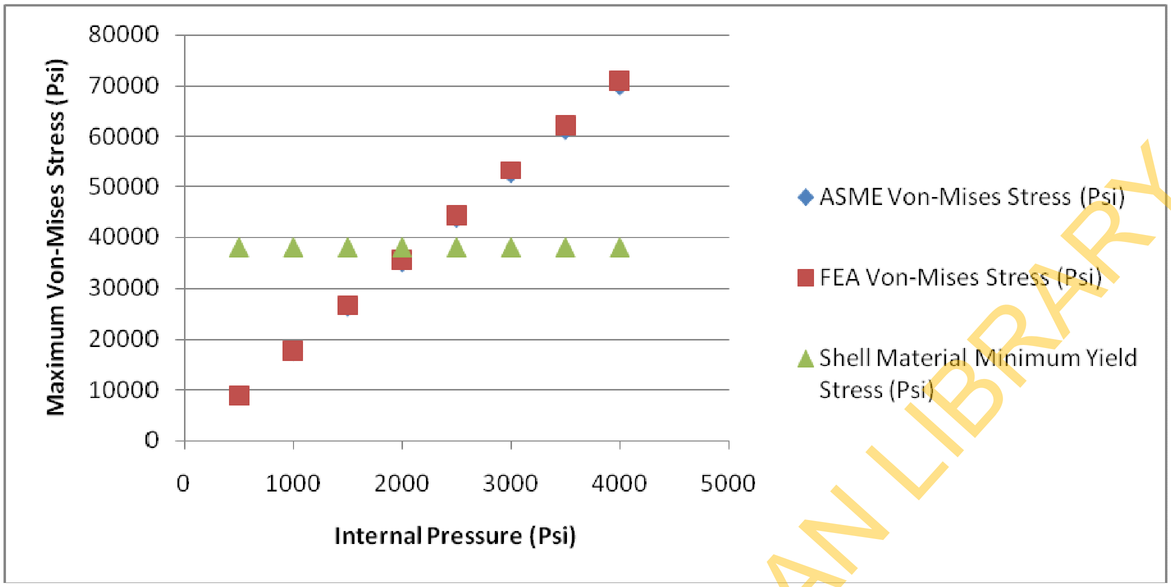

Fig. 4b: Maximum Von-Mises Stress (Psi) Against Internal Pressure Loading (Psi) for 70 in. dia. Vessel showing the material yield point acting as limiting value for loading in spherical vessels.

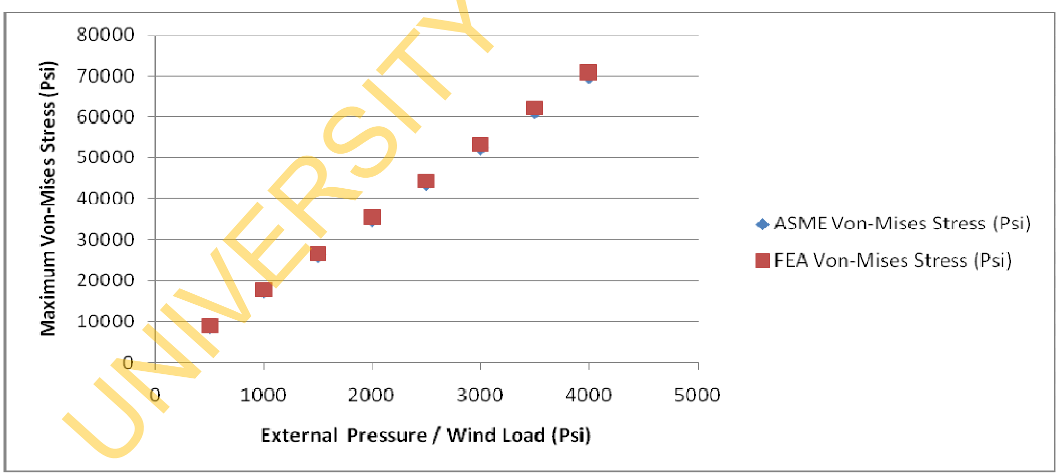

Fig. 4c: Maximum Von-Mises Stress (Psi) Against External wind Pressure Loading (Psi) for 70 in. dia. vessel.

4.0 Conclusions

The usefulness of finite element modeling of spherical steel shells for LNG pressure storage vessels has been indicated in this work. Also, the usefulness of ASME standards as guides in modeling and design. Engineers must design and model based on storage tank capacities and possible pressures to obtain the right sheet thickness for general material specification.

5.0 References

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