Optimization of Nozzle Location to Minimize Stresses on Horizontal Pressure Vessel

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Optimization of Nozzle Location to Minimize Stresses on Horizontal Pressure Vessel

Major Project Report

Submitted in partial fulfillment of the requirements
for the degree of
Master of Technology in Mechanical Engineering

(CAD/CAM)

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Declaration

This is to certify that

1. The thesis comprises my original work towards the degree of Master in Technology in Mechanical Engineering(CAD/CAM) at Nirma University and has not been submitted elsewhere for a degree.

2. Due acknowledgment has been made in the text to all other material used.

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Undertaking for Originality of the Work

I, **Patel Ketankumar V.**,Roll. No.**15MMCC19**,gives undertaking that the Major Project entitled **"Optimization of Nozzle Location to Minimize Stresses on Horizontal Pressure Vessel"**submitted by me, towards the partial fulfillment of the requirements for the degree of Master of Technology in Mechanical Engineering (CAD/CAM) of Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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Place: NU, Ahmedabad.

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This is to certify that the Major Project entitled **"Optimization of Nozzle Location to Minimize Stresses on Horizontal Pressure Vessel"** submitted by **Patel Ketankumar V.(15MMCC19)**, towards the partial fulfillment of the requirements for the degree of Master in Technology in Mechanical Engineering(CAD/CAM) of Nirma University, Ahmedabad is the record of work carried out by him under my supervision and guidance. In my opinion, the submitted work has reached a level required for being accepted for examination. The results embodied in this Major Project Part-I, to the best of my knowledge, haven't been submitted to any other university or institution for award of any degree or diploma.

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Abstract

Pressure vessels are leak proof containers.They are having wide range of applications in several industries namely petrochemical,chemical,steel cement,etc.They are the important equipment for the process industry.

Nozzles or openings are necessary in the pressure vessels to satisfy certain requirements such as an inlet or outlet connections, manholes, vents and drains, etc. To incorporate a nozzle on the vessel wall it is supposed to remove some amount of material from the vessel. Then the stress distribution is not uniform.The distribution of stress in the juncture area and the rest will differ as nozzles cause a geometric discontinuity of the vessel wall. It results in to a development of stress concentration is created around the opening. The junction may fail due to these high stresses.

In the present work, the importance of the effect of the discontinuity is mentioned, codes related to design of vessels and its components are discussed, nozzle and vessel parameters are calculated using ASME code formulas. Available Software is used in the design of pressure vessel like the thickness of the shell and nozzle data. Different nozzle locations with and without reinforcement of nozzle for offset of 0 ,8, 16, 24 and 32 in. from the vertical center line at the central cross section with different inclination angles like 0°, 15°, 30° and 45° are modeled with Creo Parametric 2.0. Ansys workbench is used to analyze the models prepared in Creo Parametric 2.0 by importing then in the workbench environment by generating proper meshing, and applying boundary conditions.

Contents

Declaration ii

Undertaking for Originality of the work iii

Certificate iv

Acknowledgments v

Abstract vi

Table of Contents viii

List of Figures x

List of Tables xi

1 Introduction 1

1.1 Pressure Vessel[1] 1

1.2 Classification of Vessels[1] 1

1.2.1 Stresses develop in cylindrical Pressure Vessels due to internal pressure[2] 2

1.2.2 Circumference stresses[2] 2

1.2.3 Longitudinal Stresses[2] 3

1.2.4 Structural and Material Considerations[3] 4

1.3 Failure in Pressure Vessel[4] 5

1.3.1 Modes of Failures 5

1.3.2 Different Modes of Failure[4] 6

2 Literature Review 7

2.1 Previous Study Conducted 7

3 Pressure Vessel Design 13
3.1 Calculation of Shell Thickness and Maximum Allowable Pressure . . . 14
3.1.1 Cylindrical Shape Shell . . . 14
3.1.2 Hemispherical Head . . . 14
3.1.3 Elliptical Shape Head . . . 15
3.1.4 Torispherical Head . . . 15
3.1.5 Conical Section and Cone type Head . . . 16
3.2 Shell Thickness for Various Heads and Shell . . . 16
3.3 Concept of Offset and Angle of Nozzle . . . 17
3.3.1 Nozzle Offset Dimension L for Cylindrical Shells . . . 17
3.3.2 Nozzle Offset Dimension L for Heads . . . 17
3.4 Details of Material Consideration for the Design of the Pressure Vessel. 18
3.5 Summary . . . 19

4 Finite Element Modeling and Analysis of Pressure Vessel without Reinforced Nozzle 20
4.1 Introduction . . . 20
4.2 Finite Element Analysis Procedure in ANSYS . . . 20
4.2.1 Pre-Processing . . . 20
4.2.2 Solving the Model . . . 22
4.2.3 Post-Processing . . . 23
4.3 Assumptions for Finite Element Analysis of Pressure Vessel . . . 23
4.4 Pressure Vessel with Nozzle at 0 in. Offset from Center . . . 23
4.4.1 FEA of Pressure Vessel with Nozzle at 0 in. Offset and 0 Degree Inclination . . . 23
4.4.2 FEA Result Summary of Pressure Vessel with Nozzle at 0 in. Offset . . . 25
4.5 Pressure Vessel with Nozzle at 8 in. Offset from Center . . . 25
4.5.1 FEA of Pressure Vessel with Nozzle at 8 in. Offset at 0 Degree Inclination . . . 25
4.5.2 FEA result summary of Pressure vessel with nozzle at 8 in. offset . . . 27
4.6 Pressure Vessel with Nozzle at 16 in. Offset from Center . . . 27
4.6.1 FEA of Pressure vessel with nozzle at 16 in. offset at 0 degree inclination . . . 27
4.6.2 FEA Result Summary of Pressure Vessel with Nozzle at 16 in. Offset . . . 29
4.7 Pressure Vessel with Nozzle at 24 in. Offset from Center . . . 29

4.7.1 FEA of Pressure Vessel with Nozzle at 24 in. Offset at 0 Degree Inclination . . . 29
4.7.2 FEA Result Summary of Pressure Vessel with Nozzle at 24 in. Offset . . . 31
4.8 Pressure Vessel with Nozzle at 32 in. Offset from Center . . . 31
4.8.1 FEA of Pressure Vessel with Nozzle at 32 in. Offset at 0 Degree Inclination . . . 31
4.8.2 FEA Result Summary of Pressure Vessel with Nozzle at 32 in. Offset . . . 33

5 Analysis of Pressure Vessel with Reinforced Nozzle 34
5.1 Nozzle Reinforcement[4] . . . 34
5.2 Reinforcement of Openings[4] . . . 35
5.3 Reinforcement Calculation for Reinforced Nozzle Using a Commercially Available Software . . . 37
5.4 Finite Element Analysis of Pressure Vessel with Reinforced Nozzle . . . 37
5.5 Pressure Vessel with Reinforced Nozzle at 0 in. Offset from Center . . . 38
5.5.1 FEA of Pressure Vessel with Reinforced Nozzle at 0 in. Offset at 0 Degree Inclination . . . 38
5.5.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 0 in. Offset . . . 40
5.6 Pressure Vessel with Reinforced Nozzle at 8 in. Offset from Center . . . 40
5.6.1 FEA of Pressure Vessel with Reinforced Nozzle at 8 in. Offset at 0 Degree Inclination . . . 40
5.6.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 8 in. Offset . . . 42
5.7 Pressure Vessel with Reinforced Nozzle at 16 in. Offset from Center . . . 42
5.7.1 FEA of Pressure Vessel with Reinforced Nozzle at 16 in. Offset at 0 Degree Inclination . . . 42
5.7.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 16 in. Offset . . . 44
5.8 Pressure Vessel with Reinforced Nozzle at 24 in. Offset from Center . . . 44
5.8.1 FEA of Pressure Vessel with Reinforced Nozzle at 24 in. Offset at 0 Degree Inclination . . . 44
5.8.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 24 in. Offset . . . 46
5.9 Pressure Vessel with Reinforced Nozzle at 32 in. Offset from Center . . . 46

5.9.1 FEA of Pressure Vessel with Reinforced Nozzle at 32 in. Offset at 0 Degree Inclination 46
5.9.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 32 in. Offset 48

6 Conclusion and Future Scope 49
6.1 Results for Pressure Vessel without Reinforcement 49
6.2 Results for Pressure Vessel with Reinforcement 51
6.3 Future Scope 54

List of Figures

1.1 Modes of failure for cylindrical shell 2
1.2 Circumferential stresses 3
1.3 Longitudinal stress 4

3.1 Hemispherical head with cylindrical shell 14
3.2 Elliptical head with cylindrical shell 15
3.3 Conical head with cylindrical shell 16
3.4 Nozzles on cylindrical shell 17
3.5 Nozzles on head 17

4.1 Pressure vessel model 21
4.2 Pressure vessel model with fixed boundry 21
4.3 Pressure vessel model with pressure load condition 22
4.4 Meshed model of pressure vessel 22
4.5 Pressure vessel model with nozzle at 0 in. offset and 0 degree inclination 24
4.6 Stress profile of pressure vessel with nozzle at 0 in. offset and 0 degree inclination 24
4.7 Plot of Von-Misses stresses on pressure vessel without reinforcement nozzle at 0 in. offset 25
4.8 Pressure vessel with nozzle at 8 in. offset and 0 degree inclination . . 26
4.9 Stress profile of pressure vessel with nozzle at 8 in. offset and 0 degree inclination 26
4.10 Plot of Von-Misses stresses on pressure vessel without reinforcement nozzle at 8 in. offset 27
4.11 Pressure vessel with nozzle at 16 in. offset and 0 degree inclination . 28
4.12 Stress profile of pressure vessel with nozzle at 16 in. offset and 0 degree inclination 28
4.13 Plot of Von-Misses stresses on pressure vessel without reinforcement nozzle at 16 in. offset 29
4.14 Pressure vessel with nozzle at 24 in. offset at 0 degree inclination . . 30

4.15 Stress profile of pressure vessel with nozzle at 24 in. offset and 0 degree inclination 30
4.16 Plot of Von-Misses stresses on pressure vessel without reinforcement nozzle and 24 in. offset 31
4.17 Pressure vessel with nozzle at 32 in. offset and 0 degree inclination . 32
4.18 Stress profile of pressure vessel with nozzle at 32 in. offset and 0 degree inclination 32
4.19 Plot of Von-Misses stresses on pressure vessel without reinforced nozzle at 32 in. offset 33

5.1 Reinforcement opening 34
5.2 Nomenclature and formulas for reinforced opening 35
5.3 Variation in stress in region of a circular hole in (a) cylinder, (b) sphere subjected to internal pressure 36
5.4 Reinforcement boundaries for circular openings in cylindrical and spherical vessels 37
5.5 Meshed model of pressure vessel 38
5.6 Pressure vessel model with reinforced nozzle at 0 in. offset and 0 degree inclination 39
5.7 Stress profile of pressure vessel with reinforced nozzle at 8 in. offset and 0 degree inclination 39
5.8 Plot of Von-Misses stresses on pressure vessel with reinforcement nozzle at 0 in. offset. 40
5.9 Pressure vessel with reinforced nozzle at 8 in. offset and 0 degree inclination 41
5.10 Stress profile of pressure vessel with reinforced nozzle at 8 in. offset and 0 degree inclination 41
5.11 Plot of Von-Misses stresses on pressure vessel with reinforcement nozzle at 8 in. offset. 42
5.12 Pressure vessel with reinforced nozzle at 16 in. offset and 0 degree inclination 43
5.13 Stress profile of pressure vessel with reinforced nozzle at 16 in. offset and 0 degree inclination 43
5.14 Plot of Von-Misses stresses on pressure vessel with reinforcement nozzle at 16 in. offset 44
5.15 Pressure vessel with reinforced nozzle at 24 in. offset and 0 degree inclination 45

5.16 Stress profile of pressure vessel with reinforced nozzle at 24 in. offset and 0 degree inclination 45
5.17 Plot of Von-Misses stresses on pressure vessel with reinforcement nozzle at 24 in. offset 46
5.18 Pressure vessel with reinforced nozzle at 32 in. offset and 0 degree inclination 47
5.19 Stress profile of pressure vessel with reinforced nozzle at 32 in. offset and 0 degree inclination 47
5.20 Plot of Von-Misses stresses on pressure vessel with reinforced nozzle at 32 in. offset 48

6.1 Plot of Von-Mises stresses on pressure vessel without reinforcement at different offset distance by varying nozzle inclination angle 50
6.2 Plot of Von-Mises stresses on pressure vessel without reinforcement at different offset distance by varying nozzle inclination angle 51
6.3 Plot of Von-Mises stresses on pressure vessel with reinforcement at different offset distance by varying nozzle inclination angle 52
6.4 Plot of Von-Mises stresses on pressure vessel without reinforcement at different offset distance by varying nozzle inclination angle 53

List of Tables

3.1 Design data for the pressure vessel 13
3.2 Design parameter of pressure vessel 16
3.3 Design data 18

4.1 Result summary of pressure vessel with nozzle at 0 in. offset 25
4.2 Result summary of pressure vessel with nozzle at 8 in. offset. 27
4.3 Result summary of pressure vessel with nozzle at 16 in. offset 29
4.4 Result summary of pressure vessel with nozzle at 24 in. offset 31
4.5 Result summary of pressure vessel with nozzle and 32 in. offset . . . 33

5.1 Pad properties of nozzle with different offsets 37
5.2 Result summary of pressure vessel with reinforced nozzle at 0 in. offset 40
5.3 Result summary of pressure vessel with reinforced nozzle at 8 in. offset. 42
5.4 Result summary of pressure vessel with reinforced nozzle at 16 in. offset 44
5.5 Result summary of pressure vessel with reinforced nozzle at 24 in. offset 46
5.6 Result summary of pressure vessel with reinforced nozzle at 32 in. offset 48

6.1 Summary of Von-Mises stresses developed for a pressure vessel without reinforcement 49
6.2 Summary of Von-Mises stresses on pressure vessel with reinforcement 50
6.3 Summary of Von-Mises stresses on pressure vessel with reinforcement 51
6.4 Summary of Von-Mises stresses developed for a pressure vessel without reinforcement 52

Chapter 1

Introduction

1.1 Pressure Vessel[1]

If we want to store fluid with pressure, it is necessary to use pressure vessel. The fluid stored in the pressure vessel may change its state, for example vapors to water or react with any chemical components.The design of pressure vessel is very important because breakage or blast of the pressure vessel is very dangerous to human and property. The material selection is also play important role in the design of pressure vessel and protect from accidents. Different size and shape of pressure vessels are generally and repeatedly used in the industry to store liquids and gases under required pressure of pressure vessel.The materials are subjected to pressure and loading, so stresses generated in all directions. This stresses may be repeated and for a long time, so pressure vessel fails due to fatigue or creep. For the design of Pressure Vessels Codes and Standards like the ASME Boiler and Pressure Vessel Code or the BS 5500 Unfired Fusion Welded Pressure Vessels (BS 5500, 2004) are used.This type of codes and formula provide guideline for calculating the pressure vessel dimensions.

1.2 Classification of Vessels[1]

Vessels are mainly classified according to following:

⇒ According to Storage pressure

  - Atmospheric tanks (up to 0.5 psi)
  - Low pressure tanks (up to 15 psi)
  - Pressure Vessels (above 15 psi)

⇒ According to the diameter (d) to wall thickness (t) ratio

  - Thin cylinder ($d/t > 15$)

- Thick cylinder ($d/t < 15$)

⇒ According to Geometry of the vessel

- Cylindrical
- Spherical
- Combination of Cylinder or Sphere with Cones.

⇒ According to vessel end shape

- Flat head
- Convex head

1.2.1 Stresses develop in cylindrical Pressure Vessels due to internal pressure[2]

During analysis of stresses following assumptions are considered
1) Curvature effects on cylinder walls are neglected.
2) Tensile stresses are neglected on the wall.
Whenever internal pressure is generated in a cylindrical pressure vessel is fail in following two ways
1) It will be failed at the longitudinal section (i.e. Circumstantially) break the cylinder into two halves, as shown in Fig.1.1 (a).
2) It will be failed around the transverse section (i.e. Longitudinally) break the cylinder into two cylindrical shells, as shown in Fig. 1.1 (b).
So the wall of a cylindrical shell withstand to an internal pressure has to tensile stresses of the following two types:
(a) Circumferential or hoop stress
(b) Longitudinal stress.

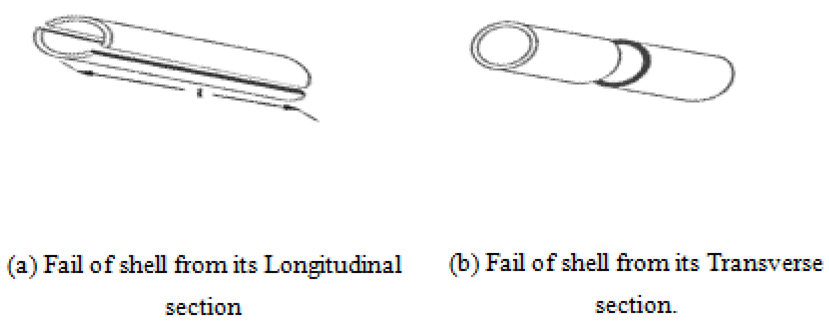
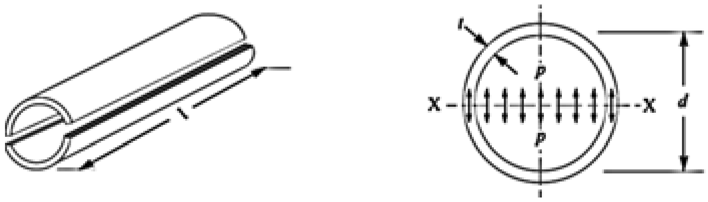


Figure 1.1: Modes of failure for cylindrical shell

1.2.2 Circumference stresses[2]

In following case of thin cylindrical shell is subjected to an internal pressure as shown in Fig.1.2 (a) and (b).Circumference (hoop) stresses is defined as tensile stresses an act in the direction tangential to the circumference.In different way we can say that,this is a tensile stress on the longitudinal section.



(a) Shell view (b) Cross-Sectional of Shell

Figure 1.2: Circumferential stresses

p =Cylinder shell Internal pressure,
d =Cylindrical shell Internal diameter
l =Cylindrical shell Length
t =Cylindrical shell Thickness
σt_1 =Hoop stresses in the cylindrical shell
The total force acting on a longitudinal section (i.e. Along the diameter$X - X$) of the shell
= Intensity of pressure $\times$ Projected area $=p \times d \times l$.............................. (i)

The total resisting force acting on the cylinder walls $=\sigma t_1 \times 2t \times l$............. (ii)
According to the equations (i) and (ii), we have

$$\sigma t_1 \times 2t \times l = p \times d \times l$$

$$\sigma t_1 = \frac{p*d}{2t}$$

1.2.3 Longitudinal Stresses[2]

In following case of thin cylindrical shell subjected to an internal pressure as shown in Fig. 1.3 (a) and (b).

Longitudinal stresses are defined as in the cylindrical shell, tensile stresses acting in the direction of the shell axis. In the different way, it is tensile stress acting on the transverse section $Y - Y$.

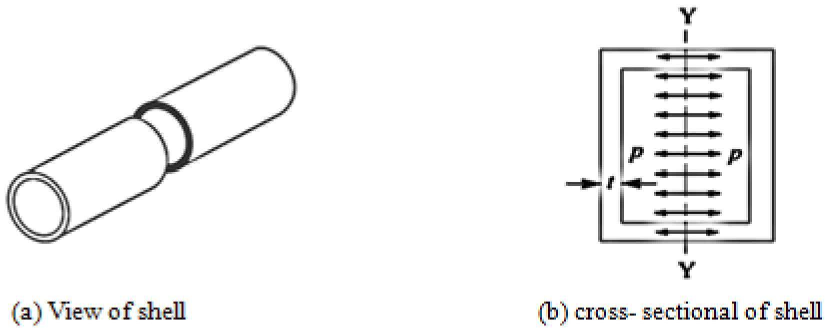


Figure 1.3: Longitudinal stress

Let,σt_2=Longitudinal stress.
In above case, the total force acting on the transverse section (Y-Y)
= Intensity of pressure $\times$ Cross-sectional area
$=p * \frac{\pi}{4} d^2$ .... (i)

And Total resisting force = $\sigma t_2 \times \pi d.t$ .... (ii)

From comparing above equations number (i) and (ii), we have
$\sigma t_2 \times \pi.d.t = p * \frac{\pi}{4} d^2$

$\sigma t_2 = \frac{p*d}{4t}$

1.2.4 Structural and Material Considerations[3]

The material selection is very crucial in case of pressure vessel design because a continue and several years of usage of a pressure vessel in a worst,dusty and corrosive environment. Pressure vessels are required to operate at a temperature range from as 600°C to as –20 degree °C, with pressures as high as 140 MPa because it is applicable in variety of purpose like power generation, nuclear and chemical plant and industrial processing and storage require etc.Some pressure vessels operate in corrosive environments or non-corrosive environments. Because of high and low temperature, other loading conditions and for other mechanical property requirements, material selection is very important. On other side ease of fabrication and manufacturing, less maintenance and for cost reduction purposes material selection is an important criteria.

General Types of Materials:

⇒ Carbon steel – strength & moderate corrosion resistance

⇒ Low-alloy steels – strength at high temperatures

⇒ Stainless steels – corrosion resistance

⇒ Nickel alloys – corrosion resistance

⇒ Copper alloys – sea water resistance

⇒ Aluminum – light, low temperature toughness

⇒ Titanium – sea water, chemical resistance

⇒ Nonmetallic – aqueous corrosion & chemicals

The major properties required for pressure vessel:

⇒ Yield strength

⇒ Ultimate strength

⇒ Fracture toughness

⇒ Corrosion resistance

So, according to the above discussion, it is clear that it is necessary to have theoretical and experimental knowledge about stress distribution, its operating pressure, material behavior at operating temperature and other factors so there are need of analytical method to design the pressure vessel.

1.3 Failure in Pressure Vessel[4]

1.3.1 Modes of Failures

A. MATERIAL

It is due to improper selection of materials and its grade and major micro and macro level defects in material.

B. DESIGN

For end users provide wrong data and may be modeling and analysis technique is improper.

C. FABRICATION

There is a poor quality of fabrication and manufacturing may be casting defects or joining or welding defect.

D. SERVICE

Customers do not have enough knowledge about operation, maintenance service condition. Because of unskilled labors, rules and regulations are not follow, so slowly and gradually service life is decreased .

1.3.2 Different Modes of Failure[4]

⇒ Elastic deformation

- Vessel can go under elastic deformation and buckled.

⇒ Brittle fracture

- This type of fracture occurs when vessel operates at low temperature but low Carbon steel materials have brittle fracture at intermediate temperature.

⇒ Excessive plastic deformation

- Up to a certain level of plastic deformation is recommended, but excessive plastic deformation causes fracture of vessels..

⇒ Stress rupture

- There are two possibilities for stress rupture (1) Fatigue failure-This due to cyclic loading. (2) Creep failure-This is due to time dependent phenomena.

⇒ High strain

- Low cycle fatigue is strain-governed and occurs mainly in lower-strength high-ductile materials.

⇒ Stress corrosion

- This is due to stored material like chlorine, caustic and ammonia etc.

⇒ Corrosion fatigue

- This occurs because of stress corrosion and fatigue loading observed simultaneously. This is propagating crack on a surface.

Chapter 2

Literature Review

A literature review was conducted to consider previous work related to pressure vessels. The technical works reviewed are as follows.

2.1 Previous Study Conducted

Here some papers are discussed as a part of literature review.

Josip Kacmarcik, et al. [5] concluded that comparison of result between the stress concentration factor resulted with two different method, here two different method strain gauge with experimental set-up and finite element analysis with ABAQUS software are used for two different nozzle geometries investigation, here two Sinle cell function defined by maximum principal stress and maximum von-misses stress are calculated by strain gauge measurement and compared with ABAQUS software, in this paper nozzle external radius are different C1 Nozzle has higher radius then C2 nozzle, but both nozzle have same thickness of vessel wall and external radius of a vessel, in this paper only 1/8 of the vessel part and ¼ of nozzle part is modeled because it is possible to defined three symmetry planes and here as a mesh generation 3D tetragonal elements are implemented, stress concentration factor is obtained by the value of stress(principal and von misses) obtained via Finite element modeling, analysis and strain gauge measurement, when compared both method it is shown that the maximum deviation of 15.5% is acceptable for engineering application of stress concentration factor and Finite element modeling analysis is very reliable enough for determining stress concentration factor in pressure vessel design and this research also show advantages of finite element method analysis in possibility to defining stresses on vessel inner side that can be greater than outer stresses which is very difficult for strain gauge measurement.

V.N. Skopinsky, et al. [**6, 7**] worked on modeling and stress analysis of nozzle connection in Ellipsoidal head of pressure vessel under external loading, in this paper he used Timoshenko shell theory and the finite element method, the effect of stress concentration in external loading has more effect than in the internal pressure, there is an increase of the maximum stress for shell in the interaction region even at the small level of nominal stress, non-radial and offset connection have non-uniform distribution of stress on the interaction curve between the nozzle and the head, the influence of angular parameter α for non-radial nozzle connection is shown in this paper, a decrease of maximum effective stress as an angle α increase is more significant for non-central connection, and in case of torsion moment loading, the angle affects the stress in the opposite manner, the stress in the shell increase as alpha angle increase.

J. Fang, et al. [8] worked on a the study of strength behavior on cylindrical shell interaction with and without pad reinforcement under out-of-plane moment loading on nozzle, three pairs of full-scale test vessel with different mean diameter of nozzle to mean diameter of cylindrical vessel ratio were designed and fabricated for testing and analysis ,the material of the cylinder, reinforcement pad and the nozzle are low carbon steel, result from this research indicate that the maximum elastic stress and stress ratio are decreased by pad reinforcement, they found that in test reduction rate is 20-60% and in finite element analysis reduction rate is 28-59% and its rate of reduction depend upon structure and dimension of the vessel for example D/d ratio, and result also it showed that the plastic limit of nozzle in cylinder vessel is increased by pad reinforcement, generally rate of increase is about 40-70% from test and its larger than 40% from finite element analysis, so the conclusion given from the result that the reinforcement structure are useful under static external load on nozzle.

Pravin Naral, et al. [**9**] analyzed work on structural analysis of nozzle attachment on pressure vessel design, they said if the nozzle is kept in peak of the dished end it do not disturb the symmetry of the vessel, but if it is placed on the placed on the periphery of the vessel, it may be disturbed the symmetry of the vessel. Size, diameter, angle, etc. of nozzle connection may significantly vary even in one horizontal pressure vessel, these nozzle cause geometric discontinuity of vessel wall, so a stress created around a opening, the junction may fail due to high stress ,so detailed analysis is must be required, in this paper conduct a study analysis, what will be the effect of the nozzle angle and increase number of nozzle on the periphery of pressure vessel until the symmetry is achieved, and find out optimize angle such that the stress are maintained within limits. in this paper first one nozzle placed on top on shell and calculated stresses with finite element analysis, then two nozzle placed with angle 60 degree from each other, then again two nozzle placed at angle 90 degree from each other, then also again two nozzle placed at angle 180 degree from each other ,then three nozzle placed at angle 60 degree from each other, then again three nozzle placed at angle 90 degree from each other, then four nozzle placed at angle 60 degree and again four nozzle placed at 90 degree from each other and calculated stress from ANSYS software ,from this study they found the result that peak stresses for symmetrical nozzle attachment is lowest than the others and stress increment factor for symmetric nozzle attachment is lower than other, here the stress value is minimum at two nozzle which is placed at angle 180 degree and four nozzle placed at angle 90 degree from each other, this state that the symmetry nozzle attachment had always lower stress than others.

James J Xu, et al. [10] worked on local pressure stress on lateral pipe-nozzle with various angle of interaction, this paper report variation of local pressure stress factor at the junction of pipe-nozzle when its angle varies from 90 to 30 degree, the circumferential and longitudinal stress at four symmetric points around the pipe-nozzle junction are plotted as function of an angle, the ALGOR software was employed to model for the true pipe-nozzle geometry, the numeric stress result come from parameters beta and gamma which are the nozzle mean radius and pipe thickness, at angle 90 degree at this angle result had low value local stress, these stress increase as angle of interaction is decrease from 90 degree and stress value more decrease when angle is decrease from 45 degree, the inside crotch point B has worst circumferential stress value, and concluded that angle 90 degree local pressure stress are same at point A and B as same as point C and D due to symmetry. And it had low stress value than other angle.

Amran Ayob [11]worked on stress analysis of torispherical shell with radial nozzle, in this work, experimental reading was taken with help of 0.0625-in. foil strain gauge which was bonded to the external and internal surface of the shell, the model was instrumented with thirty nine pairs of 0.0625-in. foil strain gauge, these gauge have been locate between S=-0.1 to S=0.5 in the meridional direction. The experimental set up result used here is the part of test engineer carried out by drabbles to determine the shakedown behavior of a torispherical vessel with nozzle,. There are three interacting geometric location which could influence stress field, the maximum stress could occur any of sphere-nozzle, sphere-knuckle and cylinder-knuckle junction the graph of the elastic stress factor distribution along meridional plane due to four load case shown in this paper, the crotch corner and the weld-crown region are the highest stress area with ESF approximately 2.

V. N. Skopinsky [6] worked on stresses in elliptical pressure vessel heads with non-central nozzle, an objective of the work is investigation of shell intersection problem, the shell theory and finite element method are used for stress analysis of nozzle connections in ellipsoidal heads of the pressure vessel, here nozzle is considerably displaced on ellipsoidal head from head axis is considered in this paper, the feature of numerical procedure, structural modeling of nozzle-head shell intersections and SAIS special-purpose computer program are discussed. The result of stress analysis and parametric study of elliptical vessel head with a non-central nozzle under internal pressure loading are presented, in many practical design, the nozzle was put on at a relatively large distance from the head axis. this stress analysis result better understanding of this poorly investigated problem and give the possibility of achieving a most reliable design of nozzle connections on the horizontal pressure vessel heads, also the SAIS program can be used for design optimization purpose e.g. nozzle location finding.

P balicevic [**12**] analyzed work on ANLYTICAL and NUMERICAL solution of inner forces by cylindrical horizontal pressure vessel with semi-elliptical heads, in this paper, solution for internal forces and displacement in the thin-walled cylindrical pressure vessel with ellipsoidal head using general theory of thin walled shell of resolution have been proposed, distribution of the forces and displacement in thin walled shell are given in mathematical form, finite element analysis of the cylindrical vessel with semi-elliptical head has been done by using ANSYS 10 code for to confirms analytical solution, here ellipsoidal head model made as ax-symmetric problem to avoid bending effect on the contact between heads and cylinders and author concluded principal stresses calculated analytically are very close to the finite element result(the difference is less than 3%.

M F hsieh [**13**] worked on the torispherical head with knuckle region on nozzle, in this paper limit load interaction plot for pressure vs. nozzle axial force, in-plane moment, out-of-plane moment and for in-plane moment versus out-of-plane moment are also present, here six model included with nozzle offset location nozzle offset/vessel external diameter in present study, model 1 is the ax symmetric case with nozzle located in the center of the crown, the model 3 offset the outside weld location at knuckle junction and in this work FE model was created with using PATRAN mesh generation program and ABAQUS program used for stress analysis, they concluded that the nozzle was very influence on the limit pressure of the head, but when it was located in the knuckle region of the head, for external load applied to the nozzle, the effect of increasing the offset is to increase the loads.

B.S.Thakkar [**14**] did a case study and put efforts to design the pressure vessel using ASME codes & standards to legalize the design. The performance of a pressure vessel under pressure can be determined by conducting a series of tests to the relevant ASME standard in future scope they have mentioned Design of pressure vessel in commercially available software can be accrued. Further FEA analysis can be done to verify the above design procedure, they concluded that the design of the pressure vessel is more of a selection procedure, selection of its components to be more precise rather designing of every component, pressure vessel components are selected on the basis of available ASME standard and the manufactures also follow the ASME standard while manufacturing the components so that leaves designer free from designing the components. This aspect of design greatly reduces the development time of the new pressure vessel, it also allows the designer to keep free from multiple prototype for pressure vessel before finalizing the design, here standard part are used so it reduces time for replacement so less overall cost.

Shaik Abdul Lathuef [15] discussed some of the potential unintended consequences related to Governing Thickness of shell as per ASME. Here have a scope to change the code values by taking the minimum governing thickness of pressure vessel shell to the desired requirements and also relocate of nozzle location to minimize the stresses in the shell. In this paper nozzle located at five places and analysis with ANSYS here nozzle locates at shell left end, at the shell middle, at the shell right end, at dished end of both side and calculate the stress. And they found from the result that the stress would be Minimum at the dished end with hillside orientation. A low value of the factor of safety results in economy of material this will lead to thinner and more flexible and economical vessels. Here we evaluated the stress in the vessel by Zick analysis approach.

Vikram V. Mane [16] done their work on stress analysis of Ellipsoidal head pressure vessel the in finite element analysis and experimental work, they used electrical strain gauges for strain measurement and compared results with ANSYS software, and they found the results of the stress analysis by classical methods are more than the actual stresses measured by strain gauges and less than the finite element analysis.

Bandarupalli Praneeth [17] compared the theoretical values and ANSYS value for both solid wall and multi layer pressure vessels. And they concluded that multi layered pressure vessels are superior for high pressures and high temperature operating conditions over the conventional mono block pressure vessel, here various parameter of solid pressure vessel are designed and checked according to American society of mechanical engineering (A.S.M.E) Section VII Division I, same way multi layer pressure vessel designed and checked according to A.S.M.E section VII Division I, stresses produced in solid and multi layer pressure vessel is analyzed by ANSYS software, Theoretical calculated values by using Different formulas are very close to that of the values obtained from ANSYS analysis is suitable for multi layer pressure vessels.

Drazan kozak [18] presented work on overloading effect on the carrying capacity of a cylindrical Tank with torispherical heads for the underground storage of petrol, horizontal cylindrical double skin steel tank with torispherical heads for the underground storage of petrol has been manufactured, Before exploitation it has to be tested with a pressure of 2 bars according EN 12285-1 norm. During the pressurization uncontrolled pressure increased.The Effects on this overloading have been analyzed by using finite element method.

M. Pradeep Kumar [19] presented work on Design and Implementation of Circular Cross Sectional Pressure Vessel Using Pro-E and ANSYS, EW system frame assembly is a leak proof and contains high pressure to hold precious electronic at a pressure different from the ambient pressure, they concluded the stresses developed in the circular cross section with hemispherical end caps are very less as compared to a rectangle cross sectional vessel which is used in the submarine EW system. Also in the circular cross section, the stresses and deflections are minimized. Design of pressure vessel done by ASME code section VII and analysis is done by ANSYS software.

Nishant M. Tandel [**20**] had presented work on pressure vessel design and analysis, this paper deal with vessel are subjected to various applied forces acting in combination with internal or external pressure and some design principle, design of pressure vessel is governed by ASME pressure vessel code, design of different pressure vessel concerned with element such as shell, dish end, operating man hole, support leg, based on standard and code and evaluation of shell and dish end analyzed by means of analysis, and this paper they concluded that finite element analysis is an extremely powerful tool foe pressure vessel and also concluded the design method to be used in pressure vessel are depend upon stresses and internal or external pressure.

Modi A J [**9**] concluded the radial stresses for the case of a hemispherical head pressure vessel is low compared to other types of head. The study of the comparative structural behavior of different types of geometry of pressure vessel, here three model was set without any types of nozzle and any support, the head is in internal uniform pressure, the analytical and finite element method used need to find stresses in pressure vessel, the aim is finding an optimum head for a specific parameter with finite element analysis of thin cylindrical pressure vessel, here three types of geometry consider for an example, Hemisphere, flat and ellipsoidal and computation result compare with finite element analysis.

M J Mungla [**21**] worked on the design and analysis of various components of pressure vessels like shell, heads, flanges, and nozzle and support structures along using ASME code, shell, head, nozzle and skirt support dimension are based on various design condition it is under internal pressure and an external pressure along with wind load / seismic effect and dimension arrived by this effect is compared with an allowable limit within allowable limits. Stress analysis of the flange has been carried out as per Appendix 3 of ASME Section VIII Div.2, it has been determined that induced stresses (longitudinal, tangential and radial) are within the corresponding given allowable limits. Design of the base ring and skirt sections has not been under ASME code and their dimensions were calculated with common design principles. Stress analysis of these components has been done with combined loads, and this work it was found that,the stress produced due to combined loads are within its allowable limit.

Chapter 3

Pressure Vessel Design

The design data considered for a pressure vessel taken as part of this work is given in the table 3.1.

Table 3.1: Design data for the pressure vessel

Sr. No	Description	Shell Side	Head Side
1	Design Internal Pressure	30.29 bar	30.29 bar
2	Temperature for Internal Pressure	50°C	50°C
3	Material Specification	SA-516 70	SA-516 70
4	Tensile strength of material	485 Mpa	485 Mpa
5	Yield Stress at Temperature	262 Mpa	262 Mpa
6	Joint efficiency for Shell/head Joint	1.000	1.000
7	Internal Corrosion Allowance	0.0000	0.000
8	Inside Diameter of Cylindrical Shell/head	2133 mm	2133 mm
9	Aspect ratio of elliptical head	50.8 mm	
10	Inside crown radius of Torisphercial head	1706.88 mm	
11	Inside knuckle Radius of Tori spherical Head	213.36 mm	
12	Length of Cylindrical Shell	5791.2 mm	

3.1 Calculation of Shell Thickness and Maximum Allowable Pressure

The calculation of different types of heads with cylindrical shell is carried out using commercially available software and same is presented here.

3.1.1 Cylindrical Shape Shell

Thickness of shell Due to Internal Pressure:

t $=\frac{PR}{(SE-0.6P)}$

$=\frac{439.46*2}{20000*1-0.6*439.36}$

$= 0.935 in.$

MAWP means maximum all working pressure at given thickness

p $=\frac{SEt}{R+0.6t}$

$=\frac{20000*1*0.9325}{42+0.6*0.9325}$

$=439.46 psi$

3.1.2 Hemispherical Head

Because of Internal Pressure Thickness of hemispherical head

t $=\frac{PR}{(2SE-0.2P)}$ in.

$=\frac{439.46*42}{2*20000*1-0.2*439.46}$

$=0.4624 in.$

Max, allowable working pressure for head: (MAWP)

$p=\frac{2SEt}{R+0.2t}$

$=\frac{2*20000*1*0.4625}{42+0.2*0.4625}$

$=439.50 psi$

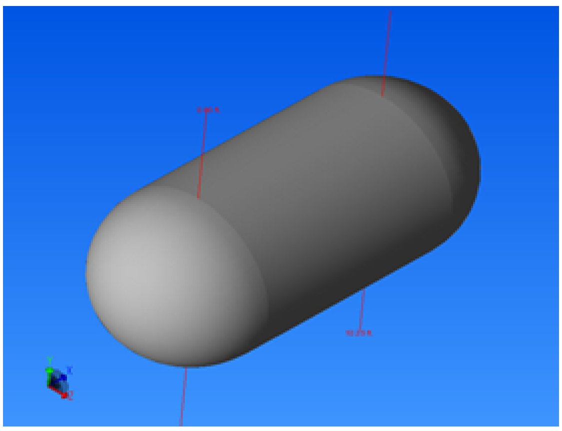


Figure 3.1: Hemispherical head with cylindrical shell

3.1.3 Elliptical Shape Head

Because of internal pressure thickness of elliptical head

t $=\frac{PD}{(2SE-0.2P)}$ in.

$=\frac{439.46*84}{2*20000*1-0.2*439.46}$

$= 0.9249in.$

Maximum allowable working pressure ellipsoidal head, (MAWP)

p $=\frac{2SEt}{D+0.2t}$

$=\frac{2*20000*1*0.9249}{84+0.2*42}$

$= 439.46psi$

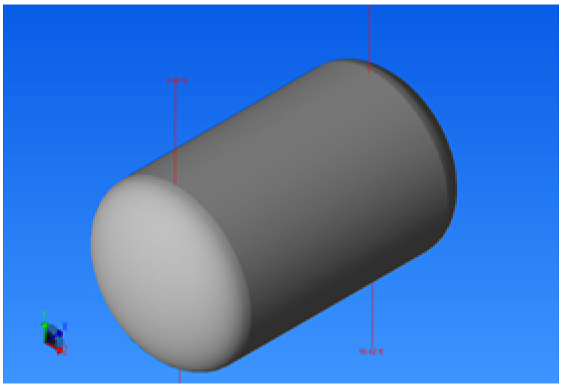


Figure 3.2: Elliptical head with cylindrical shell

3.1.4 Torispherical Head

Inside radius (L) is $0.8Di = 0.8 \times 84 = 67.200in.$
Knuckle radius (ri) is $0.1Di = 0.1 \times 84 = 8.4in.$
Therefore, inside radius (L) is $0.8Di = 0.8 \times 84 = 67.200in.$
Knuckle radius (ri) is$0.1Di = 0.1 \times 84 = 8.4in.$

Therefore, M factor $= \frac{1}{4}\left(3+\sqrt{\frac{L}{ri}}\right)$

$=\frac{1}{4}\left(3+\sqrt{\frac{67.200}{8.4}}\right)$

$=1.4571in.$

Thickness of head,

t $=\frac{PLM}{(2SE-0.2P)}in.$

$=\frac{439.46*67200*1.45}{2*20000*1-0.2*439.46}$

$= 1.0782in.$

MAWP means maximum all working pressure at given thickness

p $=\frac{2SEt}{LM+0.2t}$

$=\frac{2*20000*1*1.0782}{67.200+0.2*42}$

$=566.22psi$

3.1.5 Conical Section and Cone type Head

When half apex angle α = 30 degree is considered.

Than thickness of head

t=$\frac{PD}{(2cos\alpha(SE-0.6P)}$

=$\frac{439.46*84}{(2cos30(20000*1-0.6*438.46)}$

= $1.0564in.$

Maximum allowable working pressure at given thickness (MAWP)

P=$\frac{(2SEtcos\alpha)}{(D+1.2tcos\alpha)}$

=$\frac{2*20000*1*1.056*cos30}{(84+1.2*1.056*cos30)}$

= $448.97psi$

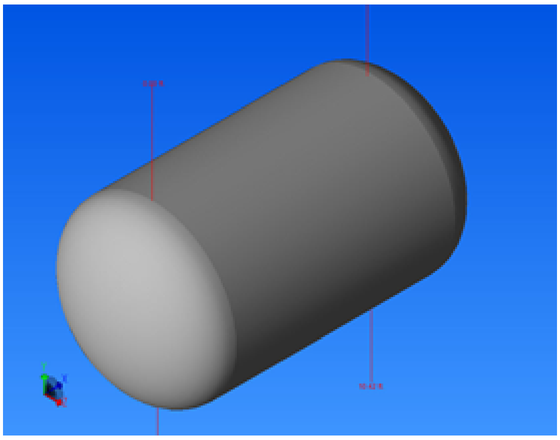


Figure 3.3: Conical head with cylindrical shell

3.2 Shell Thickness for Various Heads and Shell

Table 3.2: Design parameter of pressure vessel

	Required Thickness by Commercially available software (inch)
Cylindrical shell	0.94
Elliptical head	0.94
Hemispherical head	0.4624
Torispherical head	1.0782
Conical head	1.0564

3.3 Concept of Offset and Angle of Nozzle

The concept of offset and angle of nozzle is being explained here by considering horizontal vessel and looking it from the left head.The similar situation is shown in the figure 3.4.The angle of nozzle is the angle around the cylinder from the reference start point which is located at the top of the cylinder and nozzle axis.The angle of nozzle is denoted as ' α'. Thenozzle axis might not pass through vessel axis, if it is offset.

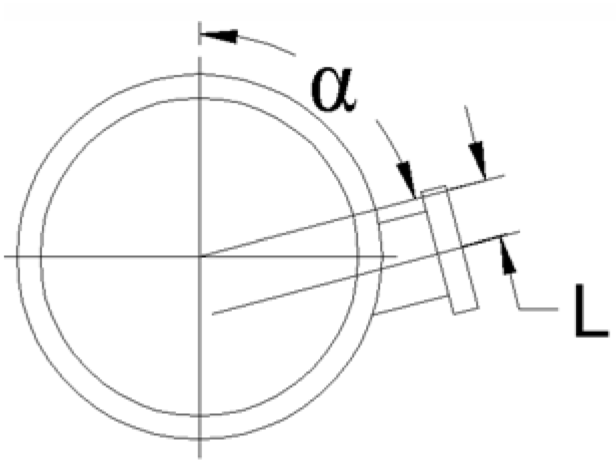


Figure 3.4: Nozzles on cylindrical shell

3.3.1 Nozzle Offset Dimension L for Cylindrical Shells

Looking at the illustration above, the offset dimension is the ‘L’ dimension, the distance from the vessel axis to the nozzle axis.

3.3.2 Nozzle Offset Dimension L for Heads

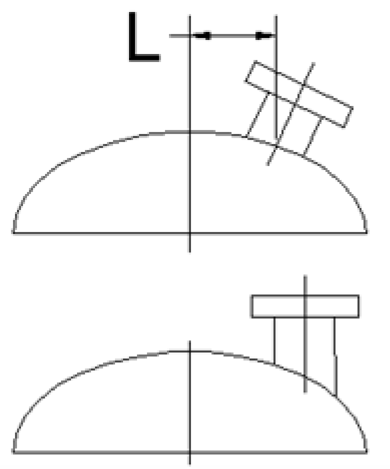


Figure 3.5: Nozzles on head

The Offset Dimension L is the radial distance of the nozzle from the vessel axis. It is taken from the center line of the vessel to where the axis of the nozzle cuts the outside surface of the head. This distance applies to both radial nozzles and hillside nozzles (figure 3.5).Care must be taken to ensure the outside diameter and their reinforcement pads are reasonably placed.

3.4 Details of Material Consideration for the Design of the Pressure Vessel.

The details of the materials considered for the design of a pressure vessel is given in the table 3.3.The material details is very useful for the conventional design of the pressure vessel.Using ASME codes and design carried out using commercially available software.

Table 3.3: Design data

Nozzle Flange Material	**SA-516 70**
Flange Type for Nozzle	Weld Neck Flange
Corrosion Allowance for Nozzle	0.0000 mm.
Shell Seam Joint Efficiency at Nozzle	1.00
Nozzle Neck Joint Efficiency	1.00
Projection of Nozzle from outside	304.79 mm.
In between Nozzle and Pad/Shell size of weld leg	19.04
In between Nozzle and Vessel Groove weld depth	12.7 mm.
Pad Material	**SA-516 70**
Pad diameter	60959 mm.
Pad thickness	50.8 mm.
In between Pad and Shell weld leg size.	19.04 mm.
In between Pad and Nozzle Groove weld depth	19.049 mm.
Reinforcing Pad Width	101.6 mm.
Class of attached Flange	300
Grade of attached Flange	GR 1.2

3.5 Summary

The shell thickness and maximum allowable working pressure for various heads have been calculated using commercially available software.Looking to the result we can conclude that both in the standard elliptical head and the cylindrical shell. The calculated head thickness is almost same.It is better to use the standard elliptical head rather than the cylindrical shell from manufacturing point of view.

Chapter 4

Finite Element Modeling and Analysis of Pressure Vessel without Reinforced Nozzle

4.1 Introduction

Finite Element Analysis (FEA) is a computer-based numerical technique for calculating the strength and behavior of engineering structures. It can be used to calculate deflection, stress, vibration, buckling behavior and many other phenomena. It can be used to analyze either small or large-scale deflection under loading or applied displacement.In the finite element method, a structure is broken down into many small simple blocks or elements. The behavior of an individual element can be described with a relatively simple set of equations. Just as the set of elements would be joined together to build the whole structure, the equations describing the behaviors of the individual elements are joined into an extremely large set of equations that describe the behavior of the whole structure. The computer can solve this large set of simultaneous equations. From the solution, the computer extracts the behavior of the individual elements. From this, it can get the stress and deflection of all the parts of the structure. The stresses will be compared to allowable values of stress for the materials to be used, to see if the structure is strong enough.

4.2 Finite Element Analysis Procedure in ANSYS

4.2.1 Pre-Processing

Pre-processing comprises of building, meshing and loading the model created.
Define Type of Analysis
ANSYS work bench provide wide verity of analysis for real life problem for mechanical and other engineering problems. Static Structural analysis is used for solving

current problem.

Define Engineering Data for Analysis

The material that is considered for the shell as well as nozzle is SA 516 GRADE 70; it is having mechanical properties like young's modulus of $2.0e^5$ Mpa and poison's ratio of 0.29.

Define or Import 3D Model

Pressure vessel models of different condition are modeled with application of PRO-Engineer. The models are exported as STEP file with solid as option. Same models are imported into ANSYS Workbench Environment.

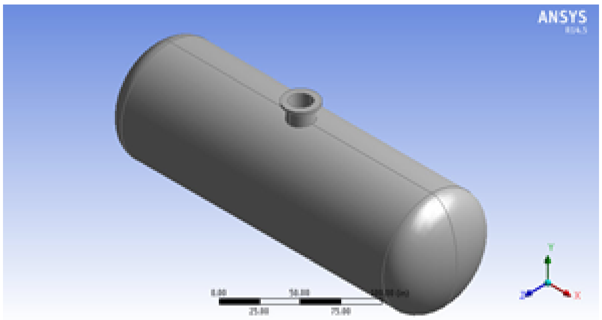


Figure 4.1: Pressure vessel model

Define Boundary Condition for Analysis

All the degrees of freedom of the pressure vessel are arrested at the right side edges at shell and head joint location for all models of pressure vessel under study throughout the thesis.

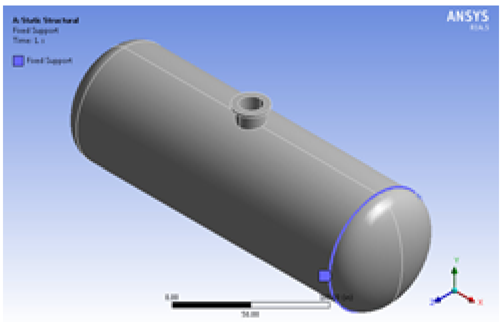


Figure 4.2: Pressure vessel model with fixed boundry

The magnitude of the pressure considered for the nozzle is 439.46 psi at all internal faces.

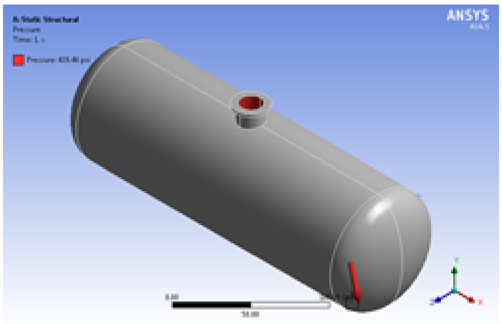


Figure 4.3: Pressure vessel model with pressure load condition

ANSYS offers a complete set of tools for automatic mesh generation, including mapped mesh generation and free mesh generation. Both mapped meshing and free meshing can access geometric information in the form of point, curves and surfaces. The model is meshed with10-noded tetrahedral (Tet-10) solid elements.

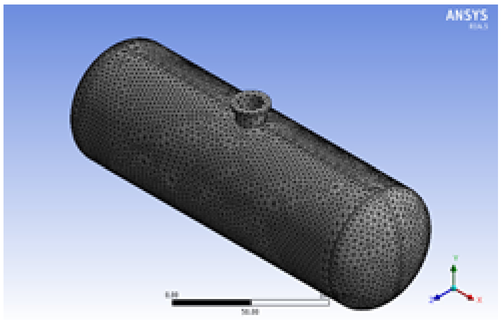


Figure 4.4: Meshed model of pressure vessel

Mesh Statics:
Type of Element : Tetrahedrons
Number of Nodes : 76612
Number of Elements : 38346

4.2.2 Solving the Model

With all parts of the model defined, nodes, element, restraints and loads, the analysis part of the model is ready to begin. The system can determine the values of stresses, deflection, temperature, pressure and vibration.
An analysis requires the following information.

⇒ Nodal points

⇒ Element connecting the nodal points

⇒ Material and its physical properties

⇒ Boundary conditions, which consists of loads and constraints

⇒ Analysis options: how the problem will be evaluated

4.2.3 Post-Processing

The post-processing task displays and studies the result of an analysis, which exists in the model as analysis data sets. This task can generate displays of stress contours, deformed geometry etc.

4.3 Assumptions for Finite Element Analysis of Pressure Vessel

⇒ Analysis type taken is static structural while neglecting effect of loading and boundary condition with time.

⇒ Model is simplified as no saddle support is considered while modeling the pressure vessel.

⇒ Both Shell and Head thickness is approximated and considered as 1 in..

⇒ The model is considered with single vertical outlet nozzle vertically at center of cross section of vessel at center of shell.

⇒ Vertical nozzle at center considered as reference condition with 0 in. offset distance and as base condition 0 ˚ inclination with vertical center line.

⇒ Only internal pressure is consider as load while neglecting all External loads.

⇒ One side shell to head joining edges is considered as fixed boundary condition.

4.4 Pressure Vessel with Nozzle at 0 in. Offset from Center

The model is considered with single vertical nozzle which is at center so offset distance is 0 in. and also initially as base condition inclination is also 0 ˚ .While exploring other design offset is considered from center of cross section at center of shell and vertical line is considered as reference for inclination.

4.4.1 FEA of Pressure Vessel with Nozzle at 0 in. Offset and 0 Degree Inclination

Structural analysis of Pressure vessel with nozzle at 0 in. offset and 0 degree inclination is carried out with 3D solid model shown in figure 4.5 to evaluate and compare the results with other option of inclination. Structural boundary conditions and loading are kept same as reference condition.

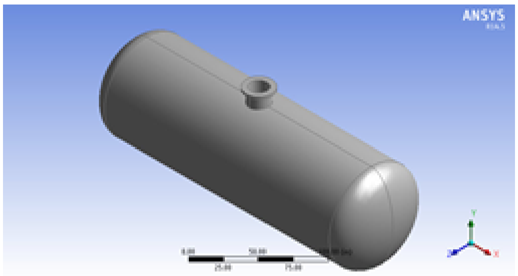
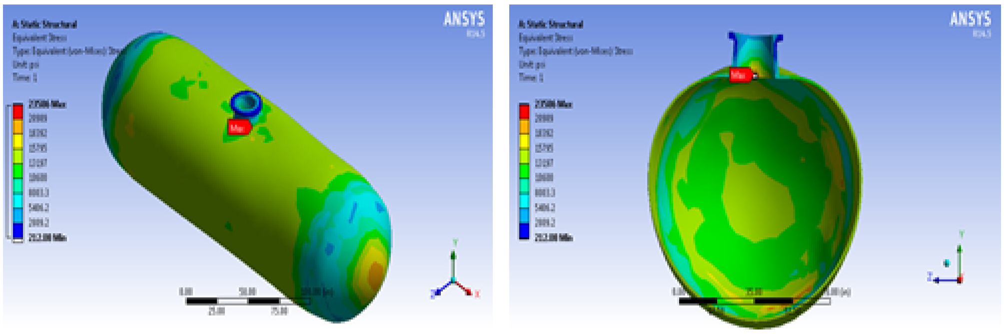


Figure 4.5: Pressure vessel model with nozzle at 0 in. offset and 0 degree inclination

Result summary
Maximum stress on nozzle: 23586 psi
Minimum stress on vessel: 212.08 psi
Maximum deformation on vessel: 0.099313 in.



(a) (b)

Figure 4.6: Stress profile of pressure vessel with nozzle at 0 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 0 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 4.5 and figure 4.6, now same way another structural analysis of pressure vessel with nozzle at 0 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions, and the result is shown in given table 4.1.

4.4.2 FEA Result Summary of Pressure Vessel with Nozzle at 0 in. Offset

Table 4.1: Result summary of pressure vessel with nozzle at 0 in. offset

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	23586	212.08	0.099313	0
15	26263	560.55	0.099418	0
30	28644	422	0.099726	0
45	24308	384.02	0.099556	0

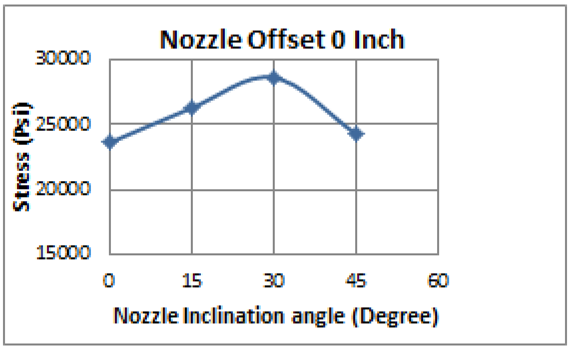


Figure 4.7: Plot of Von-Misses stresses on pressure vessel without reinforcement nozzle at 0 in. offset

4.5 Pressure Vessel with Nozzle at 8 in. Offset from Center

The model is prepared with single nozzle with offset distance is 8 in. and a base inclination of 0 $^\circ$.

4.5.1 FEA of Pressure Vessel with Nozzle at 8 in. Offset at 0 Degree Inclination

Structural analysis of pressure vessel with nozzle at 8 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 4.8.The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition.

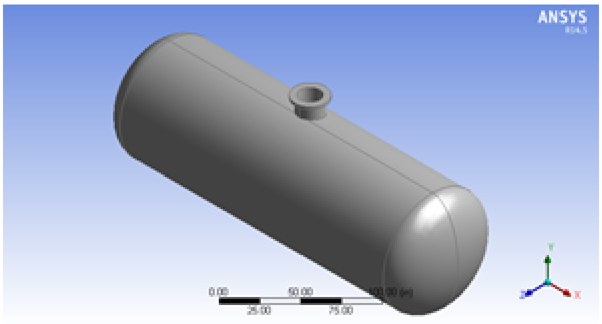


Figure 4.8: Pressure vessel with nozzle at 8 in. offset and 0 degree inclination

Result Summary
Maximum stress on nozzle: 29743 psi
Minimum stress on vessel: 369.78 psi
Maximum deformation on vessel: 0.0996 in.
The stress profiles are as shown in the figure 4.8 and 4.9

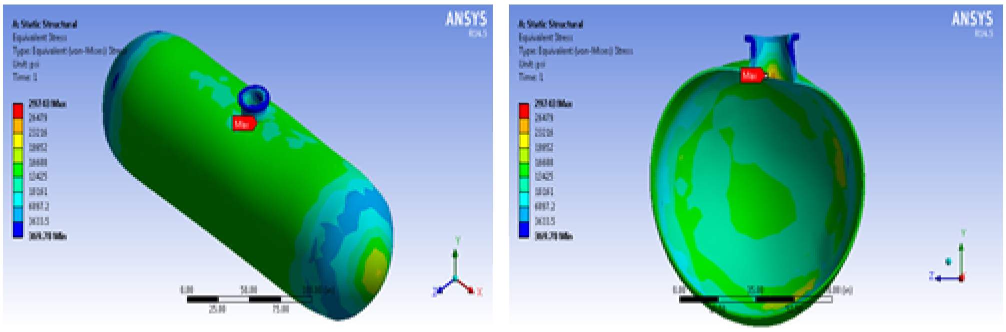


Figure 4.9: Stress profile of pressure vessel with nozzle at 8 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 8 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 4.8 and figure 4.9, now same way another structural analysis of pressure vessel with nozzle at 8 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 4.2.

4.5.2 FEA result summary of Pressure vessel with nozzle at 8 in. offset

Table 4.2: Result summary of pressure vessel with nozzle at 8 in. offset.

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	27243	369.78	0.0996	0
15	23829	310.89	0.09982	0
30	21686	216.67	0.1005	0
45	23244	157.94	0.099622	0

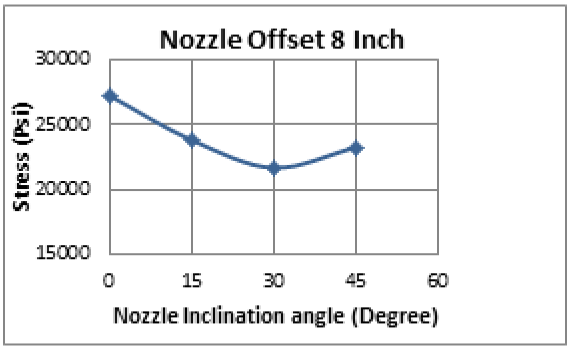


Figure 4.10: Plot of Von-Misses stresses on pressure vessel without reinforcement nozzle at 8 in. offset

4.6 Pressure Vessel with Nozzle at 16 in. Offset from Center

The model is prepared with single nozzle with offset distance is 16 in. and a base inclination of 0 $^{\circ}$.

4.6.1 FEA of Pressure vessel with nozzle at 16 in. offset at 0 degree inclination

Structural analysis of pressure vessel with nozzle at 16 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 4.11.The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition.

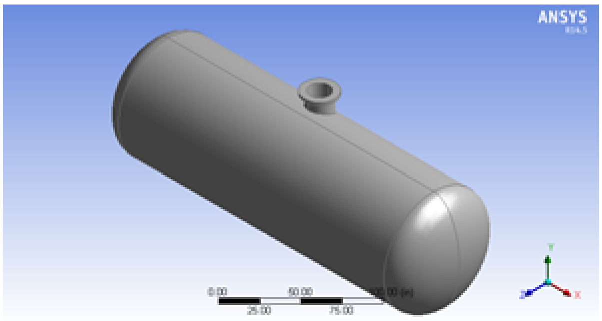


Figure 4.11: Pressure vessel with nozzle at 16 in. offset and 0 degree inclination

Result Summary
Maximum stress on nozzle: 22132 psi
Minimum stress on vessel: 212.58 psi
Maximum deformation on vessel: 0.10014 in.
The stress profiles are as shown in the figure 4.12.

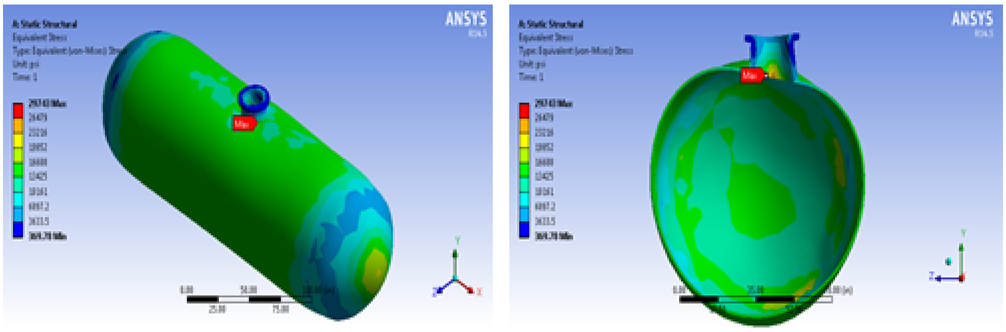


Figure 4.12: Stress profile of pressure vessel with nozzle at 16 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 16 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 4.11 and figure 4.12,now same way another structural analysis of pressure vessel with nozzle at 16 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 4.3.

4.6.2 FEA Result Summary of Pressure Vessel with Nozzle at 16 in. Offset

Table 4.3: Result summary of pressure vessel with nozzle at 16 in. offset

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	22132	212.58	0.10014	0
15	23042	258.59	0.1005	0
30	22745	348.05	0.099617	0
45	23320	222.82	0.09927	0

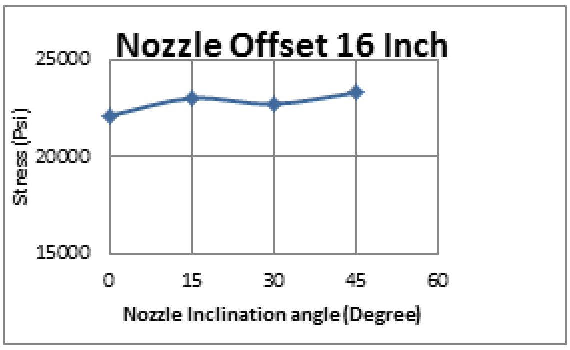


Figure 4.13: Plot of Von-Misses stresses on pressure vessel without reinforcement nozzle at 16 in. offset

4.7 Pressure Vessel with Nozzle at 24 in. Offset from Center

The model is prepared with single nozzle with offset distance is 24 in. and a base inclination of 0 $^{\circ}$.

4.7.1 FEA of Pressure Vessel with Nozzle at 24 in. Offset at 0 Degree Inclination

Structural analysis of pressure vessel with nozzle at 24 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 4.14.The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition

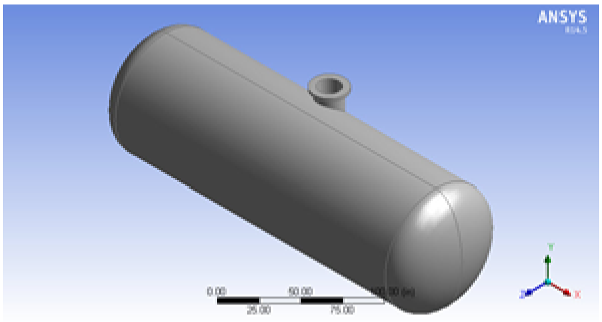


Figure 4.14: Pressure vessel with nozzle at 24 in. offset at 0 degree inclination

Result summary
Maximum stress on nozzle: 26931 psi
Minimum stress on vessel: 222.1 psi
Maximum deformation on vessel: 0.1003 in.
The stress profiles are as shown in the figure 4.15

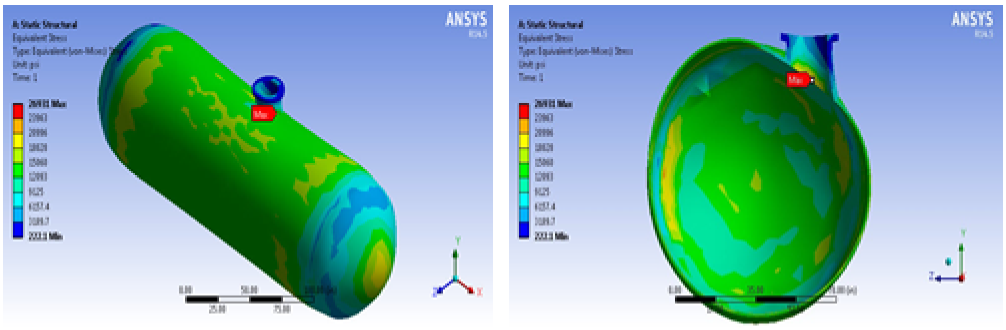


Figure 4.15: Stress profile of pressure vessel with nozzle at 24 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 24 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 4.14 and figure 4.15,now same way another structural analysis of pressure vessel with nozzle at 24 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 4.4.

4.7.2 FEA Result Summary of Pressure Vessel with Nozzle at 24 in. Offset

Table 4.4: Result summary of pressure vessel with nozzle at 24 in. offset

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	26931	222.1	0.1003	0
15	26333	219.35	0.1002	0
30	24904	128.29	0.099015	0
45	25056	73.719	0.098619	0

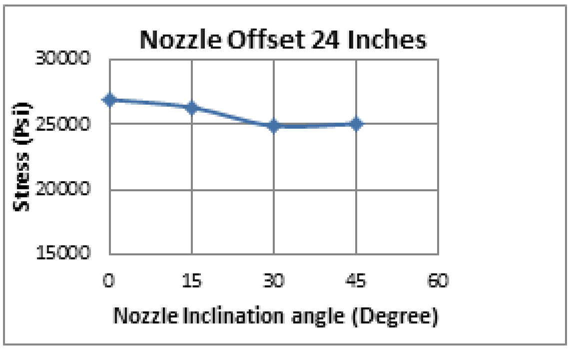


Figure 4.16: Plot of Von-Misses stresses on pressure vessel without reinforcement nozzle and 24 in. offset

4.8 Pressure Vessel with Nozzle at 32 in. Offset from Center

The model is prepared with single nozzle with offset distance is 32 in. and a base inclination of 0°.

4.8.1 FEA of Pressure Vessel with Nozzle at 32 in. Offset at 0 Degree Inclination

Structural analysis of pressure vessel with nozzle at 32 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 4.17.The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition.

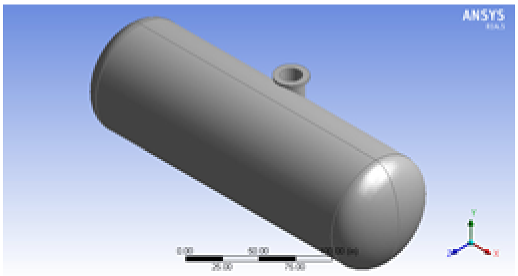


Figure 4.17: Pressure vessel with nozzle at 32 in. offset and 0 degree inclination

Result Summary
Maximum stress on nozzle: 18645 psi
Minimum stress on vessel: 164.98 psi
Maximum deformation on vessel: 0.068283 in.
The stress profiles are as shown in the figure 4.18

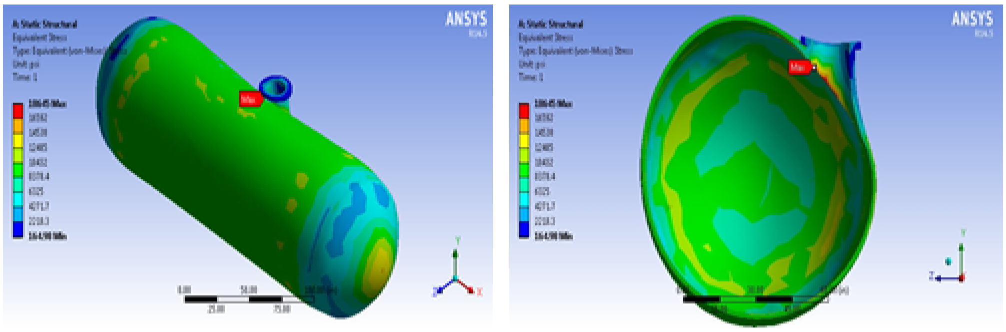


Figure 4.18: Stress profile of pressure vessel with nozzle at 32 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 32 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 4.17 and figure 4.18,now same way another structural analysis of pressure vessel with nozzle at 32 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 4.5.

4.8.2 FEA Result Summary of Pressure Vessel with Nozzle at 32 in. Offset

Table 4.5: Result summary of pressure vessel with nozzle and 32 in. offset

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	18645	164.98	0.068283	0
15	21107	260.37	0.09925	0
30	21764	233.42	0.099008	0
45	23121	150.48	0.099527	0

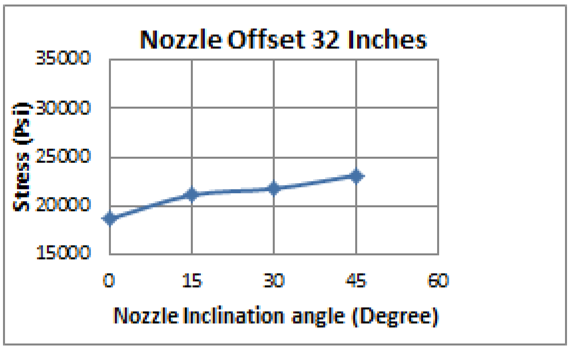


Figure 4.19: Plot of Von-Misses stresses on pressure vessel without reinforced nozzle at 32 in. offset

Chapter 5

Analysis of Pressure Vessel with Reinforced Nozzle

The most commonly used openings in vessels that require reinforcement, are the nozzles. The reinforcement material should be integral with the vessel or nozzle, such as obtained with weldments or forging as compared to riveted or bolted constructions, so as to offer no additional discontinuities other than those presented by geometry.

5.1 Nozzle Reinforcement[4]

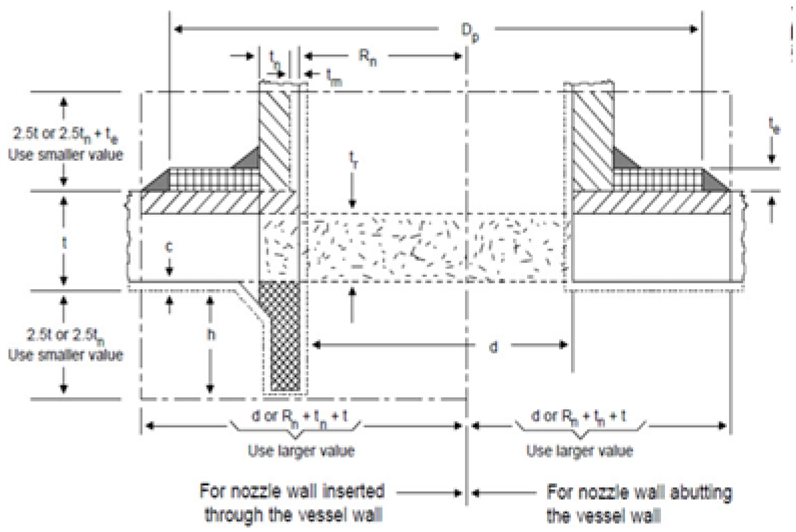


Figure 5.1: Reinforcement opening

For nozzle wall inserted through the vessel wall without reinforcing element larger value of d or $Rn + tn + t$ has been used.
For nozzle wall abutting the vessel wall larger value of d or $Rn + tn + t$ has been used.

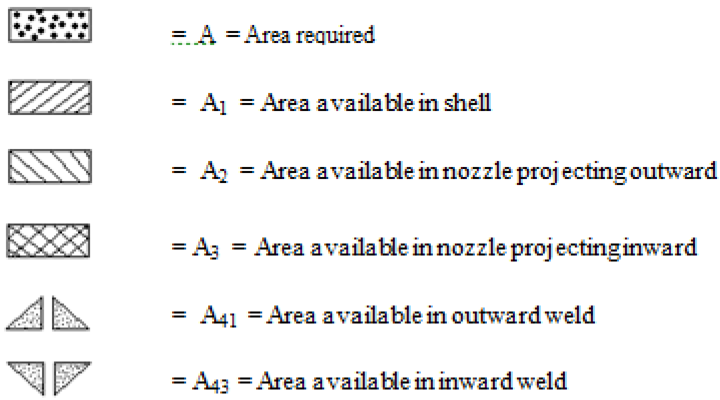


Figure 5.2: Nomenclature and formulas for reinforced opening

If $A1 + A2 + A3 + A41 + A43 > A$ Opening is adequately reinforced.
If $A1 + A2 + A3 + A41 + A43 < A$ Opening is not adequately reinforced reinforcing elements must be added and/or thickness must be increased.

5.2 Reinforcement of Openings[4]

The four basic requirements for reinforcement are:

⇒ Sufficient metal should be added to compensate for the weakening effect of the opening, yet preserving the strain pattern predominating in the vessel.

⇒ The reinforcing material should be placed immediately adjacent to the opening, but suitably disposed in profile and contour so as not to introduce stress concentration itself.

⇒ Reinforcement of an opening cannot be obtained by adding huge amounts of material because this has the reverse effect. It creates a "hard spot" on the vessel which does not allow general strain pattern occurring throughout the vessel to take place at this over-reinforced location. The result is a local over stressing. It is difficult to determine the correct amount of reinforcement.

⇒ The boundaries of the addition of effective reinforcing material can be obtained by examining the stress gradient along the cross section n-n with distance from the edge of the hole as shown in fig. 5.3.

The stress decreases rapidly with distance from the edge of the hole as shown by the shaded area in Fig. 5.3 (a). At the edge of the hole, $r = a$ and the maximum

stress is 2.5σ. At a distance from the edge of the hole equal to the radius, $r = 2a$, the stress has fallen to 1.23σ. Accordingly, at a distance from the whole edge equal to the radius, the effect of the opening on the stress is negligible, and this distance is usually accepted as the boundary limit for effective reinforcement to the vessel surface.
The boundary limit in the direction perpendicular to the plate surface can be approximated from the deflection characteristics of the nozzle or ring, which is doing the reinforcement.

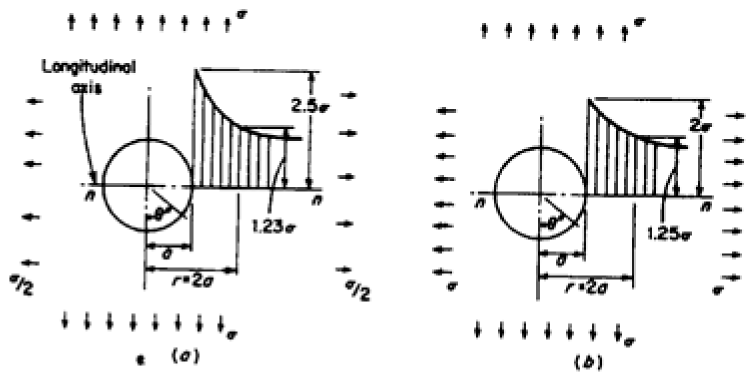


Figure 5.3: Variation in stress in region of a circular hole in (a) cylinder, (b) sphere subjected to internal pressure

In the case of a cylindrical nozzle this is a distance L beyond the surface of the vessel, and if an average nozzle wall thickness is taken as one-tenth of the nozzle radius, this gives,

L=$\sqrt{\frac{r^2h1}{1.285}}$

$= \sqrt{\frac{r^2 0.1}{1.285}}$

=$0.25r$

Thus, the reinforcement boundary limits ABCD can be completely established by the nozzle radius as shown in fig. 5.4 if it is required to replace the area OPQR removed from the vessel plate by the opening, the shaded nozzle area may be counted as compensating reinforcing material. If additional area is required, this must be located within the boundary ABCD for full effectiveness. This is the basic area replacement method of reinforcement used in pressure vessel design.

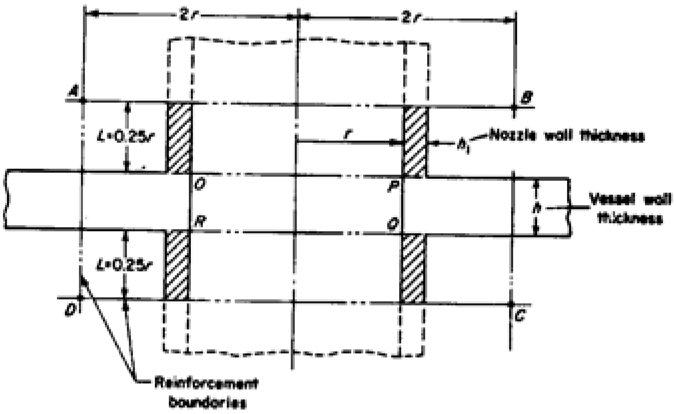


Figure 5.4: Reinforcement boundaries for circular openings in cylindrical and spherical vessels

5.3 Reinforcement Calculation for Reinforced Nozzle Using a Commercially Available Software

The material properties used for the analysis is shown in the table 5.1.The design data like internel pressure,temprature,etc has been obtained from paper of V. Skopinsky[6].The Conventional calculation and commercial software outputs are also compared. The nozzle to shell junction is to be designed for the given data mentioned below.

Table 5.1: Pad properties of nozzle with different offsets

Offset distance of nozzle (inch)	Pad diameter (inch)	Pad width (inch)	Pad thickness (inch)
0 inch offset	24	4	2
8 inch offset	22	3	2
16 inch offset	24	4	2
24 inch offset	24	4	2
32 inch offset	24	4	2

The entire reinforcement pad is made from same material SA-516 grade 70 having mechanical properties like young's modulus of $2.05e^5$ Mpa and Poison's ratio of 0.29.

5.4 Finite Element Analysis of Pressure Vessel with Reinforced Nozzle

FEA procedure for analysis of pressure vessel with reinforced nozzle is same as for pressure vessel without reinforcement. Load and constraints conditions are same as previous case but the model geometry is different due to reinforcement of nozzle which in turn affects meshing and result. The model is being meshed by using

ANSYS 2.0 as solid elements as analysis of pressure vessel with reinforcement except the introduction of reinforcement in the model.

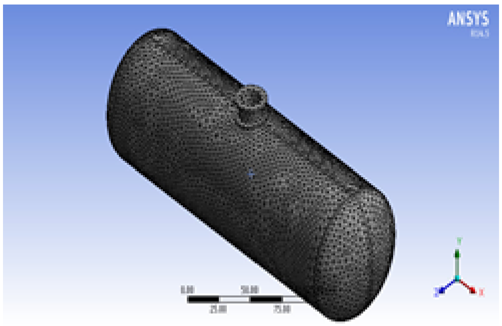


Figure 5.5: Meshed model of pressure vessel

Mesh Statics
Type of Element : Tetrahedrons
Number of Nodes : 77542
Number of Elements : 38849

5.5 Pressure Vessel with Reinforced Nozzle at 0 in. Offset from Center

The model is prepared with single nozzle with offset distance is 0 in. and a base inclination of 0 ° .

5.5.1 FEA of Pressure Vessel with Reinforced Nozzle at 0 in. Offset at 0 Degree Inclination

Structural analysis of pressure vessel with nozzle at 0 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 5.6.The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition.

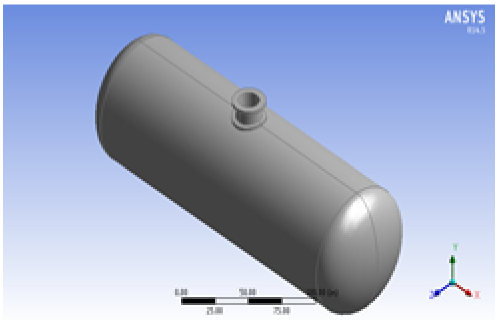
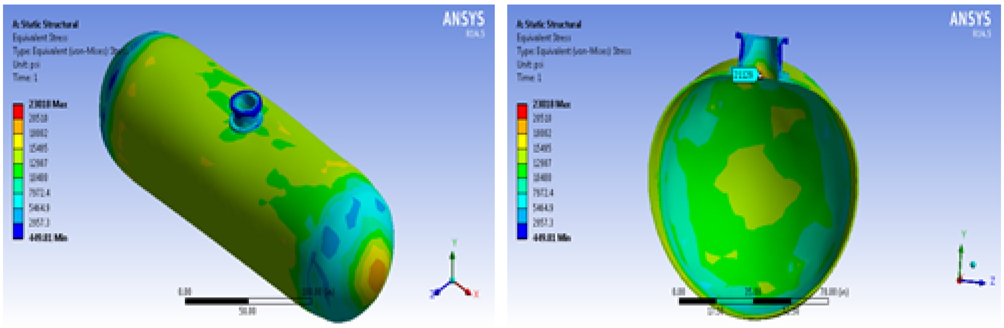


Figure 5.6: Pressure vessel model with reinforced nozzle at 0 in. offset and 0 degree inclination

Result Summary

Maximum stress on nozzle: 21129 psi
Minimum stress on vessel: 449.81 psi
Maximum deformation on vessel: 0.099313 in.



(a) (b)

Figure 5.7: Stress profile of pressure vessel with reinforced nozzle at 8 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 8 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 5.6 and figure 5.7,now same way another structural analysis of pressure vessel with nozzle at 0 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 5.2.

5.5.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 0 in. Offset

Table 5.2: Result summary of pressure vessel with reinforced nozzle at 0 in. offset

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	21129	449.81	0.099112	0
15	19845	384.18	0.099055	0
30	24286	388.47	0.099263	0
45	22329	330.2	0.10339	0

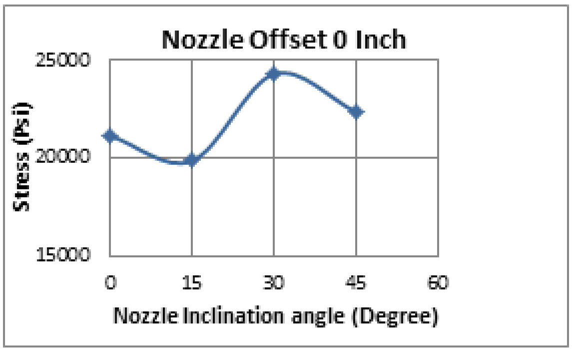


Figure 5.8: Plot of Von-Misses stresses on pressure vessel with reinforcement nozzle at 0 in. offset.

5.6 Pressure Vessel with Reinforced Nozzle at 8 in. Offset from Center

The model is prepared with single nozzle with offset distance is 8 in. and a base inclination of 0 $^\circ$.

5.6.1 FEA of Pressure Vessel with Reinforced Nozzle at 8 in. Offset at 0 Degree Inclination

Structural analysis of pressure vessel with nozzle at 8 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 5.9.The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition.

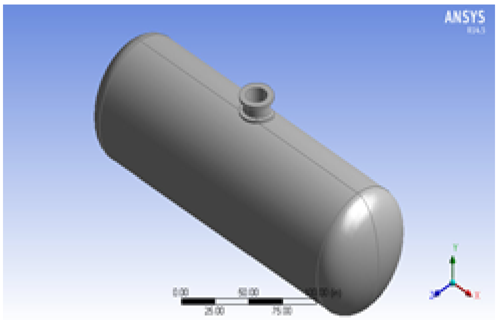
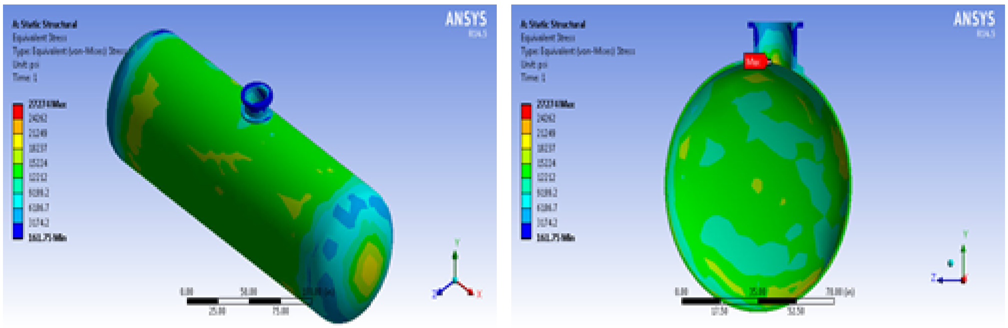


Figure 5.9: Pressure vessel with reinforced nozzle at 8 in. offset and 0 degree inclination

Result summary
Maximum stress on nozzle: 23774 psi
Minimum stress on vessel: 161.75 psi
Maximum deformation on vessel: 0.099239 in.
The stress profiles are as shown in the figure 5.10 (a) and (b)



(a) (b)

Figure 5.10: Stress profile of pressure vessel with reinforced nozzle at 8 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 8 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 5.9 and figure 5.10,now same way another structural analysis of pressure vessel with nozzle at 8 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 5.3.

5.6.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 8 in. Offset

Table 5.3: Result summary of pressure vessel with reinforced nozzle at 8 in. offset.

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	23774	161.75	0.099239	0
15	18954	434.91	0.09922	0
30	19222	268.98	0.1003	0
45	22678	365.84	0.099165	0

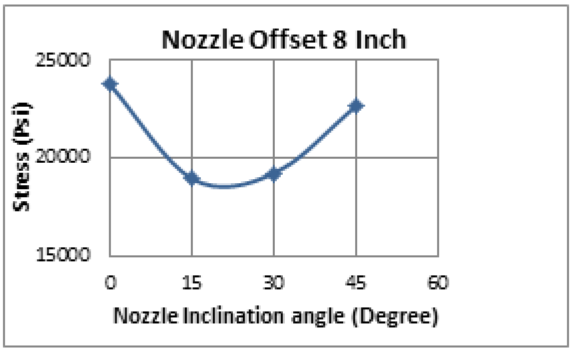


Figure 5.11: Plot of Von-Misses stresses on pressure vessel with reinforcement nozzle at 8 in. offset.

5.7 Pressure Vessel with Reinforced Nozzle at 16 in. Offset from Center

The model is prepared with single nozzle with offset distance is 16 in. and a base inclination of 0 ° .

5.7.1 FEA of Pressure Vessel with Reinforced Nozzle at 16 in. Offset at 0 Degree Inclination

Structural analysis of pressure vessel with nozzle at 16 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 5.12 .The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition.

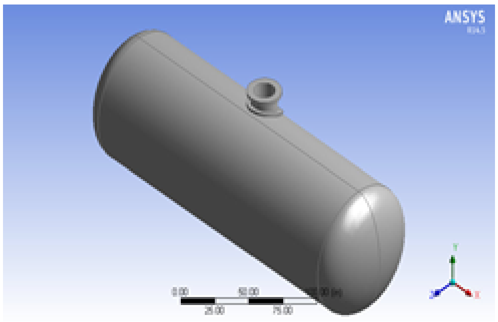
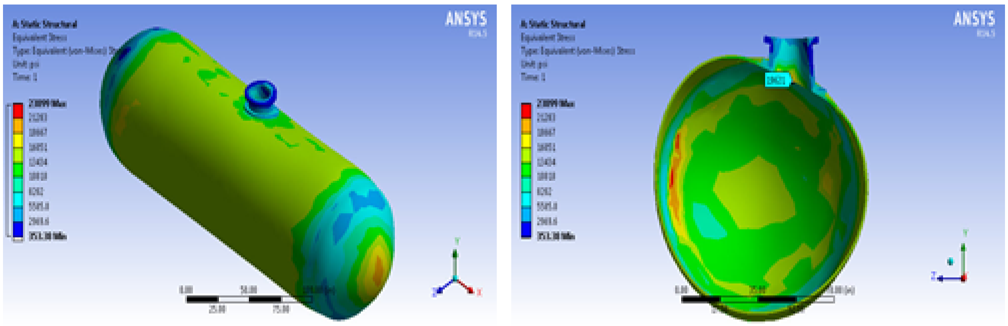


Figure 5.12: Pressure vessel with reinforced nozzle at 16 in. offset and 0 degree inclination

Result summary
Maximum stress on nozzle: 19621 psi
Minimum stress on vessel: 353.38 psi
Maximum deformation on vessel: 0.099667 in.
The stress profiles are as shown in the figure 5.13 (a) and (b)



(a) (b)

Figure 5.13: Stress profile of pressure vessel with reinforced nozzle at 16 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 16 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 5.12 and figure 5.13,now same way another structural analysis of pressure vessel with nozzle at 16 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 5.4.

5.7.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 16 in. Offset

Table 5.4: Result summary of pressure vessel with reinforced nozzle at 16 in. offset

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	19621	353.38	0.099667	0
15	20123	418.51	0.10004	0
30	19340	352.76	0.10021	0
45	19224	391.21	0.099082	0

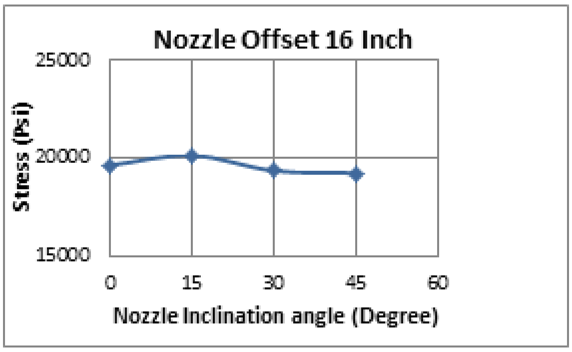


Figure 5.14: Plot of Von-Misses stresses on pressure vessel with reinforcement nozzle at 16 in. offset

5.8 Pressure Vessel with Reinforced Nozzle at 24 in. Offset from Center

The model is prepared with single nozzle with offset distance is 24 in. and a base inclination of 0°.

5.8.1 FEA of Pressure Vessel with Reinforced Nozzle at 24 in. Offset at 0 Degree Inclination

Structural analysis of pressure vessel with nozzle at 24 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 5.15.The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition.

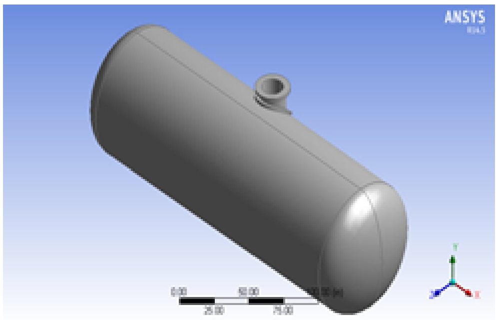
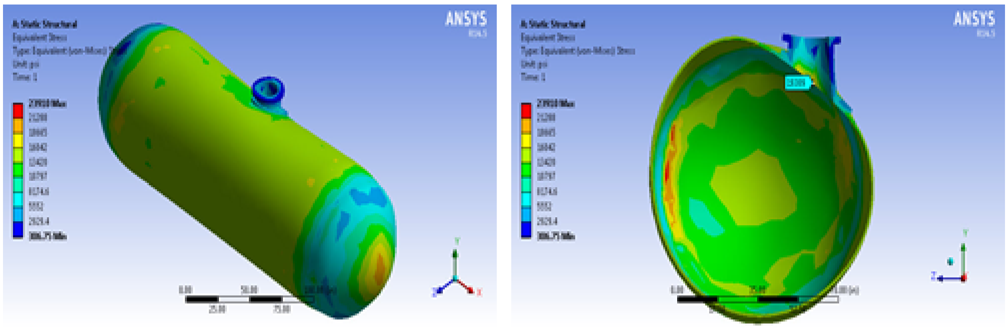


Figure 5.15: Pressure vessel with reinforced nozzle at 24 in. offset and 0 degree inclination

Result summary
Maximum stress on nozzle: 19309 psi
Minimum stress on vessel: 306.75psi
Maximum deformation on vessel: 0.10036 in.
The stress profiles are as shown in the figure 5.16 (a) and (b)



(a) (b)

Figure 5.16: Stress profile of pressure vessel with reinforced nozzle at 24 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 24 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 5.15 and figure 5.16,now same way another structural analysis of pressure vessel with nozzle at 24 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 5.5.

5.8.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 24 in. Offset

Table 5.5: Result summary of pressure vessel with reinforced nozzle at 24 in. offset

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	19309	306.75	0.10036	0
15	17952	324.78	0.099578	0
30	18412	229.97	0.099586	0
45	17662	212.26	0.099485	0

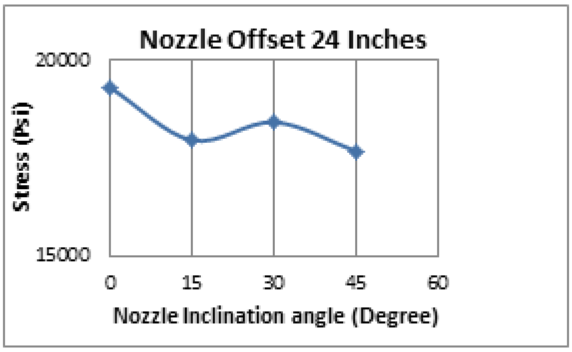


Figure 5.17: Plot of Von-Misses stresses on pressure vessel with reinforcement nozzle at 24 in. offset

5.9 Pressure Vessel with Reinforced Nozzle at 32 in. Offset from Center

The model is prepared with single nozzle with offset distance is 32 in. and a base inclination of 0 $^\circ$.

5.9.1 FEA of Pressure Vessel with Reinforced Nozzle at 32 in. Offset at 0 Degree Inclination

Structural analysis of pressure vessel with nozzle at 32 in. offset and 0 degree inclination is carried out and 3D solid model for the same is shown in figure 5.18.The evaluation and comparison of the same with the results of other inclination options have been attempted.Structural boundary conditions and loading are kept same as reference condition.

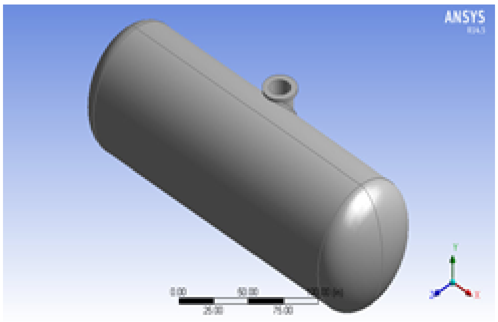
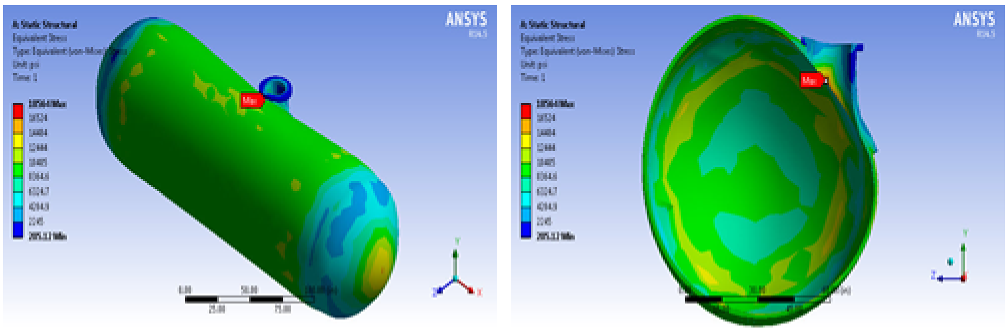


Figure 5.18: Pressure vessel with reinforced nozzle at 32 in. offset and 0 degree inclination

Result summary
Maximum stress on nozzle: 18564 psi
Minimum stress on vessel: 205.12 psi
Maximum deformation on vessel: 0.068204 in.
The stress profiles are as shown in the figure 5.19 (a) and (b)



(a) (b)

Figure 5.19: Stress profile of pressure vessel with reinforced nozzle at 32 in. offset and 0 degree inclination

Structural analysis of pressure vessel with nozzle at 32 in. offset at 0 degree inclination is carried out with 3D solid model shown in figure 5.18 and figure 5.19,now same way another structural analysis of pressure vessel with nozzle at 32 in. offset and different inclination angles of 15º, 30º and 45º modeled and analyzed for same pressure loading and boundary conditions and the result is shown in given table 5.6.

5.9.2 FEA Result Summary of Pressure Vessel with Reinforced Nozzle at 32 in. Offset

Table 5.6: Result summary of pressure vessel with reinforced nozzle at 32 in. offset

Inclination Angle	Stress (psi)		Deformation (inch)	
	Maximum	Minimum	Maximum	Minimum
0	18564	205.12	0.068204	0
15	18362	271.02	0.099411	0
30	21986	316.79	0.098684	0
45	20386	227.32	0.099807	0

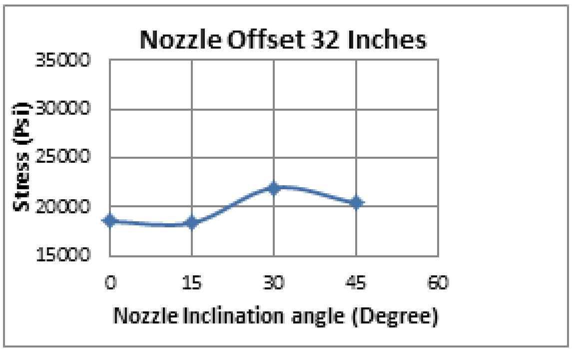


Figure 5.20: Plot of Von-Misses stresses on pressure vessel with reinforced nozzle at 32 in. offset

Chapter 6

Conclusion and Future Scope

6.1 Results for Pressure Vessel without Reinforcement

Pressure vessel with and without reinforcement with different type of nozzle location, different offset distances and nozzle inclination angles are modeled and analyzed for same pressure loading and boundary conditions.It is found that stress developed less when a nozzle inclination angle is 45º with 0,16 and 32 in.offset.

Table 6.1: Summary of Von-Mises stresses developed for a pressure vessel without reinforcement

Offset (Inches)→ Inclination ↓	0	8	16	24	32
0° Inclination	23586	27243	22132	26931	18645
15° Inclination	26263	23829	23042	26333	21107
30° Inclination	28644	21686	22745	24904	21764
45° Inclination	24308	23244	23320	25056	23121

Table 6.1 shows the maximum Von-mises stress value for evaluation of the effect of inclination angle on offset distance of nozzle. It is found from result that minimum stress is achieved when inclination angle is 45 ° and offset distance is 32 in.. Von-mises stress plot in figure 6.1 shows the effect of different inclination angle on different offset distances.

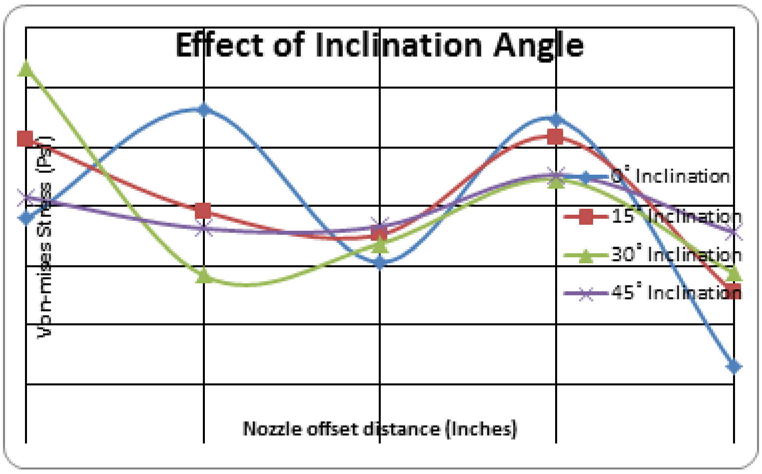


Figure 6.1: Plot of Von-Mises stresses on pressure vessel without reinforcement at different offset distance by varying nozzle inclination angle

Table 6.2: Summary of Von-Mises stresses on pressure vessel with reinforcement

Inclination (Degree) → Offset (Inches) ↓	0	15	30	45
0 inches	23586	26263	28644	24308
8 inches	27243	23829	21686	23244
16 inches	22132	23042	22745	23320
24 inches	26931	26333	24904	25056
32 inches	18645	21107	21764	23121

It is found from result that minimum stress is achieved when offset distance is 32 in.es and inclination angle is 0 ° .At 32 in.es offset stress is minimum for most of the inclination angles and effect of inclination angle is also less for 16 and 24 in. offsets.

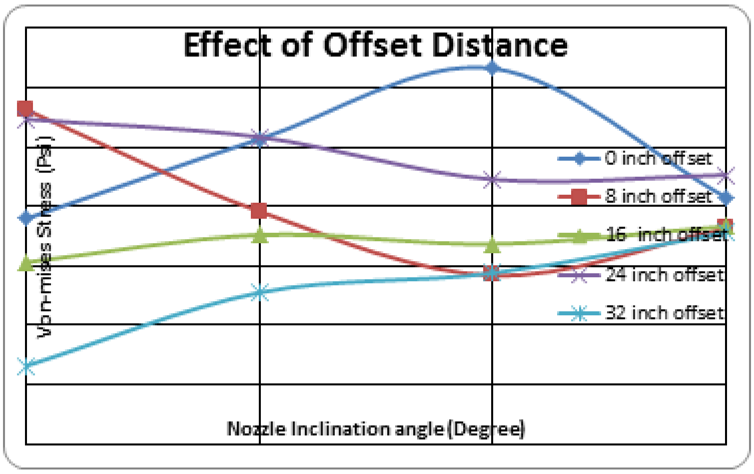


Figure 6.2: Plot of Von-Mises stresses on pressure vessel without reinforcement at different offset distance by varying nozzle inclination angle

6.2 Results for Pressure Vessel with Reinforcement

Pressure vessel with reinforced nozzle are modeled and analyzed for same pressure loading and boundary conditions. The result of the effect of single parameter are tabulated and also shown in the graph.The combine result of different factors is shown below.

Table 6.3: Summary of Von-Mises stresses on pressure vessel with reinforcement

Offset (Inches)⟶ Inclination ↓	0	8	16	24	32
0° Inclination	21129	23774	19621	19309	18564
15° Inclination	19845	18954	20123	17952	18362
30° Inclination	24286	19222	19340	18412	21986
45° Inclination	22329	22678	19224	17662	20386

Table 6.3 shows the maximum Von-Mises stress value for evaluation of the effect of inclination angle on offset distance of nozzle. It is found from result that minimum stress is achieved when inclination angle is 45 ° and offset distance is 24 in.. Von-Mises stress plot in figure 6.3 shows the effect of different inclination angle on different offset distances.

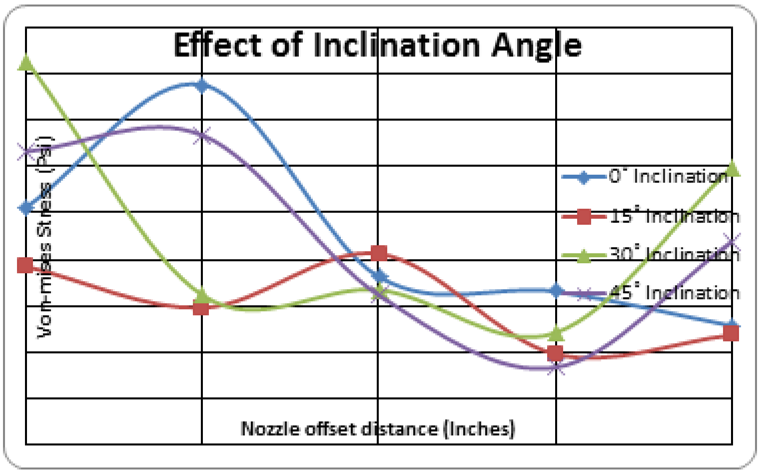


Figure 6.3: Plot of Von-Mises stresses on pressure vessel with reinforcement at different offset distance by varying nozzle inclination angle

It is found from result that minimum stress is achieved when offset distance is 24 in.es and inclination angle is 45 ° .At 24 in.es offset stress is minimum for most of inclination angles and effect of inclination angle is also less.

Table 6.4: Summary of Von-Mises stresses developed for a pressure vessel without reinforcement

Inclination (Degree) → / Offset (Inches) ↓	0	15	30	45
0 inches	21129	19845	24286	22329
8 inches	23774	18954	19222	22678
16 inches	19621	20123	19340	19224
24 inches	19309	17952	18412	17662
32 inches	18564	18362	21986	20386

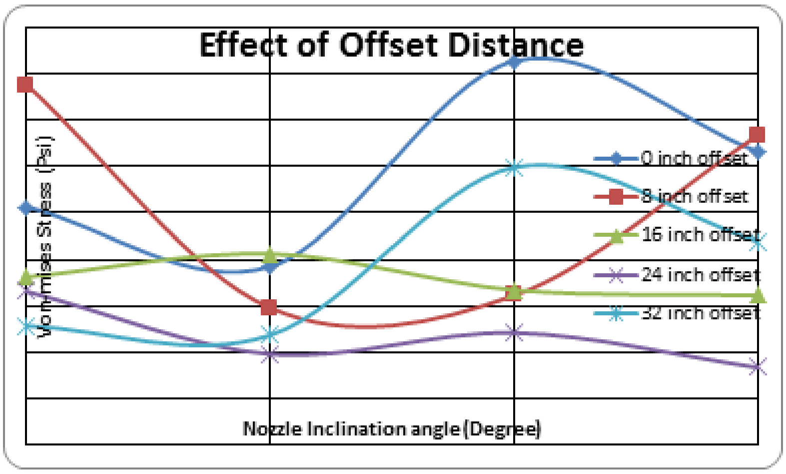


Figure 6.4: Plot of Von-Mises stresses on pressure vessel without reinforcement at different offset distance by varying nozzle inclination angle

Conclusion

⇒ It is found that the stress level is within the allowable stress in a pressure vessel with reinforcement which is within the range of allowable stress in pressure vessel without reinforcement.

⇒ It is found from results that minimum Von-mises stress at nozzle is achieved for pressure vessel without reinforcement when offset distance is 32 in. and inclination angle is 0 ° .For nozzle with reinforcement minimum stress is for 24 offset and 45 ° inclination.

⇒ Stress variation with change in inclination angle is less in case of 24 in. and 32 in. offsets.

⇒ The effect of different inclination angles on different offset distances have been studied at 0° inclination angles stress is less for 0, 16 and 32 in.offsets for pressure vessel without reinforcement. For pressure vessel with reinforcement at 15° inclination stress is minimum at 0,8,24 and 32 in.es offset.

⇒ Minimum stress observed at 0 in. offset and at 0 ° Inclination, at 8 in. offset and at 30 ° inclination, at 16 in. offset and at 0 ° inclination, at 24 in. offset and at 30 ° inclination, and at 32 in. offset and at 0 ° inclination for pressure vessel without reinforcement.

⇒ Minimum stress observed at 0 in. offset and at 15 ° inclination, at 8 in. offset is and 15 ° inclination, at 16 in. offset and at 45 ° inclination, at 24 in. offset and at 45 ° inclination, at 32 in. offset and at 15 ° inclination for pressure vessel with reinforcement.

6.3 Future Scope

⇒ Pressure vessel with different type of nozzle location with different offsets distances and inclination angles are modeled and analyzed for same pressure loading and boundary conditions.

⇒ There is also promising possibility to study the effect of nozzle location on pressure vessel head.

⇒ Effect of external loading on nozzle can be considered with the pressure load which is neglected in current work.

⇒ There is also further scope of study effect of multiple nozzles for their locations and saddle support for pressure vessel.

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