MODELING AND FINITE ELEMENT ANALYSIS OF PRESSURE VESSEL SHELL WITH RADIAL OPENINGS

Ch.Vasantha Lakshmi[1], A.V.Ramana Rao[2], G.Satish[3], M.Suryanarayana[4]

[1]*Associate professor, Department of Mechanical Engineering, Pragati Engineering College, Surampalem, A.P, INDIA*
[1]chvasantha31@gmail.com

[2]*Assistant professor, Department of Mechanical Engineering, Pragati Engineering College, Surampalem, A.P, INDIA*
[2]allaboina@gmail.com

[3]*Assistant professor, Department of Mechanical Engineering, Pragati Engineering College, Surampalem, A.P, INDIA*
[3]gsatish323@gmail.com

[4]*Assistant professor, Department of Mechanical Engineering, Pragati Engineering College, Surampalem, A.P, INDIA*
[4]msn1631@gmail.com

***Abstract*— It is to conduct stress analysis of a pressure vessel shell at the radial openings on the surface. The radial openings are not avoided from the surface of shell due to various piping attachments. Hence the stress analysis of shell and its ultimate failure under internal pressure beyond elastic limit is an appropriate scenario. The plastic zone appearing in vicinity of internal surface of cylinder propagates more fastly along radial openings side. When cylinder is unloaded it will cause reverse plasticity. Therefore it is proposed to obtain numerical solution using Finite Element analysis of cylindrical shell to obtain the displacement & radial stress distribution.**

In the work the stress analysis of pressure vessel shells with different pressure conditions is conducted Elastic analysis of uniform cylindrical shell with radial openings is found by Finite element method. It is observed that there are several factors which influence stress intensity factors. The Finite element analysis is conducted using Solid Edge. The results are presented in form of graphs and tables.

***Keywords*— Pressure vessel shell, radial openings, FEM, displacement, radial stress, Solid Edge.**

I. INTRODUCTION

Pressure vessels are used in chemical, petroleum; military industries and nuclear power plants .They are usually subjected to high pressures & temperatures. Industrial pressure vessel problems are witness ductile fracture of materials due to discontinuity in geometry or material of vessel characteristics. The conventional elastic analysis of pressure vessels to radial stresses & hoop stresses is applicable for the internal pressures up to yield strength of material. But the industrial vessels are undergoing pressure about yield strength of material. Hence a precise elastic-plastic analysis of all the properties of material is needed in order to make a use of load carrying capacity of the material & ensure safety with respect to strength of cylinders. As per Hook's law the stress is directly proportional to strain up to yield point. Beyond elastic point , the pressure vessel shell materials are partly elastic and partly plastic as shown in FIG. Plasticity is a mechanical property of materials to undergo irreversible deformation without any stresses or loads increase. Plastic materials with hardening in higher stresses at further plastic deformation. The two phases meet at one point, this phase exists till radial openings material becomes plastic. This phase is called Elastic-Plastic phase.

$$\sigma = E_T \epsilon^n$$

Where ET = Modulus of strain hardening.

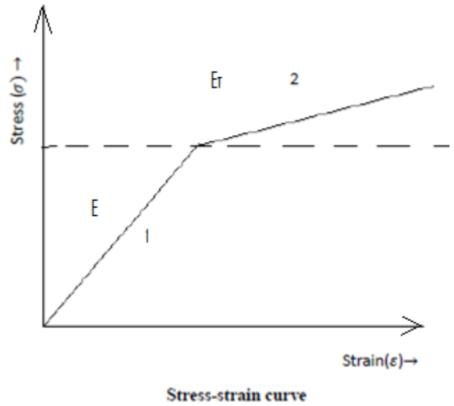

Fig.-1 Stress strain curve

The analysis of uniform pressure vessel shell can be conducted based on axi-symmetric conditions. However most of industrial pressure vessel providing openings in the cylindrical shell for various reasons such as Instrumentation, Burst in caports and Transfer of fluids. Presence of opening in the shell causes all local stress concentration on there. These stress concentration factors depends on size, shape, location of opening. These openings are called radial openings.

It is used to minimize the increasing of stress effect at the radial opening. 3-D solid model is needed to analyse the cylindrical shell with radial openings subjected to internal pressures. The radial radial openings are the initiation of plastic effects at lower pressures. The first plastic state point appears at edges intersection with cylindrical shell generated by radial openings axis. Therefore it is enough to analyse only one cylindrical shell going through shell and axis of radial openings. This plastic phase zone rapidly changed along radial openings and reaches external edge.

A. Applications:

General applications of pressure vessels used in metallurgical operations, process plants, air compressor units, hot water storage tanks, pneumatic reservoir, hydraulic tanks, storage for gasses like butane, LPG etc. The radial openings are cannot be avoided because of various duct arrangements or measuring gauge attachments. Hence to find stress distributions around the radial opening area is for design purpose. The radial openings are embedded in pressure vessel shell creates a problem in designing. The operating pressures are reduced or the material properties are strengthened. There is no such existing theory for the stress distributions around radial openings under varying internal pressures. That why in this work we concentrate on this area and relation between pressure and stress distribution. Here focus is on pure mechanical analysis. The pressure vessels are used to store fluid (i.e liquids and gases) under pressure. The fluid in the vessel may undergo a change in its state. The pressure vessels are designed with great care because explosion which may cause loss of life as well as property. The material of pressure vessels may be brittle or ductile such as cast iron and mild steel.

B. Objectives of work:

The following are the principal objectives of the work.

1. Stress and displacement analysis of pressure vessel shell with radial radial openings & understand the effect of relative parameters of radial openings on equivalent stress developed due to internal pressure.
2. Find out the residual stresses and displacements by using FEM Method by considering elasto-plastic state, for pressure vessel shell with radial radial openings.
3. Find the relationship between internal pressure applied and equivalent stress graphically for elastic-plastic state of pressure vessel shell with radial radial openings.

C. Elastic plastic state

To find the stress for elastic-plastic area have been derived by considering power-law hardening model, strain gradient theory for axi-symmetric problem.

$$\sigma_\theta - \sigma_r = \frac{r\partial \sigma_r}{\partial_r}$$

$$r\left(\frac{\partial \varepsilon_\theta}{\partial_r}\right) = \varepsilon_r - \varepsilon_\theta$$

From above equations is classical plasticity solution, final equations are:

$$p_i = \left(\frac{\sigma_y}{\sqrt{3}}\right)\left[\left(1 - \frac{r_c^2}{r_o^2}\right) + 2ln\frac{r_c}{r_i}\right]$$

$$\sigma_r = \left(\frac{\sigma_y}{\sqrt{3}}\right)\left[\left(-1 - \frac{r_c^2}{r_o^2}\right) - 2ln\frac{r_c}{r}\right]$$

$$\sigma_\theta = \left(\frac{\sigma_y}{\sqrt{3}}\right)\left[\left(1 - \frac{r_c^2}{r_o^2}\right) - 2ln\frac{r_c}{r}\right]$$

Where

σ_y = The yield strength of material

p_i = The internal pressure applied

D. Residual stress distributions:

It is assumed, the material follows HOOKE's law at unloading and the pressure is reduced elastically across the cylindrical shell.

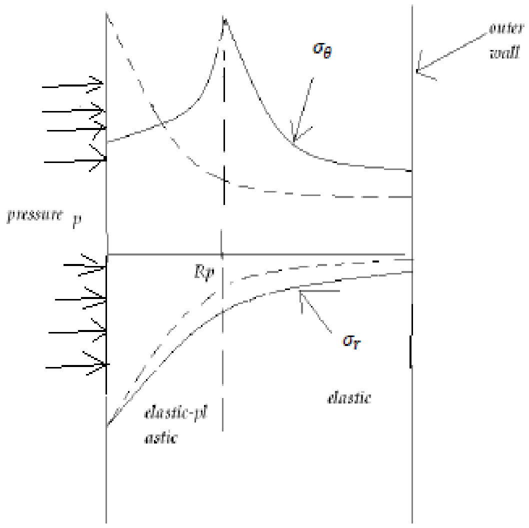

Fig.-2 Residual stress distribution

The dotted lines represent unloading distribution curves and continuous lines represents the loading distribution curves

$$\sigma_r = P_a\left[\frac{1 - \frac{r_o^2}{r^2}}{k^2 - 1}\right]$$

$$\sigma_\theta = P_a\left[\frac{1 + \frac{r_o^2}{r^2}}{k^2 - 1}\right]$$

Where

$$k = \frac{r_o}{r_i}, m = \frac{R_p}{r_i}, R_p = \sqrt{r_i * r_o}$$

The elastic stresses developed during loading condition can be given as

$$\sigma_\theta = \sigma_y[1 + \ln\left(\frac{r}{R_p}\right) - \left(\frac{1}{2}\right)\left(1 - \left(\frac{R_p}{r_o}\right)^2\right)$$

for $r_i \leq r \leq R_p$

$$\sigma_\theta = \frac{\sigma_y R_p^2}{2r_o^2(r_o^2 - R_p^2)}\left[1 - \frac{r_o^2}{r^2}\right]$$

for $R_p \leq r \leq r_o$

$$\sigma_r = \sigma_y[\ln\left(\frac{r}{R_p}\right) - \left(\frac{1}{2}\right)\left(1 - \left(\frac{R_p}{r_o}\right)^2\right)$$

for $r_i \leq r \leq R_p$

$$\sigma_r = \frac{\sigma_y R_p^2}{2r_o^2(r_o^2 - R_p^2)}\left[1 + \frac{r_o^2}{r^2}\right]$$

for $R_p \leq r \leq r_o$

$$\sigma_{res\ hoop} = \sigma_{\theta\ unloading} - \sigma_{\theta\ loading}$$

$$\sigma_{res\ hoop} = \sigma_{r\ unloading} - \sigma_{r\ loading}$$

Here no yielding occurs due to residual stresses. The stress distributions on the earlier loading distributions allow the two curves to be subtracted both hoop stress and radial stress and it generate residual stresses.

E. Cylinder with radial opening:

The elastic hoop stress concentration factor is defined as the ratio of maximum principal stress and lame's hoop stress on the inside surface of the pressurized cylindrical shell.

$$SCF = \frac{\sigma_{max}}{\sigma_{lame}}$$

k=1/β=Cylinder with wall ratio and
p = internal pressure
The reference stress is:

$$\sigma_{lame} = (\frac{k^2+1}{k^2-1})$$

SCF is used to define the peak loads for cyclic loading.
SCF= Actual stresses (with radial openings) theoretical stresses(without radial openings).

II. Finite Element Model

In the Finite Element method, symmetry is employed to avoid the analysis of radial openings cylindrical vessel shell. The uniform cylindrical shells having axis of symmetry and these are analysed using axi-symmentric elements. These elements adapted from a stress strain matrix and stiffness matrix is derived to the following formula

$$k = \iint B^T.D.B.dr.d\theta$$

B= strain displacement matrix.
D = stress strain matrix.
K= stiffness matrix.
Generally all iso-parametric elements can be used as axi-symmetric elements. It requires Young's modulus, Poisson's ratio, and yield stress and strain hardening modulus to conduct the finite element stress analysis. When there are radial openings on the surface of cylinder. The analysis has to be done using 3-D solid model. It can be created with the help of Solid Edge and finite element analysis is also done in the same software what I mentioned above. The 3 degree of freedom solid models are commonly generated in any modeling software's like Pro.E, CATIA , Solid Works and solid edge, the tetrahedron elements are by default.

III. Results and Discussions

The stress analysis results of uniform cylindrical pressure vessel shell and shell with radial radial openings subjected to internal pressures. Initially material & geometric data is described.

A. The geometry and material properties considered

In the designing of pressure vessels, generally ductile materials are used The main reason being, the ductile materials to withstand higher internal pressures and their ductile fracture can also be observed. In the present work, iron, grey cast type 20 is chosen for analysis taking for commercial application point of view.
The dimensions of the iron grey cast type 20 cylindrical shell taken:

Ri = 150 mm
Ro = 225 mm
L= 600 mm.

B. Material properties:

Study Property	Value
Study name	Static Study 1
Study Type	Linear Static
Mesh Type	Tetrahedral
Iterative Solver	On
NX Nastran Geometry Check	On
NX Nastran command line	
NX Nastran study options	
NX Nastran generated options	
NX Nastran default options	
Surface results only option	On

C. Constraints:

Constraint Name	Constraint Type	Degrees of Freedom
Fixed 1	Fixed	FREE DOF: None

D. Meshing Information:

Mesh type	Tetrahedral
Total number of bodies meshed	1
Total number of elements	2,205
Total number of nodes	3,856
Subjective mesh size (1-10)	3

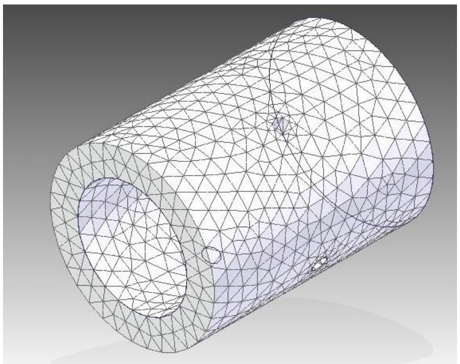

Fig.-3 Meshing Model of cylindrical pressure vessel shell

E. Translation analysis:

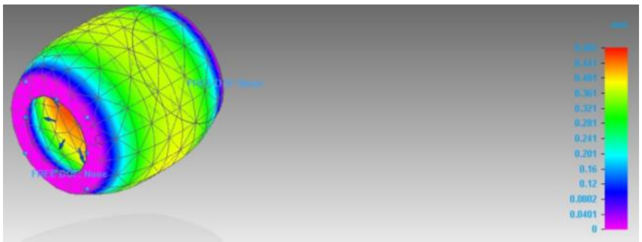

Fig.-4
Total translation of pressure vessel shell at one radial opening

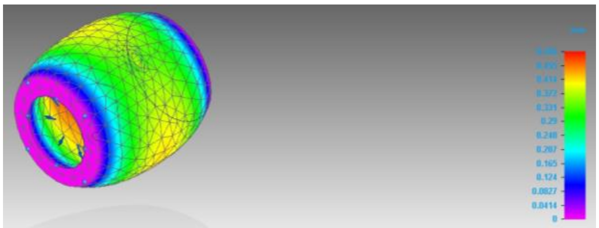

Fig.-5
Total translation of pressure vessel shell at two radial openings

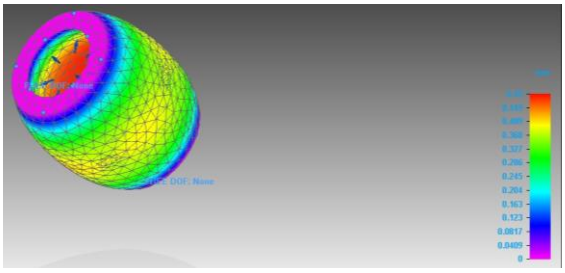

Fig.-6
Total translation of pressure vessel shell at three radial openings

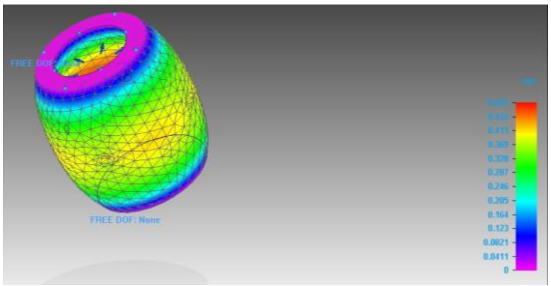

Fig.-7
Total translation of pressure vessel shell at four radial openings

TABLE-1
Total translations of pressure vessel shell at different pressures and openings

S.No	Pressure	No of Radial openingss	Result component: Total Translation	
			Extent	Value mm
1	70	1	Min.	0
	80			
	90		Max.	0.481
2	70	2	Min.	0
	80			
	90		Max.	0.496
3	70	3	Min.	0
	80		Max.	0.49
	90			
4	70	4	Min.	0
	80		Max.	0.493
	90			

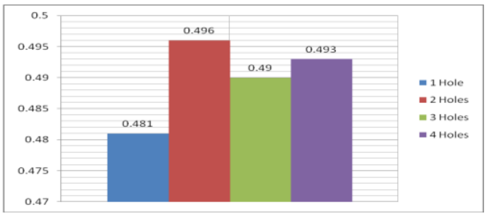

Fig.-8
Maximum translations at different no. of openings

F. *Factor of safety analysis*

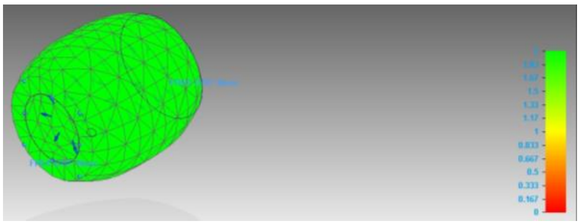

Fig.-9
Factor of safety of pressure vessel shell at one radial opening

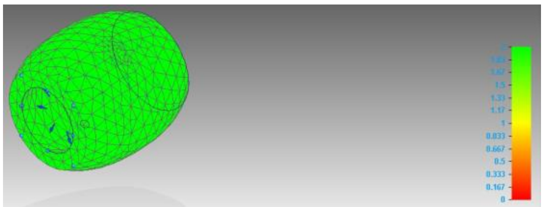

Fig.-10
Factor of safety of pressure vessel shell at two radial openings

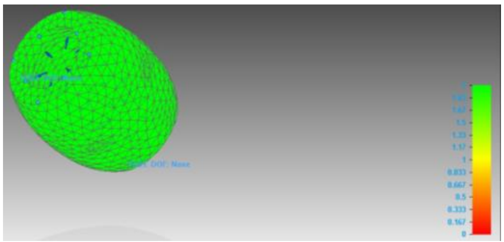

Fig.-11
Factor of safety of pressure vessel shell at three radial openings

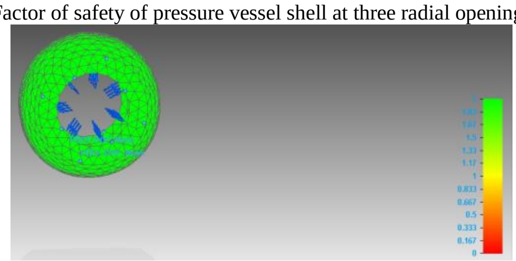

Fig.-12
Factor of safety of pressure vessel shell at four radial openings

TABLE-2

Factor of safety of pressure vessel shell at different pressures and openings

S.No	Pressure	No. of radial holes	Result component: Factor of safety	
			Extent	Value mm
1	70	1	Min.	0
	80			
	90		Max.	2
2	70	2	Min.	0
	80			
	90		Max.	2
3	70	3	Min.	0
	80		Max.	2
	90			
4	70	4	Min.	0
	80		Max.	2
	90			

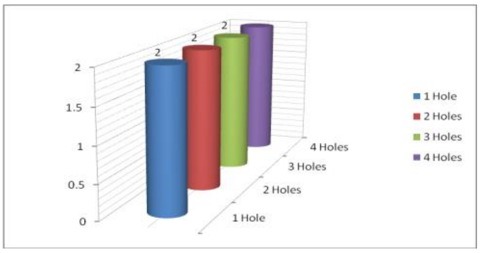

Fig.-13
Factor of safeties at different no. of openings

G. *Von-mises stresses analysis:*

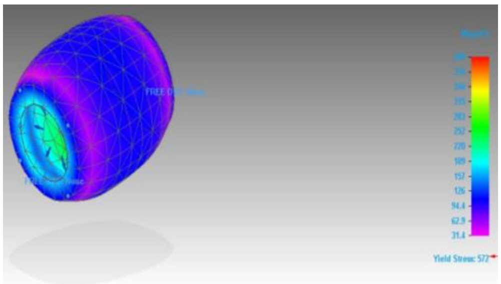

Fig.-14
Von mises stresses of pressure vessel shell at a pressure of 70 Mpa for one radial opening

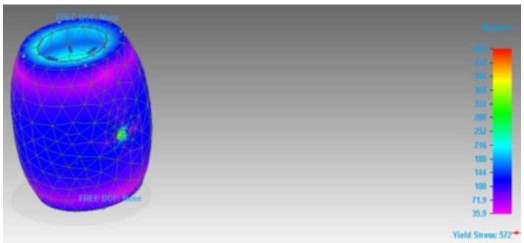

Fig.-15
Von mises stresses of pressure vessel shell at a pressure of 80 Mpa for one radial opening

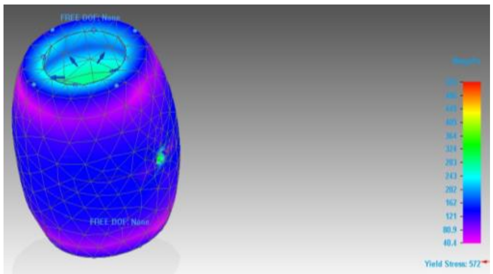

Fig.-16
Von mises stresses of pressure vessel shell at a pressure of 90 Mpa for one radial opening

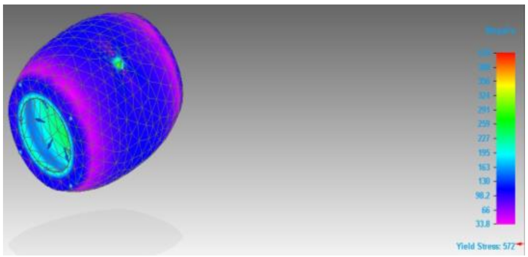

Fig.-17
Von mises stresses of pressure vessel shell at a pressure of 70 Mpa for two radial openings

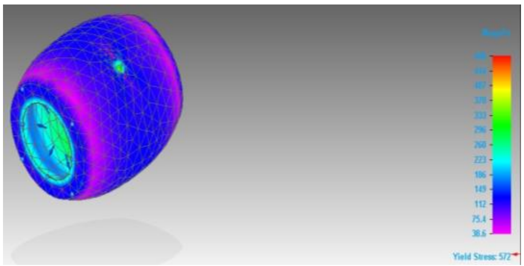

Fig.-18
Von mises stresses of pressure vessel shell at a pressure of 80 Mpa for two radial openings

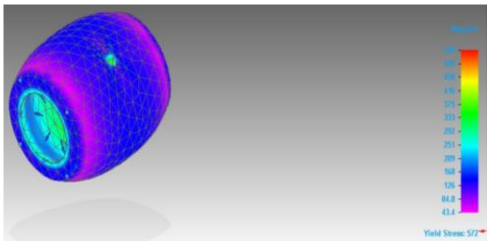

Fig.-19
Von mises stresses of pressure vessel shell at a pressure of 90 Mpa for two radial openings

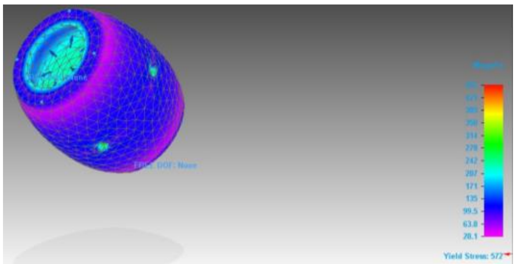

Fig.-20
Von mises stresses of pressure vessel shell at a pressure of 70 Mpa for three radial openings

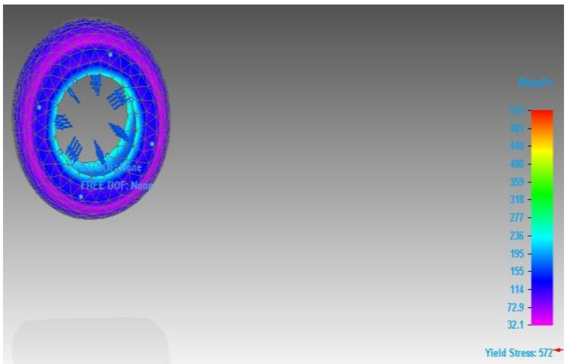

Fig.-21
Von mises stresses of pressure vessel shell at a pressure of 80 Mpa for three radial openings

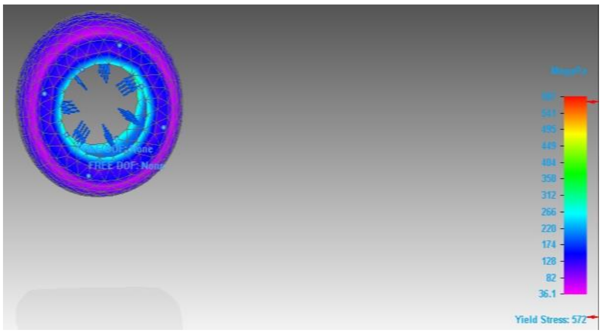

Fig.-22
Von mises stresses of pressure vessel shell at a pressure of 90 Mpa for three radial openings

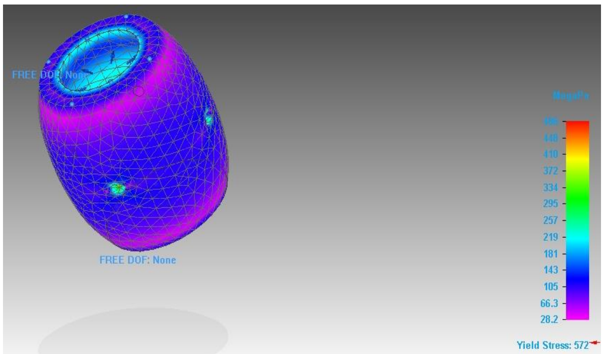

Fig.-23
Von mises stresses of pressure vessel shell at a pressure of 70 Mpa for four radial openings

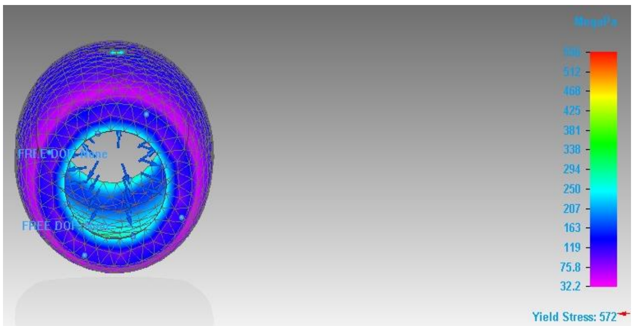

Fig.-24
Von mises stresses of pressure vessel shell at a pressure of 80 Mpa for four radial openings

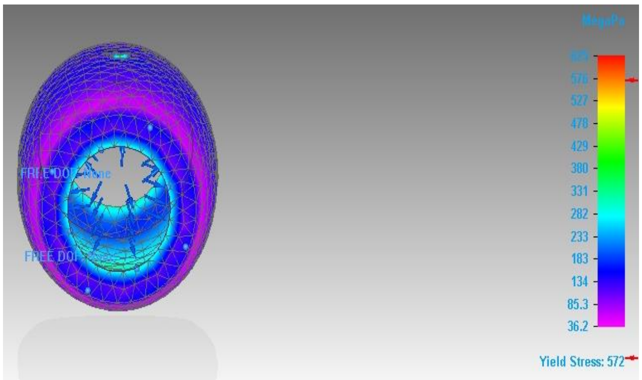

Fig.-25
Von mises stresses of pressure vessel shell at a pressure of 90 Mpa for four radial openings

TABLE-3
Von-mises stresses of pressure vessel shell at different pressures and openings

S.No	No. of radial openings	Pressure Mpa	Result component: Von Mises	
			Extent	**Value MPa**
1	1	70	Minimum	31.4
2			Maximum	409
3		80	Minimum	35.9
4			Maximum	468
5		90	Minimum	40.4
6			Maximum	526
7	2	70	Minimum	33.8
8			Maximum	420
9		80	Minimum	38.6
10			Maximum	480
11		90	Minimum	43.4
12			Maximum	540
13	3	70	Minimum	28.1
14			Maximum	457
15		80	Minimum	32.1
16			Maximum	522
17		90	Minimum	36.1
18			Maximum	587
19	4	70	Minimum	28.2
20			Maximum	486
21		80	Minimum	32.2
22			Maximum	556
23		90	Minimum	36.2
24			Maximum	625

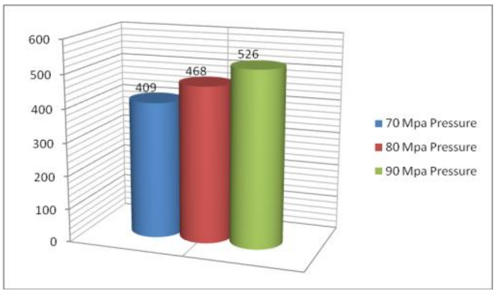

Fig.-26
Maximum Von-mises stresses at different pressures with one radial opening

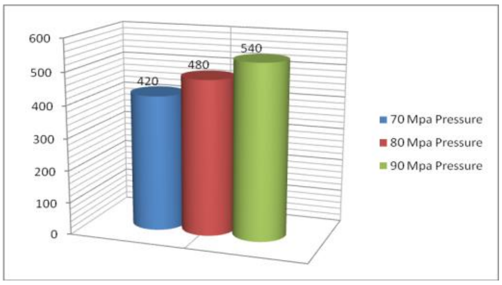

Fig.-27
Maximum Von-mises stresses at different pressures with two radial openings

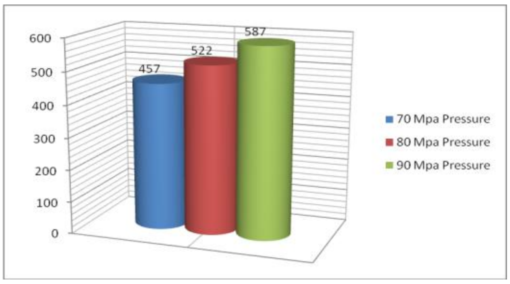

Fig.-28
Maximum Von-mises stresses at different pressures with three radial openings

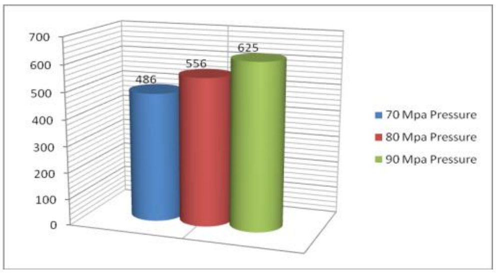

Fig.-29
Maximum Von-mises stresses at different pressures with four radial openings

IV. FUTURE SCOPE OF WORK

In this paper we attempted with a single radial openings vessel, the methodology can be extended to multiple radial openings case. This work has been made to know the load capacity of a cylindrical pressure vessel shell with radial radial openings (i.e 1, 2, 3 and 4). Based on available finite element models, three dimensional analyses have been carried out to predict the Von-mises stresses, displacements and factor of safety behavior along the shell wall especially at the cylinder bore. Solid Edge software have been used for develop 3-D model as well as analysis. The results can be compared with each other. The unloading behavior of cylindrical pressure vessel shell with radial openings is extension of this paper.

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