

# **Chapter 1**

## **Introduction**

## 1.1 What are pressure vessels?

According to the ASME Boiler and Pressure Vessel Code (BPVC), Code Section VIII, pressure vessels are containers for the containment of pressure, either internal or external.

The first requirement in vessel design is to determine the actual values of the loads and the conditions in which the vessel will be subjected in operations. A design engineer should determine conditions and all pertaining data as thoroughly and as accurately as possible and rather be conservative. The principle loads to be considered in the design of PVs are Design Pressure, dead loads, wind loads, earthquake loads, temperature loads, piping loads, impact and cyclic loads.

Many different combinations of the above loadings are possible, the designer must select the most probable combination of simultaneous loads for an economical and safe design.

## 1.2 Aim

- i. To design a horizontal pressure vessel using **ASME code Section VIII Division 1**.
- ii. To design saddle for seismic and wind loads using **IS codes IS 1893 Part I and IS 875 Part III** respectively.
- iii. To compare manual results with FEA result using two analysis software Ansys R15 and **BInventor 2015**.
- iv. To suggest a better MOC.

## 1.3 Scope

Scope of work includes design and analysis of horizontal pressure vessel it also includes the design of saddle, base plate, anchor bolts considering static, dynamic, earthquake and wind loads. It also includes CAD modeling and analysis of pressure vessel according to data sheet given by the client.

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## **Chapter 2**

### **Literature Survey**

- 1) Comparing the FEA Results with experimental data showed that the FEA software predicted the failure pressure and location very well for the symmetric shaped pressure vessel, however for the nonsymmetrical shaped pressure vessel, the FEA software predicted the failure pressure within a reasonable range, but the component failed at a weld instead of the predicted location. This difference in failure location was likely caused by varying material properties in both the weld and the location where the vessel was predicted to fail.

*Failure prediction of pressure vessels using finite element analysis-*

*By Christopher J Evans and Timothy F Miller*

*asmedigitalcollection.asme.org*

*OCTOBER 2015, Volume 137/051206-9*

- 2) In recent years, several designs were developed to employ bolted devices to perform local Pressure testing of flange-to-nozzle, flange-to-pipe, and nozzle-to shell attachment welds. Due to the cost and equipment downtime associated with performing a full conventional pressure test and the desire to reduce repair costs, several petrochemical companies adopted the use of such devices. The purpose of this paper is to compare the stress values and stress distribution associated with conventional and local pressure testing techniques. The advantages and disadvantages of both approaches are discussed and the conclusions are supported by a practical example.

*Comparative Study for Stresses in Nozzle and Flange Welds Generated During Conventional Pressure Testing and Local Pressure Testing Using Bolted Devices*

*By -Ayman M. Cheta*

*-Richard Brodzinski*

*Vol. 129, NOVEMBER 2007*

## **Chapter 3**

### **Design Methodology**

In general, pressure vessels designed in accordance with the **ASME Code, Section VIII, Division 1**, are designed by rules and do not require a detailed evaluation of all stresses. It is recognized that high localized and secondary bending stresses may exist but are allowed for by use of a higher safety factor and design rules for details. It is required, however, that all loadings (the forces applied to a vessel or its structural attachments) must be considered.

While the Code gives formulas for thickness and stress of basic components, it is up to the designer to select appropriate analytical procedures for determining stress due to other loadings. The designer must also select the most probable combination of simultaneous loads for an economical and safe design.

## **3.1 Loads**

### **3.1.1 Pressure Loads**

Design pressure is the used to determine the minimum required thickness of each vessel shell component and denotes the difference between the internal and external pressure (usually the design and atmospheric pressures). It includes a suitable margin above the operating pressure (10% of operating pressure or 10 psi min) plus any static head of operating liquid. Minimum design pressure for Code stamping is not required. Vessels with negative gauge operating pressure are generally designed for full vacuum.

The maximum allowable working (operating) pressure is then, by the Code definition, the maximum gauge pressure permissible at the top of the completed vessel in its operating position at the designated temperature

### **3.1.2 Temperature Loads**

Design temperature is more a design environmental condition than a design load, +since only a temperature change combined with some body restraint or certain temperature gradients will originate thermal stresses. However, it is an important design condition that

influences a great degree the suitability of the selected material for construction. Decrease in metal strength with rising temperatures, increased brittleness with falling temperature, and the accompanying dimensional changes are just a few of the phenomena to be taken into account for the design.

For most standard vessels the design temperature is the maximum temperature of the operating fluid plus 50° F as a safety margin, or the minimum temperature of the operating fluid, if the vessel is designed for low- temperature service (below -20 ° F)

### **3.1.3 Dead Loads**

Dead Loads are the loads due to the weight of the vessel itself and any part permanently connected with the vessel. Depending on the overall state, a vessel can have three different weights important enough to be considered in the vessel design.

#### **3.1.3.1 Erection**

Dead load is the weight of the vessel without any external extension, fireproofing, operation contents, or any external structural attachments and piping. Basically, it is the weight of a stripped vessel as hoisted on the job site. In some small-diameter columns the removal internals (trays) are shop-installed, and they have to be included in the erection weight. Each such case has to be investigated separately.

#### **3.1.3.2 Operating Dead Load**

#### **3.1.3.3 Shop Test Dead Load**

### **3.1.4 Wind Loads**

Wind can be described as a highly turbulent flow of air sweeping over the earth's surface with a variable velocity, in gust rather than in a steady flow. The wind can also be assumed to possess a certain mean velocity on which local three-dimensional turbulent fluctuation are superimposed. The direction of the flow is usually horizontal; however, it may possess a vertical

component when passing over a surface obstacle. The wind velocity  $V$  is affected by the earth surface friction and increase gradient level above which the wind velocity remains constant.

## **3.2 Stresses**

ASME Code, Section VIII, Division 1 does not explicitly consider the effects of combined stress. Neither does it give detailed methods on how stresses are combined. ASME Code, Section VIII, Division 2, on the other hand, provides specific guidelines for stresses, how they are combined, and allowable stresses for categories of combined stresses. Division 2 is design by analysis whereas Division 1 is design by rules. Although stress analysis as utilized by Division 2 is beyond the scope of this text, the use of stress categories, definitions of stress, and allowable stresses is applicable

### **3.2.1 Types of Stress**

The following list of stresses describes types of stress without regard to their effect on the vessel or component. They define a direction of stress or relate to the application of the load.

1. Tensile
2. Compressive
3. Shear
4. Bending
5. Bearing
6. Axial
7. Discontinuity
8. Membrane
9. Principal
10. Thermal
11. Tangential
12. Load induced
13. Strain induced
14. Circumferential



15. Longitudinal

16. Radial

17. Normal

### **3.2.2 Classes of Stress**

Classes of stress are defined by the type of loading which produces them and the hazard they represent to the vessel.

1. Primary stress
  - a. General:
    - i. Primary general membrane stress
    - ii. Primary general bending stress
  - b. Primary local stress
2. Secondary stress
  - a. Secondary membrane stress
  - b. Secondary bending stress
3. Peak stress

### **3.2.3 Primary general stress**

These stresses act over a full cross section of the vessel. They are produced by mechanical loads (load induced) and are the most hazardous of all types of stress. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses are generally due to internal or external pressure or produced by sustained external forces and moments. Thermal stresses are never classified as primary stresses.

Primary general stresses are divided into membrane and bending stresses. The need for dividing primary general stress into membrane and bending is that the calculated value of a primary bending stress may be allowed to go higher than that of a primary membrane stress. Primary stresses that exceed the yield strength of the material can cause failure or gross distortion.

#### **Primary general membrane stress**

This stress occurs across the entire cross section of the vessel. It is remote from discontinuities such as head-shell intersections, cone-cylinder intersections, nozzles, and supports. Examples are:

- a. Circumferential and longitudinal stress due to pressure.
- b. Compressive and tensile axial stresses due to wind.
- c. Longitudinal stress due to the bending of the horizontal vessel over the saddles.
- d. Membrane stress in the center of the flat head.
- e. Membrane stress in the nozzle wall within the area of reinforcement due to pressure or external loads.
- f. Axial compression due to weight.

#### **Primary general bending stress**

Primary bending stresses are due to sustained loads and are capable of causing collapse of the vessel. There are relatively few areas where primary bending occurs:

- a. Bending stress in the center of a flat head or crown of a dished head.
- b. Bending stress in a shallow conical head.
- c. Bending stress in the ligaments of closely spaced openings.

### **3.2.4 Secondary stress**

The basic characteristic of a secondary stress is that it is self-limiting. As defined earlier, this means that local yielding and minor distortions can satisfy the conditions which caused the stress to occur. Application of a secondary stress cannot cause structural failure due to the restraints offered by the body to which the part is attached. Secondary mean stresses are developed at the junctions of major components of a pressure vessel. Secondary mean stresses are also produced by sustained loads other than internal or external pressure. Radial loads on nozzles produce secondary mean stresses in the shell at the junction of the nozzle. Secondary stresses are strain-induced stresses.

#### **Secondary membrane stress**

- a. Axial stress at the juncture of a flange and the hub of the flange.

- b. Thermal stresses.
- c. Membrane stress in the knuckle area of the head.
- d. Membrane stress due to local relenting loads.

**Secondary bending stress**

- a. Bending stress at a gross structural discontinuity:
- b. The non-uniform portion of the stress distribution in a
- c. The stress variation of the radial stress due to internal

# **Chapter 4**

## **Design**

## 4.1 Design of Shell

All formulas are taken from ASME SECTION VIII DIVISION 1 UG-27 (c)

Material selection is from ASME 2011a Section II

### Material and Conditions

Material	SA 516 grade 70		
P :Design pressure	36	psi	Data Sheet
S : Allowable stress	18500	psi	
D :Shell Diameter	1100	mm	Data Sheet
	43.307	inch	
nt :Thickness of vessel	0.5	inch	Assumption
c.a :Corrosion Allowance	3	mm	Data Sheet
	0.118	inch	
utp :Under tolerance allowance	12.50%		Data Sheet

### Calculation of pressure due to static head:

Pressure due to static head	$P_s$	density*g*H	
		16794720	N/mm <sup>2</sup>
		16.795	kN/mm <sup>2</sup>
		2.436	psi
		3	psi
Total design pressure		<b>39</b>	psi

### Calculating Internal radius of shell considering Tolerance& Allowances:

Under tolerance		$nt*utp$	
		0.0625	inch
Thickness		$nt-c.a-ut$	
		0.319	inch
Internal radius	$R_i$	<b>21.334</b>	inch

### Calculating required thickness for design pressure

Along longitudinal section

$$t_a = \frac{P * R_i}{2 * S * E + 0.4 * P} + c.a \quad \mathbf{0.145} \quad \text{inch}$$

Along circumferential section

$$t_b = \frac{P * R_i}{S * E - 0.6 * P} + c.a \quad \mathbf{0.182} \quad \text{inch}$$

since

$t_b > t_a$

Take

$t_b$

Since given thickness is less than

$t_b$

Hence design is safe

**Now checking for maximum pressure**

Pressure at longitudinal section:

$$\frac{2 * S * E * t}{Ri - 0.4 * t}$$

**473.66876** psi

Pressure at circumferential section:

$$\frac{S * E * t}{Ri + 0.6 * t}$$

**192.146162** psi

Since the pressures are more than design pressure, Design is safe

## 4.2 Design of Head

### Design of Elliptical Head

**ALL FORMULAS ARE FROM ASME SECTION VIII DIVISION 1 UG 32(f)  
MATERIAL SELECTION FROM ASME 2011a SECTION II PART A**

Material :	SA516-70		
S: Allowable stress:	18500	psi	
E: Head longitudinal efficiency	1		
P: Design Pressure	39	psi	
Dimensions:			
Do: Outer diameter of head	1100	mm	data sheet
	43.3071	inch	
t <sub>b</sub> : thickness before forming:	0.5	inch	Assumption
t <sub>f</sub> : thickness after forming:	0.425	inch	
c.a: corrosion allowance:	3	mm	data sheet
	0.118	inch	
Straight skirt length	2	inch	

#### Variables:

nt: Thickness with corrosion allowance removed	t <sub>f</sub> -ca		
	0.307	Inch	
D: ID with corrosion allowance removed	D <sub>o</sub> -2*nt		
	42.69	Inch	
h: Inside crown height	D/4		
	10.67	Inch	

k: Kfactor	$0.167 \left( 2 + \left( \frac{D}{2h} \right)^2 \right)$		
	1.002		

#### Calculating Required Thickness

Required Thickness:	t <sub>req</sub>	$\frac{P * D * k}{2 * S * E - 0.2 * P} + ca$	
		0.16321	inch

Check:

$t_{req} \leq t_f$  Acceptable

**Checking for maximum Pressure**

Maximum Pressure:

$$P_{max} = \frac{2 * S * E * nt}{k * D + 0.2 * nt}$$

**265.054** psi

Check

$P_{max} \geq P$  Acceptable

HENCE DESIGN IS SAFE



## 4.3 Design of Saddle

### 4.3.1 Operating and Empty Weight of Vessel

D :Diameter of vessel	D	43.3	Inch
L :Length of shell	L	130	Inch
t :Thickness of shell	t	0.5	Inch
l :Projection of nozzle	l	9.44	Inch
Diameter of inlet	d1	1.97	inch
Diameter of outlet	d2	3.15	inch
Diameter of pressure transmitter	d3	1.97	inch
Diameter of drain	d4	1.58	inch
Diameter of manway	d5	23.7	inch
Volume of vessel	V	3414.44	liters
		3.414	m <sup>3</sup>

#### Calculating Weight of Shell

Weight of shell	$W_s = 0.898 \cdot D \cdot L \cdot t$		
		<b>2527.421</b>	lbs

#### Calculating Weight of Head

Weight of head	$W_e = 0.307 \cdot D^2 \cdot t$		
		287.7956	lbs
Weight of two heads		<b>575.5912</b>	lbs

#### Calculating Weight of Nozzles

Weight of nozzles	$W_1 = 0.898 \cdot D \cdot L \cdot t$		
		8.349963	lbs
	W2	13.35146	lbs
	W3	8.349963	lbs
	W4	6.696925	lbs
	W5	100.4539	lbs
Total weight of nozzles	$W_n$	<b>137.2022</b>	lbs

#### Calculating Weight of Flanges

Weight of flanges	F1	2.07	kg
	F2	3.98	kg
	F3	2.07	kg
	F4	1.33	kg
	F5	87.13	kg
Total weight of flange	F	96.58	kg
		<b>212.96</b>	lbs

#### Empty weight of vessel

		3453.17	lbs
		1566.065	kg

**Calculating Weight of Fluid**

Weight of fluid

density\*Volume

**3652.98**

kg

**Operating weight of vessel**

**5219.045**

kg

**11507.99**

Lbs

### 4.3.2 Seismic Load Calculation

Seismic Loads are as per IS 1893 part A

W :Operating weight of vessel	5219.05	kg
B :Effective height of vessel	0.85	m
D :Mean shell diameter	0.55	m
Seismic zone	III	
Z :Seismic zone factor	0.16	moss
I :Importance factor	1	moss
R :Response reduction factor	5	moss
Time period of vibration		

Average response acceleration factor  $S_a/g$  2.5 moss

**Design of horizontal seismic coefficient**  $A_h$

$$\frac{\left(\frac{Z}{2}\right)\left(\frac{S_a}{g}\right)}{\left(\frac{R}{I}\right)}$$

**0.04**

#### Calculating Base Shear

Base shear  $A_h * W$   
**208.762** kgf

Portion of seismic force applied at the top of vessel  $F_t$  **0** kgf

$F_t=0$  if  $(H/D \leq 3, T \leq 0.7\text{sec})$

#### Calculating Seismic Load and Moment

Seismic load  $F$   $V - F_t$   
**208.762** kgf

Seismic moment at base  $M_b$   $F_t * H + \frac{2}{3} * F * H$   
**118.298** kgfm

### 4.3.3 Wind Load Calculation

Wind Loads are as per IS 875 part 3

W <sub>o</sub> : Operating weight of vessel		4933.5	lbs
W <sub>e</sub> : Empty weight of vessel		2476	lbs
D : Effective diameter		43.3	inch
L : Length of vessel		130	ft
B : Effective height of vessel		2.79	ft
E :Saddle width		2.89	ft
I : Importance factor		1	moss
C <sub>f</sub> :Shape factor		0.76	moss
G :Gust factor		0.8	moss
Basic wind speed		44	moss
K <sub>z</sub> :Velocity pressure coefficient		0.1	moss

#### Calculating Projected Area of Vessel

Projected area in longitudinal direction	A <sub>l</sub>	<b>1471.79</b>	inch <sup>2</sup>
Projected area in transverse direction	A <sub>t</sub>	<b>5629</b>	inch <sup>2</sup>

#### Calculating Wind Pressure

Wind pressure	q <sub>z</sub>	<b>0.49562</b>	
Maximum longitudinal force of wind	F <sub>l</sub>	<b>443.501</b>	lbs
Maximum transverse force of wind	F <sub>t</sub>	<b>1696.21</b>	lbs

#### Calculating Wind Load

Q :Wind Load			
Longitudinal wind load	$\frac{W}{2} + \frac{Fl * B}{Ln}$	<b>1247.52</b>	lbs

Transverse wind load

$$\frac{W}{2} + \frac{3 * Ft * B}{E} \quad \mathbf{3694.28} \quad \text{lbs}$$

### 4.3.4 Saddle Design

W :Total weight of equipment:		5219.05	kg
sQ :load on one saddle		2609.52	kg
	5740.95		lbs
D :Diameter of shell	1100		mm
	43.307		inch
R :Radius of shell	550		mm
	21.654		inch
b :Saddle width	279.4		mm
	11		inch
θ :Saddle angle	120		degree
β :Half saddle angle	60		degree
	1.0472		radians
m :Base plate width	0.8*D		mm
	34.6457		inch
Material of saddle	SA 516		
S <sub>y</sub> :Yield strength of saddle		20000	psi
S <sub>a</sub> :Allowable stress	18500		psi

#### Dimensions of Saddle Components

Wear plate thickness		t <sub>f</sub>	<b>10</b>	<b>mm</b>
Web thickness	t <sub>w</sub>	<b>10</b>		mm
		<b>0.3937</b>		inch
Base plate thickness	t <sub>b</sub>	<b>20</b>		mm
Height of web plate	h	<b>33.46</b>		inch
No of gusset		<b>4</b>		
gusset thickness	t <sub>g</sub>	<b>10</b>		mm

#### Checking Dimensions for Loads

no of gussets:	$\frac{m}{24} + 1$		
		2.44357	
		<b>4</b>	take even no

gusset thickness

if $D \leq 6$ ft	$t_g$	0.38	inch
		9.5	mm
if $D \geq 6$ ft	$t_g$	0.5	inch
		<b>12.7</b>	mm

Acceptable

Calculation of top flange thickness

$$P = \frac{Q(1 + \cos\beta)}{r(\pi - \beta + \cos\beta * \sin\beta)}$$

157.626

lbs/inch

Bending moment:

$$M = \frac{P * b}{8}$$

216.735

lbs

$$\sqrt{\frac{6 * M}{S a}}$$

$t_f$

0.265

inch

6.73423

mm

Acceptable

Height of the web plate

$$t_w * \sqrt{\frac{1500(1800 * t_w - P)}{P}}$$

h

101.096

inch

Not Acceptable

Base plate thickness

$$\sqrt{\frac{0.75 * Q * b}{S_y * m}}$$

$t_w$

9.05792

mm

Acceptable

Check available area with required area

F :Maximum horizontal splitting force

Factor for force	k8	0.204	moss
	F	k8*Q	

532.343	Kg
1171.15	lbs

Area required at bottom cross section  $\frac{F}{0.66 * S_a}$

0.09592	inch <sup>2</sup>
61.8822	mm <sup>2</sup>
Area provided at bottom of cross section	12631.4 mm <sup>2</sup>
19.5787	Inch <sup>2</sup>

Since provided area is more  
,Design is safe

**Anchor bolt calculation for Seismic and Wind Loads**

Q<sub>o</sub> :Operating weight of vessel on one saddle 2609.52 kg  
5740.95 lbs

Q<sub>1</sub> :Wind/seismic load on one saddle 1675.28 kg

Material of construction SA-307 Gr B

S<sub>b</sub> :Allowable stress for bolt 20000 psi  
14.06 kg/mm<sup>2</sup>

B :Distance from ground to vessel center 850 mm

**Selecting bolt of size** M36

N :No of bolts per saddle 12

d :diameter of bolt 36 mm

A :Area of bolt 1017.36 mm<sup>2</sup>

Z :Section modulus of bolt 4578.12 mm<sup>3</sup>

Uplift of bolts

check(Q<sub>1</sub>>Q<sub>o</sub>) NO Uplift

uplift on the bolts F Q<sub>1</sub>-Q<sub>a</sub>  
-934.24

uplift in one bolt  $\frac{F}{N}$

-77.854

tensile stress in one bolt S<sub>t</sub> -0.0765



check( $S_b > S_t$ )

SAFE

**Calculation of Shear Stress in Bolts**

Wind/seismic load in one

bolt

139.607

Shear stress

0.13722

kg/mm<sup>2</sup>

Since stress is within  
limit, Design is Safe

Therefore the Dimensions of Saddle Are **Safe** for all **Loads**

## 4.4 Design of Manway

### 4.3 Design of nozzle

All formulas are from ASME SECTION VIII DIVISION 1 UG-27 (c),UG-45

Material selection from ASME 2011a SECTION II PART A

#### Manway

##### Shell Inputs:

Material :	SA516-70		calculation
S:Allowable stress:	18500	psi	data sheet
Thickness of shell:	0.5	inch	
$t_r$ :Minimum required thickness of shell:	0.182	inch	
c.a :Corrosion allowance:	3	mm	
	0.118	inch	

##### Nozzle Inputs:

Material:	SA516-70		
S :Allowable stress:	18500	psi	
P :Design Pressure	39	psi	
E :Weld efficiency:	1		
Mill tolerance:	12.50%		
fr:Ratio of Allowable Stress	$S_s/S_n$		
	100.00%		pipe schedule
c.a :Corrosion allowance:	3	mm	
	0.118	inch	pipe schedule
$D_o$ :Outside diameter	609.6	mm	
	<b>24.000</b>	inch	

##### Calculating thickness of nozzle

Thickness of nozzle:	$n_t$	<b>9.53</b>	mm	pipe schedule
		0.375	inch	
Internal radius:	$R_i$	$D_o/2-n_t+ca$	inch	
		<b>11.743</b>	inch	
Thickness considering mill tolerance:	$t_b$	$ut*n_t+ca$		
		<b>0.165</b>	inch	

Thickness considering design pressure:

$$t_{a1} = \frac{P * R_i}{2 * S * E + 0.4 * P} + c.a$$

**0.130** inch

$$t_{a2} = \frac{P * R_i}{S * E - 0.6 * P} + c.a$$

**0.143** inch

**Now checking for required thickness**

Required thickness:  $t_{req} = \min(t_b, \max(t_{a1}, t_{a2}))$

**0.143** inch

Thickness based on mill tolerance:  $t$

**0.328** inch

**Minimum Area Required**

Required Area Calculations:

Thickness considering corrosion:  $t_n$

nt-ca  
**0.257** inch

Internal diameter:  $D_i$

$D_o - 2 * t_n$   
**23.486** inch

Required area:  $A$

$D_i * t_r * f_r + 2 * t_n * t_r * 1 * (1 - f_r)$   
**4.274** inch<sup>2</sup>

Since required thickness is less than given thickness, Design is Safe.

**Check for flange rating:**

For 400°C selected flange can sustain 200 psi.

Our operating temperature and pressure is 340°C and 39 psi.

Since, operating pressure is less than allowable pressure, hence it is safe.

## 4.5 Design of Charge Inlet

All formulas are from ASME SECTION VIII DIVISION 1 UG-27 (c),UG-45

Material selection from ASME 2011a SECTION II PART A

### Charge Inlet

#### Shell Inputs:

Material :	SA516-70		calculation
S:Allowable stress:	18500	psi	data sheet
Thickness of shell:	0.5	inch	
$t_r$ Minimum required thickness of shell:	0.182	inch	
c.a :Corrosion allowance:	3	mm	
	0.118	inch	

#### Nozzle Inputs:

Material:	SA516-70		
S :Allowable stress:	18500	psi	
P :Design Pressure	39	psi	
E :Weld efficiency:	1		
Mill tolerance:	12.50%		
fr:Ratio of Allowable Stress	$S_s/S_n$		
	100.00%		pipe schedule
c.a :Corrosion allowance:	3	mm	
	0.118	inch	pipe schedule
$D_o$ :Outside diameter	60.33	mm	
	<b>2.375</b>	inch	

#### Calculating thickness of nozzle

Thickness of nozzle:	$n_t$	<b>3.91</b>	mm	pipe scheule
		0.154	inch	
Internal radius:	$R_i$	$D_o/2-n_t+ca$	inch	
		<b>1.152</b>	inch	
Thickness considering mill tolerance:	$t_b$	$ut*n_t+ca$		
		<b>0.137</b>	inch	
Thickness considering design pressure:	$t_{a1}$	$\frac{P * R_i}{2 * S * E + 0.4 * P} + c.a$		
		0.119	inch	
	$t_{a2}$	$\frac{P * R_i}{S * E - 0.6 * P} + c.a$		
		0.121	inch	

**Now checking for required thickness**

Required thickness:  $t_{req}$   $\min(t_b, \max(t_{a1}, t_{a2}))$   
**0.121** inch

Thickness based on mill tolerance:  $t$  **0.135** inch

**Minimum Area Required**

Required Area Calculations:

Thickness considering corrosion:  $t_n$   $t - ca$   
**0.036** inch

Internal diameter:  $D_i$   $D_o - 2 * t_n$   
**2.304** inch

Required area:  $A$   $D_i * t_r * f_r + 2 * t_n * t_r * 1 * (1 - f_r)$   
**0.419** inch<sup>2</sup>

Since required thickness is less than given thickness, Design is Safe.

**Check for flange rating:**

For 400<sup>0</sup>C selected flange can sustain 200 psi.

Our operating temperature and pressure is 340<sup>0</sup>C and 39 psi.

Since, operating pressure is less than allowable pressure, hence it is safe.

## 4.6 Outlet

All formulas are from ASME SECTION VIII DIVISION 1 UG-27 (c),UG-45

Material selection from ASME 2011a SECTION II PART A

### Charge outlet

#### Shell Inputs:

Material :	SA516-70		calculation
S:Allowable stress:	18500	psi	data sheet
Thickness of shell:	0.5	inch	
tr: minimum required thickness of shell:	0.182	inch	
c.a :Corrosion allowance:	3	mm	
	0.118	inch	

#### Nozzle Inputs:

Material:	SA516-70		
S :Allowable stress:	18500	psi	
P :Design Pressure	39	psi	
E :Weld efficiency:	1		
Mill tolerance:	12.50%		
fr:Ratio of Allowable Stress	$S_s/S_n$		
	100.00%		pipe schedule
c.a :Corrosion allowance:	3	mm	
	0.118	inch	pipe schedule
D <sub>o</sub> :Outside diameter	88.9	mm	
	<b>3.500</b>	inch	

#### Calculating thickness of nozzle

Thickness of nozzle:	$n_t$	<b>5.49</b>	mm	pipe schedule
		0.216	inch	
Internal radius:	$R_i$	$D_o/2-n_t+ca$	inch	
		<b>1.652</b>	inch	
Thickness considering mill tolerance:	$t_b$	$ut*nt+ca$		
		<b>0.145</b>	inch	
Thickness considering design pressure:	$t_{a1}$	$\frac{P * R_i}{2 * S * E + 0.4 * P} + c.a$		
		0.120	inch	
	$t_{a2}$	$\frac{P * R_i}{S * E - 0.6 * P} + c.a$		
		0.122	inch	

**Now checking for required thickness**

Required thickness:  $t_{req}$   $\min(t_b, \max(t_{a1}, t_{a2}))$   
**0.122** inch

Thickness based on mill tolerance:  $t$  **0.189** inch

**Minimum Area Required**

Required Area Calculations:

Thickness considering corrosion:  $t_n$   $n-t-ca$   
**0.098** inch

Internal diameter:  $D_i$   $D_o - 2 * t_n$   
**3.304** inch

Required area:  $A$   $D_i * t_r * f_r + 2 * t_n * t_r * 1 * (1 - f_r)$   
**0.601** inch<sup>2</sup>

Since required thickness is less than given thickness, Design is Safe.

**Check for flange rating:**

For 400<sup>0</sup>C selected flange can sustain 200 psi.

Our operating temperature and pressure is 340<sup>0</sup>C and 39 psi.

Since, operating pressure is less than allowable pressure, hence it is safe.

## 4.7 Drain

4.3 Design of nozzle

All formulas are from ASME SECTION VIII DIVISION 1 UG-27 (c),UG-45

Material selection from ASME 2011a SECTION II PART A

### Drain

#### Shell Inputs:

Material :	SA516-70		calculation
S:Allowable stress:	18500	psi	data sheet
Thickness of shell:	0.5	inch	
$t_r$ :Minimum required thickness of shell:	0.182	inch	
c.a :Corrosion allowance:	3	mm	
	0.118	inch	

#### Nozzle Inputs:

Material:	SA516-70		
S :Allowable stress:	18500	psi	
P :Design Pressure	39	psi	
E :Weld efficiency:	1		
Mill tolerance:	12.50%		
$f_r$ :Ratio of Allowable Stress	$S_s/S_n$		
	100.00%		pipe schedule
c.a :Corrosion allowance:	3	mm	
	0.118	inch	pipe schedule
$D_o$ :Outside diameter	609.6	mm	
	<b>24.000</b>	inch	

#### Calculating thickness of nozzle

Thickness of nozzle:	$n_t$	<b>9.53</b>	mm	pipe schedule
		0.375	inch	
Internal radius:	$R_i$	$D_o/2-n_t+ca$	inch	
		<b>11.743</b>	inch	
Thickness considering mill tolerance:	$t_b$	$ut*n_t+ca$		
		<b>0.165</b>	inch	



Thickness considering design pressure:

$$t_{a1} = \frac{P * R_i}{2 * S * E + 0.4 * P} + c.a$$

0.130      inch

$$t_{a2} = \frac{P * R_i}{S * E - 0.6 * P} + c.a$$

0.143      inch

**Now checking for required thickness**

Required thickness:  $t_{req} = \min(t_b, \max(t_{a1}, t_{a2}))$

**0.143**      inch

Thickness based on mill tolerance:  $t$

**0.328**      inch

**Minimum Area Required**

Required Area Calculations:

Thickness considering corrosion:  $t_n = t - ca$

**0.257**      inch

Internal diameter:  $D_i = D_o - 2 * t_n$

**23.486**      inch

Required area:  $A = D_i * t_r * f_r + 2 * t_n * t_r * 1 * (1 - f_r)$

**4.274**      inch<sup>2</sup>

Since required thickness is less than given thickness, Design is Safe.

**Check for flange rating:**

For 400<sup>0</sup>C selected flange can sustain 200 psi.

Our operating temperature and pressure is 340<sup>0</sup>C and 39 psi.

Since, operating pressure is less than allowable pressure, hence it is safe.

## 4.8 Pressure Transmitter

All formulas are from ASME SECTION VIII DIVISION 1 UG-27 (c),UG-45

Material selection from ASME 2011a SECTION II PART A

### Charge outlet

#### Shell Inputs:

Material :	SA516-70		calculation
S:Allowable stress:	18500	psi	data sheet
Thickness of shell:	0.5	inch	
t <sub>r</sub> :Minimum required thickness of shell:	0.182	inch	
c.a :Corrosion allowance:	3	mm	
	0.118	inch	

#### Nozzle Inputs:

Material:	SA516-70		
S :Allowable stress:	18500	psi	
P :Design Pressure	39	psi	
E :Weld efficiency:	1		
Mill tolerance:	12.50%		
fr:Ratio of Allowable Stress	S <sub>s</sub> /S <sub>n</sub>		
	100.00%		pipe schedule
c.a :Corrosion allowance:	3	mm	
	0.118	inch	pipe schedule
D <sub>o</sub> :Outside diameter	60.33	mm	
	<b>2.375</b>	inch	

#### Calculating thickness of nozzle

Thickness of nozzle:	n <sub>t</sub>	<b>3.91</b>	mm	pipe schedule
		0.154	inch	
Internal radius:	R <sub>i</sub>	D <sub>o</sub> /2-n <sub>t</sub> +ca	inch	
		<b>1.152</b>	inch	
Thickness considering mill tolerance:	t <sub>b</sub>	ut*nt+ca		
		<b>0.137</b>	inch	
Thickness considering design pressure:	t <sub>a1</sub>	$\frac{P * R_i}{2 * S * E + 0.4 * P} + c.a$		
		0.119	inch	

$$t_{a2} = \frac{P * R_i}{S * E - 0.6 * P} + c.a$$

0.121 inch

**Now checking for required thickness**

Required thickness:  $t_{req} = \min(t_b, \max(t_{a1}, t_{a2}))$   
**0.121** inch

Thickness based on mill tolerance:  $t =$  **0.135** inch

**Minimum Area Required**

Required Area Calculations:

Thickness considering corrosion:  $t_n = t - c.a$   
**0.036** inch

Internal diameter:  $D_i = D_o - 2 * t_n$   
**2.304** inch

Required area:  $A = D_i * t_r * f_r + 2 * t_n * t_r * 1 * (1 - f_r)$   
**0.419** inch<sup>2</sup>

Since required thickness is less than given thickness, Design is Safe.

**Check for flange rating:**

For 400<sup>o</sup>C selected flange can sustain 200 psi.

Our operating temperature and pressure is 340<sup>o</sup>C and 39 psi.

Since, operating pressure is less than allowable pressure, hence it is safe.

## **Chapter 5**

### **Fabrication**

## **5.1 Order of fabrication**

- a) Beveling of metal plates.
- b) Rolling into shells.
- c) Welding into Longitudinal seam.
- d) Re-rolling.
- e) Welding into circumferential seam.
- f) Cutting slots for nozzles and fitments.
- g) Machining and grinding.
- h) Welding the nozzles and fitments.
- i) Attaching support to the vessel.

## **5.2 Fabrication procedure**

### **5.2.1 Fabrication of shell**

A shell is made from a metal plate of required dimensions and required material. The plate is cut to the required length and breadth and later on all four sides are bevel is prepared as per the drawing. The square cutting and the bevel preparation is carried out by oxyacetylene flame for carbon steels and plasma arc process for stainless and high alloy steels. The cut edges are then ground to sound metal by 1 to 1.5 mm so that all the adversely affected material in the cut zone is removed. The dressed are then examined for defects due to incorrect cutting parameters. If irregularities are found the surface is cleaned and again dressed.

The plate is now ready for bending; once the shell is fed to the bending machine, first both ends is pressed to the required shape by the rollers. This is called Pre-pinging. After pre-pinging, the profile at the ends is checked using a template, and if satisfactory, full bending is carried out. Usually, the bending is carried out in a few stages so that the elongation is negligible and no further trimming of plate is needed.

Now the shells after bending is joined to its ends by the process called longitudinal seam welding or L-seam welding. It is a butt weld with full penetration



**Figure 5.1 Shell Rolling**

The following sequence of operation is done for the joining of shells:

**a. Assembly**

1. Arrange the shells on the rollers or roller positioners.
2. Measure the circumference of the mating parts.
3. Grind the bevels of the mating parts to bright metal.
4. Mark the orientation lines on shell with respect to L-seam welds.
5. Set the shells together and match as per orientation.
6. Assemble and tack weld the shells from outside as per approved Welding Procedure Specification and maintain a uniform gap up to 2 mm maintaining tack weld length, weld size pitch.

### **b. Gouging**

Gouge out the clamps welded on the shell for assembly. Do not dig out the parent material.

### **Grinding**

1. Grind the clamps removed areas to smoothness.
2. Grind the bevel from inside to remove spatter penetrated due to tack welding from outside and sander 50 mm on either side of the joints

### **c. Manual Arc Welding**

Weld the joint between the shells with the two layers from inside to get 5 mm weld layer with approved WPS (Welding Procedure Sequence). De slag the top surface thoroughly.



**Figure 5.2 Manual Arc Welding**

### **d. Submerged Arc Welding**

Submerge arc weld the joint between shells with approved WPS and complete it.



**Figure 5.3 Submerged Arc Welding**

**e. Back Gouging**

Back gouge from outside to sound metal circumference.

**f. Grinding**

Grind the back gouge area from outside to sound metal circumference.

**g. Die Penetration Test**

Conduct D.P. test on Back gouge portion if required by inspection.





**Figure 5.4 Die Penetration Test**

**h. Welding**

Weld manually one layer on the joint between the shells with approved WPS and de-slag the top surface thoroughly.

**i. Submerged Arc Welding**

Submerge arc weld from outside the joint between the shells with approved WPS and de-slag the top surface thoroughly.

**j. Grinding**

Grind the circumferential weld from inside and outside to smoothness.

**k. Manual Arc Welding**

Rectify the spots by build-up on the joints between shells with approved WPS.

**l. Grinding**

Grind the buildup spots to smoothness.

**m. Die Penetration Test**

Conduct Die Penetration Test.

## n. Radiography

Conduct spot radiography.

For lethal service like acids or poisonous gas, 100% radiography is preferred.



**Figure 5.5 Radiography Test**

### 5.2.2 Fabrication of Dished ends

The manufacturing of elliptical dished ends is easier than that of hemispherical dished ends. The starting material is first pressed to a certain radius and then curled at the edge creating the skirt of dished end. Vessel dished ends can also be welded together from small pieces.



**Figure 5.6 Elliptical Head forming in Hydraulic press**

The axis on the shell is marked as specified in drawing. In order to align the end to shell, the center point and the four circumferential points on the head have to be marked.

This is done as follows:

1. At the straight face of the end, take the outer circumference and locate the four center points by dividing the circumference by 4.
2. Put the dished end upside-down on the thick leveled plate and place two tri-squares at opposite center-points marked. Stretch a chalked thread on top just grazing the top most point on the dished end to put an impression of the line. Repeat the same to 90° to the first line, which will result in a point that is the center of the dished end.
3. Connect the four center points representing 0°, 90°, 180° and 270° to the center of the dished end.
4. By the same method, any angle on the dished end can be located for the placement of nozzles or other attachments.

### **5.2.3 Fitting of Subassemblies**

The marking of all the attachments like the nozzles, manholes, and other fittings are preferably to be done simultaneously so that any fouling of these components with existing weld seams or among them could be checked. This is checked with respect to orientation plan elevation for vertical vessels and end view/ elevation for horizontal vessels.

Nozzles or manholes are installed on pressure vessels by cutting a hole in the side of the vessel and welding in an appropriately sized pipe to form the nozzle or manhole. These intersections ensure a "hole weakening" on the vessel, due to the metal removed and the stress concentration created. In critical systems, this weakness must be compensated, and can be restored with a Reinforcing Pad, to strengthen the piping branch connection or the pressure vessel nozzle.



**Figure 5.7 Man-way**



**Figure 5.8 Nozzles**

#### **5.2.4 Attaching support to the vessel**

The wear plate of a saddle is welded at a location determined by the design with approved WPS and de-slag the surface after welding.



**Figure 5.9 Saddle Support**

# **Chapter 6**

## **Transportation**

## 6.1 Introduction to Transportation

The transportation of a pressure vessel by ship, barge, road, or rail will subject the vessel to one-time-only stresses that can bend or permanently deform the vessel if it is not adequately supported or tied down in the right locations. The shipping forces must be accounted for to ensure that the vessel arrives at its destination without damage. It is very frustrating for all the parties involved to have a load damaged in transit and to have to return it to the factory for repairs. The cost and schedule impacts can be devastating if a vessel is damaged in transit. Certain minimal precautions can avoid the costly mistakes that often lead to problems. Even when all precautions are made, however, there is still the potential for damage due to unforeseen circumstances involved in the shipping and handling process. Care should be taken to ensure that the size and location of the shipping saddles, tie-downs, or lashing are adequate to hold the vessel but not deform the vessel. Long, thin-walled vessels, such as tray columns, are especially vulnerable to these shipping forces. The important thing to remember is that someone must take the responsibility. The barge and rail people have their own concerns with regard to loading and lashing. These may or may not coincide with the concerns of the vessel designer. The shipping forces for ships, barges, trucks, and rail are contained in this procedure. Each method of transportation has its own unique load schemes and resulting forces. Barge shipping forces will differ from rail due to the rocking motion of the seas. Rail shipments, however, go around corners at high speed. In addition, rail forces must allow for the “humping” of rail cars when they are joined with the rest of the train. Ocean shipments have to resist storms and waves without breaking free of their lashings. Whereas horizontal vessels on saddles are designed for some degree of loading in that position, vertical vessels are not. The forces and moments that are used for the design of a vertical vessel assume the vessel is in its operating position.

Vertical vessels should generally be designed to be put on two saddles, in a horizontal position, and transported by various means. That is the purpose of this procedure. Too often the details of transportation and erection are left in the hands of people who, though well versed in their particular field, are not pressure vessel specialists. Often vessels are transported by multiple means. Thus there will be handling operations between each successive mode of transportation. Often a vessel must be moved by road to the harbor and then transferred to a barge or ship. Once it reaches its destination, it must be reloaded onto road or rail transport to the job site. There it will be offloaded and either stored or immediately erected. A final transport may be necessary to move the vessel to the location where it will be finally erected.

## 6.2 Shipping Saddle

The primary concern of the vessel designer is the location and construction of the shipping saddles to take these forces without overstressing or damaging the vessel. If saddles are to be relocated by the transporter, it is important that the new locations be reviewed. Generally only two shipping saddles should be used. However, this may not always be possible. Remember that the reason for using two saddles is that more than two saddles create a statically indeterminate structure. You are never assured that any given saddle is going to take more than its apportioned load. Here are some circumstances that would allow for more than two saddles to be used or for a special location of two saddles:

- i. Transporter objects due to load on tires.
- ii. Transporter objects due to load on barge or ship
- iii. Very thin, long vessel.
- iv. Heavy-walled vessels for spreading load on ship or transporters

Shipping saddles can be constructed of wood or steel or combinations. The saddles should be attached to the vessel with straps or bolts so that the vessel can be moved without having to reattach the saddle. Horizontal vessels may be moved on their permanent saddles but should be checked for the loadings due to shipping forces and clearances for boots and nozzles. Shipping saddles should have a minimum contact angle of  $120^\circ$ , just like permanent saddles. Provisions for jacking can be incorporated into the design of the saddles to allow loading and handling operations without a crane(s). Shipping saddles should be designed with the vessel and not left up to the transport company. In general, transportation and erection contractors do not have the capability to design shipping saddles or to check the corresponding vessel stresses for the various load cases. Whenever possible, shipping saddles should be located adjacent to some major stiffening element. Some common stiffening elements include stiffening rings, heads (both internal and external), or cones. If necessary, temporary internal spiders can be used and removed after shipment is complete.



**Key factors for shipping saddles to consider:**

- i. Included angle
- ii. Saddle width
- iii. Type of construction
- iv. Lashing lugs
- v. Jacking pockets
- vi. Method of attachment to the vessel
- vii. Overall shipping height allowable-check with shipper.

**6.3 Lashings**

Vessels are lashed to the deck of ships and barges. In like manner they must be temporarily fixed to railcars, trailers, and transporters. Lashing should be restricted to the area of the saddle locations. Vessels are held in place with longitudinal and transverse lashings. Lashings should never be attached to small nozzles or ladder or platform clips. In some cases, lashing may be attached to lifting lugs and base rings. Lashings should not exceed 45<sup>0</sup> from the horizontal plane.

**Other Factors to Consider:**

- i. Shipping clearances
- ii. Shipping orientation
- iii. Shipping route
- iv. Lifting orientation
- v. Type of transport
- vi. Water tight shipment for all water transportation
- vii. Escorts and permits
- viii. Shipping plan

## **6.4 Organizations That Have a Part in the Transportation and Handling of Pressure Vessels:**

- i. Vessel fabricator
- ii. Transport company
- iii. Engineering contractor
- iv. Railway authorities
- v. Port authorities
- vi. Erection company
- vii. Trailer/transporter manufacturer
- viii. Ship or barge captain
- ix. Crane company

### **Outline of Methods of Vessel Shipping and Transportation**

#### **Road**

- i. Truck/tractor and trailer
- ii. Special bulldozer
- iii. Frame adapters
- iv. Beams to span trailers or transporters
- v. Rollers

#### **Rail**

- i. Single car
- ii. Multiple cars
- iii. Special cars

#### **Barge**

- i. River barge
- ii. Ocean going barge

## **Ships**

- i. Roll on, Roll off type
- ii. Loading and off-loading capabilities
- iii. In-hull or on deck
- iv. Floating cranes

## **Chapter 7**

### **Finite Element Analysis**

## **7.1 Introduction**

It is not always possible to obtain the exact analytical solution at any location in the body, especially for those elements having complex shapes or geometries. Always matters are the boundary conditions and material properties. In such cases, the analytical solution that satisfies the governing equation or gives extreme values for the governing functional is difficult to obtain. Hence for most of the practical problems, the engineers resort to numerical methods like the finite element method to obtain approximate but most probable solutions.

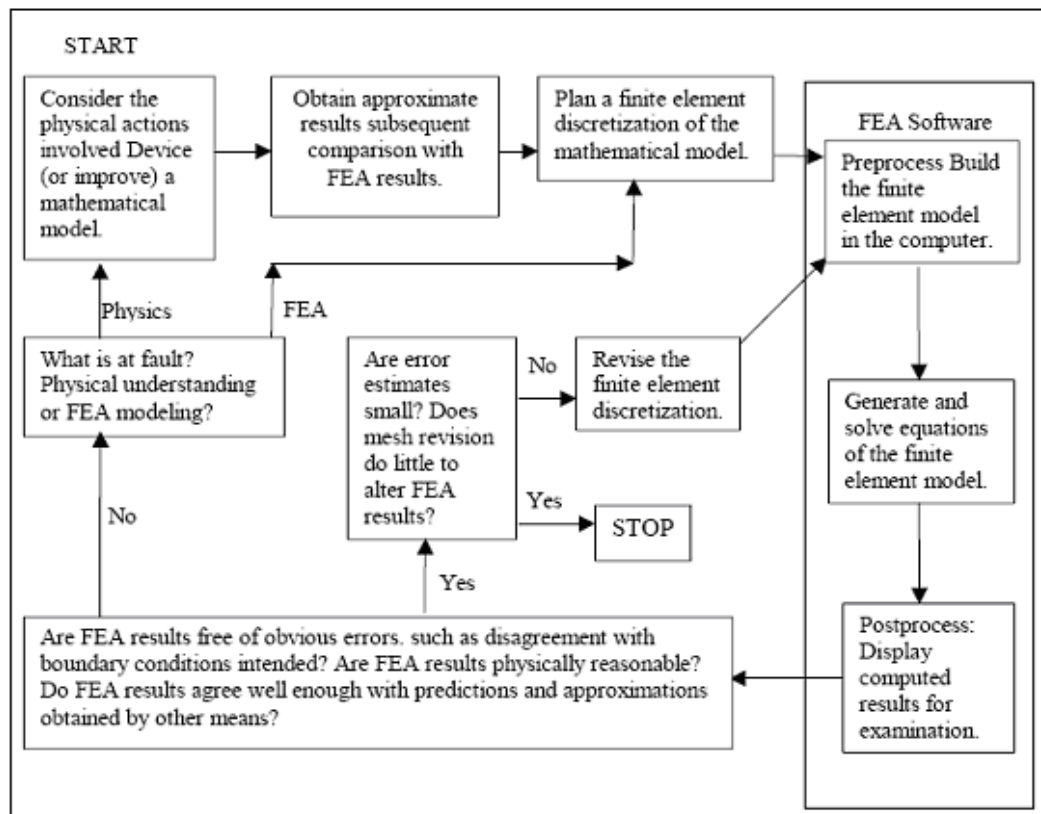
Finite element procedures are at present very widely used in engineering analysis. The procedures are employed extensively in the analysis of solids and structures and of heat transfer and fluids, and indeed, finite element methods are useful in virtually every field of engineering analysis.

## **7.2 Description of the method**

In any analysis we always select a mathematical model of a physical problem, and then we solve that model. Although the finite element method is employed to solve very complex mathematical models, but it is important to realize that the finite element solution can never give more information than that contained in the mathematical model.

## **7.3 Physical problems, mathematical models, and the finite element solution**

The physical problem typically involves an actual structure or structural component subjected to certain loads. The idealization of the physical problem to a mathematical model requires certain assumptions that together lead to differential equations governing the mathematical model. The finite element analysis solves this mathematical model. Since the finite element solution technique is a numerical procedure, it is necessary to assess the solution accuracy. If the accuracy criteria are not met, the numerical solution has to be repeated with refined solution parameters (such as finer meshes) until a sufficient accuracy is reached.



**Figure 7.1 Finite Element Process**

## 7.4 Steps followed in ANSYS Program:

The three important steps in ANSYS programming are:

- a) **Pre-processing:** This phase consists of making available the input data such as geometry, material properties, meshing of the model, boundary conditions and has the following steps:
- b) **Solution:** In this phase a solver is used to solve the basic equation for the analysis type and to compute the results. This phase is taken care by the software program. In the solution process, the solver goes through following steps to compute the solution for a steady state analysis,
- c) **Post-processing:** This is the phase where the results are reviewed for the analysis done, by obtaining graphic displays, vector-plots and tabular reports of stress and displacement, etc.

## 7.5 Mesh generation

After validation of the model next step is generation of Finite Element Mesh. A very fine mesh creates the hardware space problem because the computations become voluminous. As the number of nodes increases, the total degree of freedom of the model increases. Hence a designer has to model it optimally i.e. placing fine mesh only at critical area and coarse mesh at other. So that run times less and also the accuracy is not much affected.

### 7.5.1 Mesh Refinement

After generation of coarse mesh, it is refined as per the geometry and critical sections of the model. It can be refined in three different ways as follows <sup>[13]</sup>.

- h- Refinement: Here element size is changed (decreased) without changing the element type. The h refinement is used near the fillet area.
- p- Refinement: Here element type is changed (to higher order) without changing element size. The p – refinement converges to the solution faster than h – refinement.
- r – Refinement: Here existing nodes are moved without changing the element type and size. The ‘r’ refinement used at other locations.

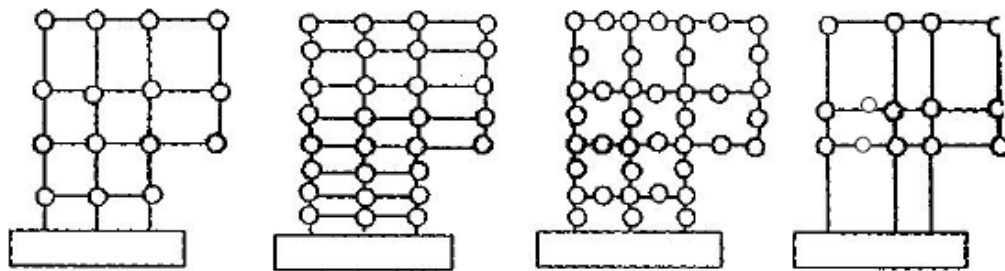
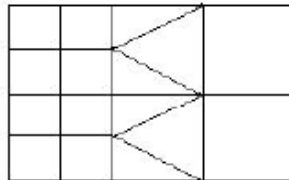


Figure 7.2 Different ways of mesh refinement

### 7.5.2 Mesh Transition:

Mesh transition occurs when refined mesh interfaces with coarse mesh. It connects different types of elements. One common method of performing a transition is to use an intermediate belt of different elements as shown in figure



**Figure 7.3 Mesh Transition**

## 7.6 Mesh Creation

In order to carry out a finite element analysis, the model we are using must be divided into a number of small pieces known as finite element. Since the model is divided into a number of discrete parts, FEA can be described as discretization technique. In simple terms, a mathematical net or “mesh” is required to carry out a finite element analysis. If the system under investigation is 1D in nature, we may use line elements to represent our geometry and to carry out our analysis. If the problem can be described in two dimensions, then a 2D mesh is required. Correspondingly, if the problem is complex and a 3D representation of the continuum is required, then we use a 3D mesh.

### 7.6.1 Area Meshing

Area elements can be triangular or quadrilateral in shape. The selection of the element shape and order is based on considerations relating to the complexity of the geometry and nature of the problem being modeled. Membrane elements don't have any thickness. As a consequence they have no bending stiffness; loads can only be carried in the element plane. Plate and shell elements are used to model thin-walled regions in 3D space.



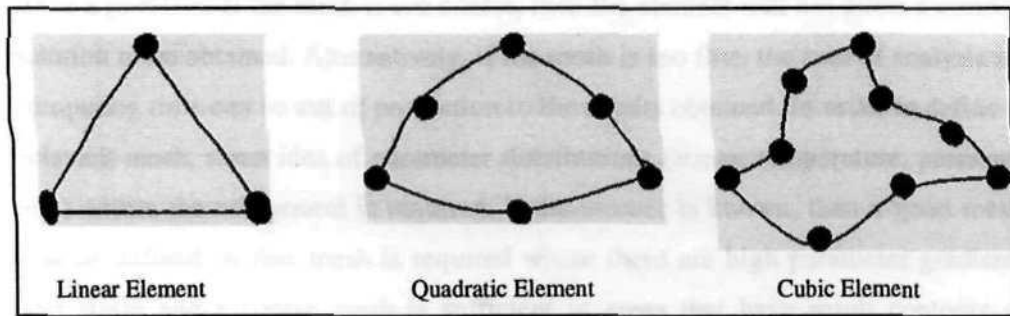


Figure 7.4 Two Dimensional Elements

:

### 7.6.2 Volume Meshing:

3D elements take the form of cubes called hexahedrons (hexes), 3D triangles called tetrahedrons (tests) and 3D wedges known as pentahedrons. Decisions on element selection hinge on understanding the role of the element shape and order of interpolation.

Modeling with 3D-Elements is the most flexible approach. These types of elements are used for thick structures that have neither a constant cross section nor an axis of symmetry. Solid modeling will nearly always make analysis preparation easier.

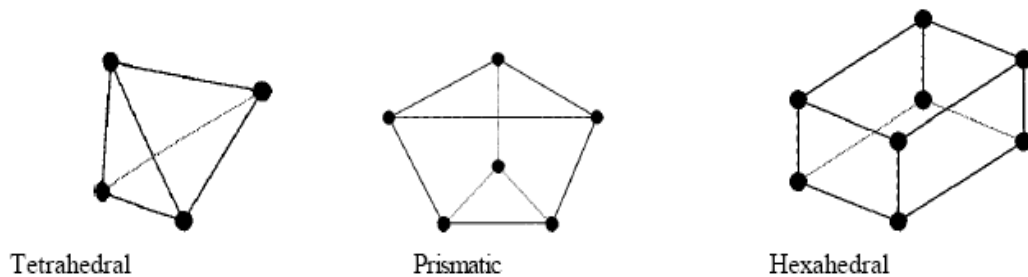


Figure 7.5 Three Dimensional Elements

### 7.6.3 Mesh Density

The art of using FEM lies in choosing the correct mesh density required to solve a problem. If the mesh is too coarse, then the element will not allow a correct solution to be obtained.

Alternatively, if the mesh is too fine, the cost of analysis in computing time can be out of proportion to the results obtained.

## 7.7 Ansys Report

### 7.7.1 Material Properties

Density	7.85e-006 kg mm <sup>-3</sup>
Coefficient of Thermal Expansion	1.2e-005 C <sup>-1</sup>
Specific Heat	4.34e+005 mJ kg <sup>-1</sup> C <sup>-1</sup>
Thermal Conductivity	6.05e-002 W mm <sup>-1</sup> C <sup>-1</sup>
Resistivity	1.7e-004 ohm mm
Compressive Ultimate Strength MPa	0
Compressive Yield Strength MPa	250
Tensile Yield Strength MPa	250
Tensile Ultimate Strength MPa	480

**Table 7.1 Material Properties**

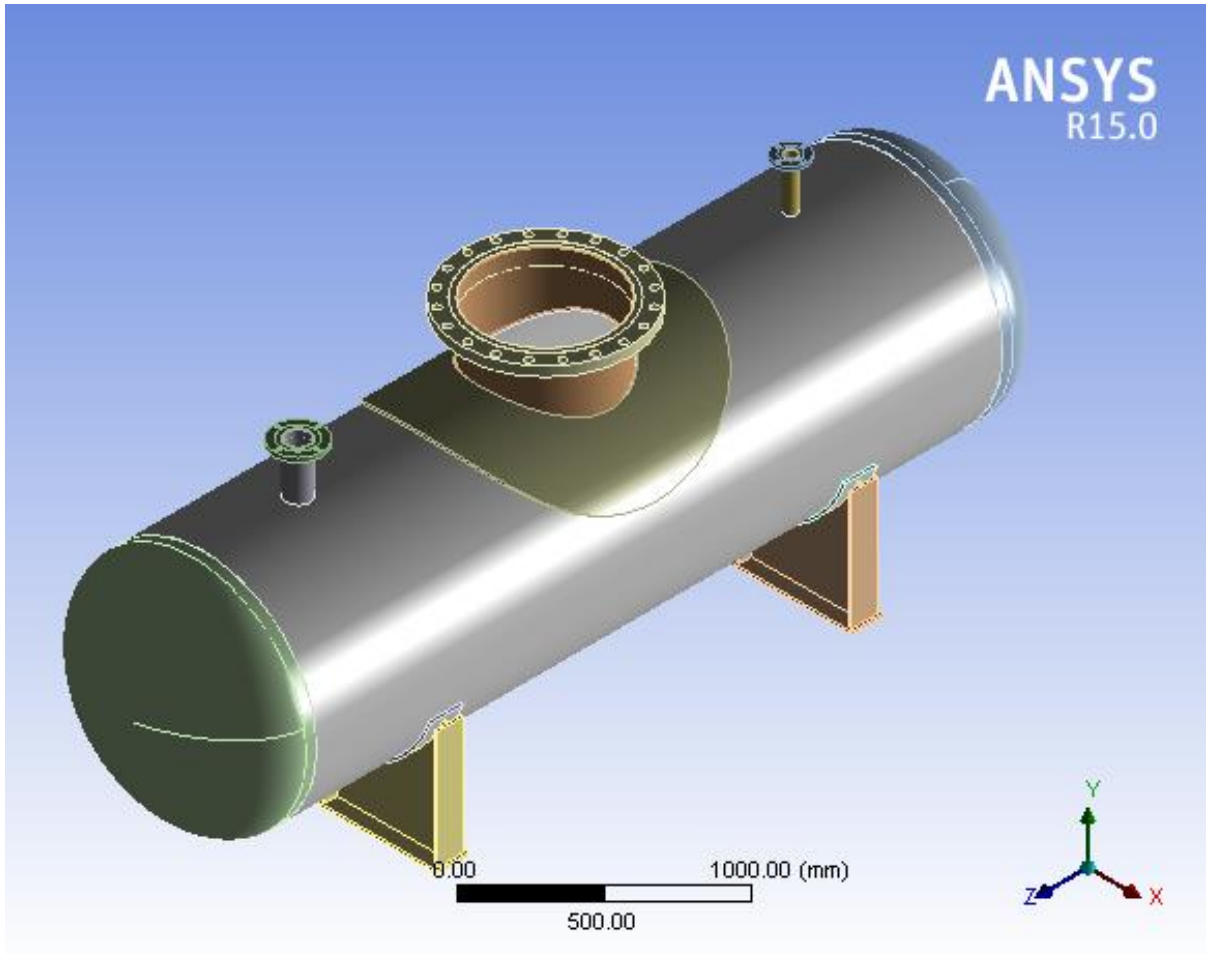
## 7.8 Conclusion

The Horizontal Pressure Vessel has been designed and analyzed and optimized successfully.

## 7.8.1 Model

Object Name	<i>Geometry</i>
State	Fully Defined
<b>Definition</b>	
Type	Design Modeler
Length Unit	Meters
Element Control	Program Controlled
Display Style	Body Color
<b>Bounding Box</b>	
Length X	1117.6 mm
Length Y	1593. mm
Length Z	3969.3 mm
<b>Properties</b>	
Volume	2.2762e+008 mm <sup>3</sup>
Mass	1786.8 kg
Scale Factor Value	1.
<b>Statistics</b>	
Bodies	16
Active Bodies	16
Nodes	109961
Elements	53461
Mesh Metric	None
<b>Basic Geometry Options</b>	
Parameters	Yes
Parameter Key	DS
Attributes	No
Named Selections	No
Material Properties	No

**Table 7.2 Model Details**

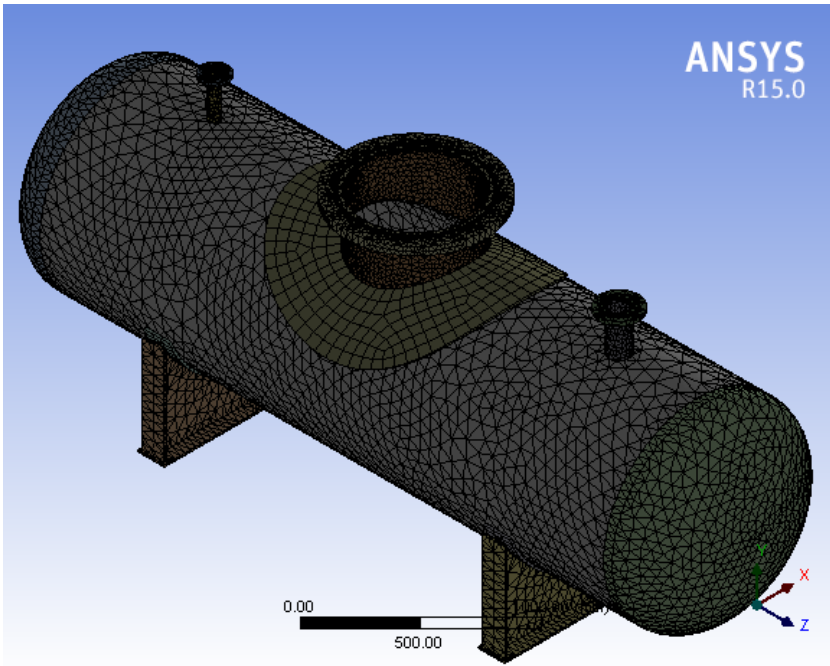


**Figure 7.6 Solid Model of Pressure Vessel**

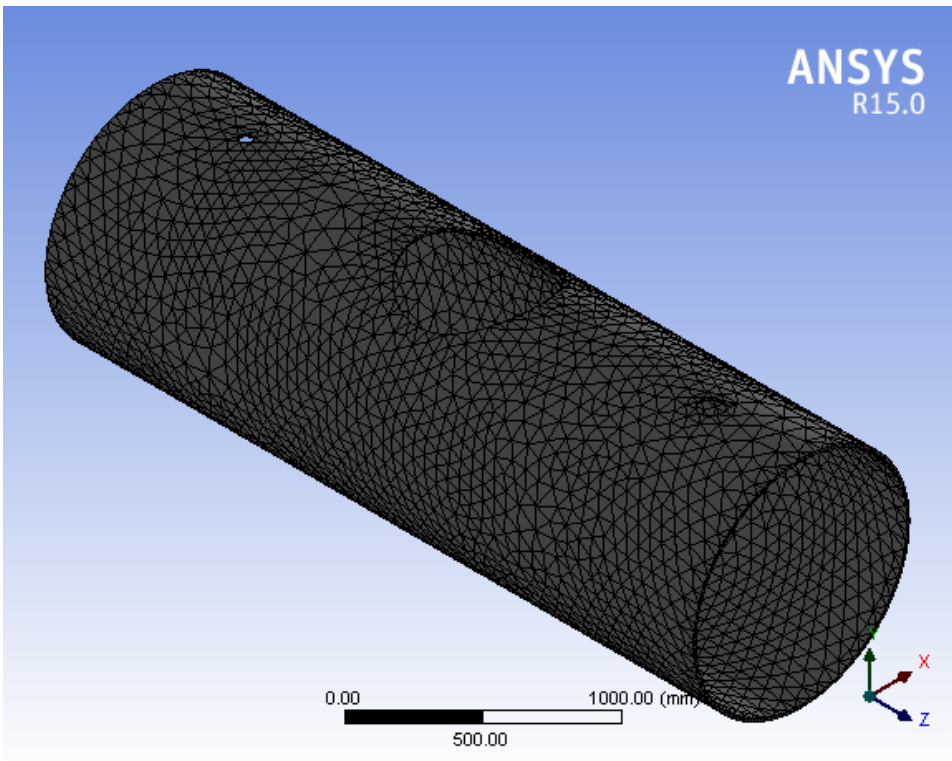
## 7.8.2 MESH

Object Name	<i>Mesh</i>
State	Solved
<b>Defaults</b>	
Physics Preference	Mechanical
Relevance	0
<b>Sizing</b>	
Use Advanced Size Function	Off
Relevance Center	Fine
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	3.680 mm
<b>Inflation</b>	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0.272
Maximum Layers	5
Growth Rate	1.2
Inflation Algorithm	Pre
View Advanced Options	No
<b>Patch Conforming Options</b>	
Triangle Surface Mesher	Program Controlled
<b>Patch Independent Options</b>	
Topology Checking	Yes
<b>Defeaturing</b>	
Pinch Tolerance	Please Define
Generate Pinch on Refresh	No
Automatic Mesh Based Defeaturing	On
Defeaturing Tolerance	Default
<b>Statistics</b>	
Nodes	109961
Elements	53461
Mesh Metric	None

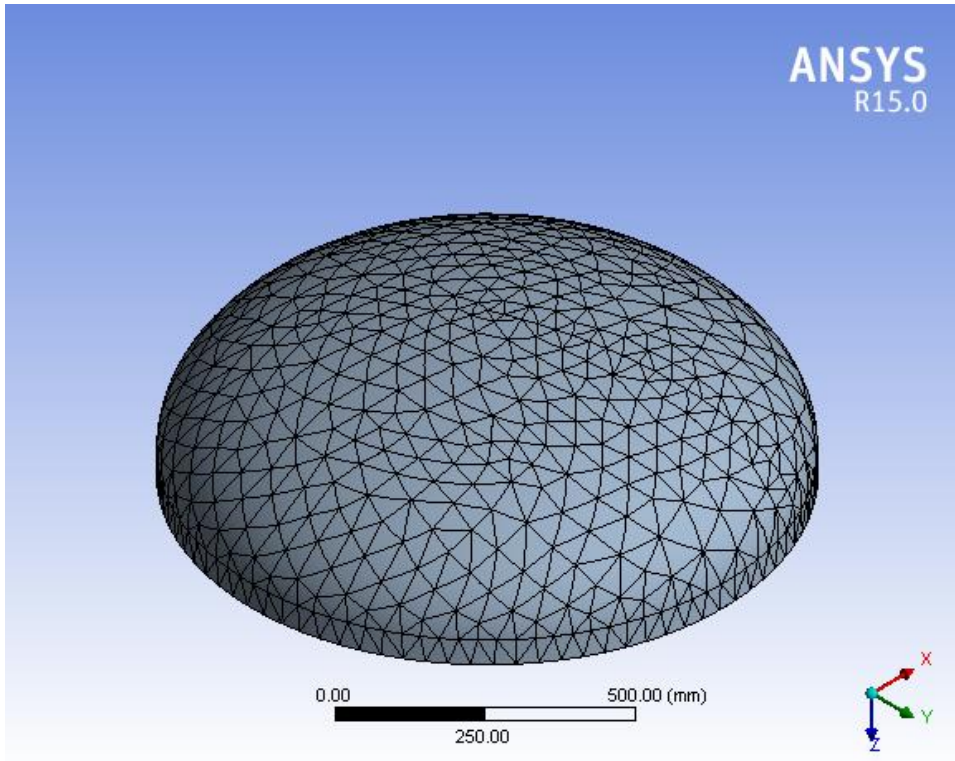
**Table 7.3 Mesh Details**



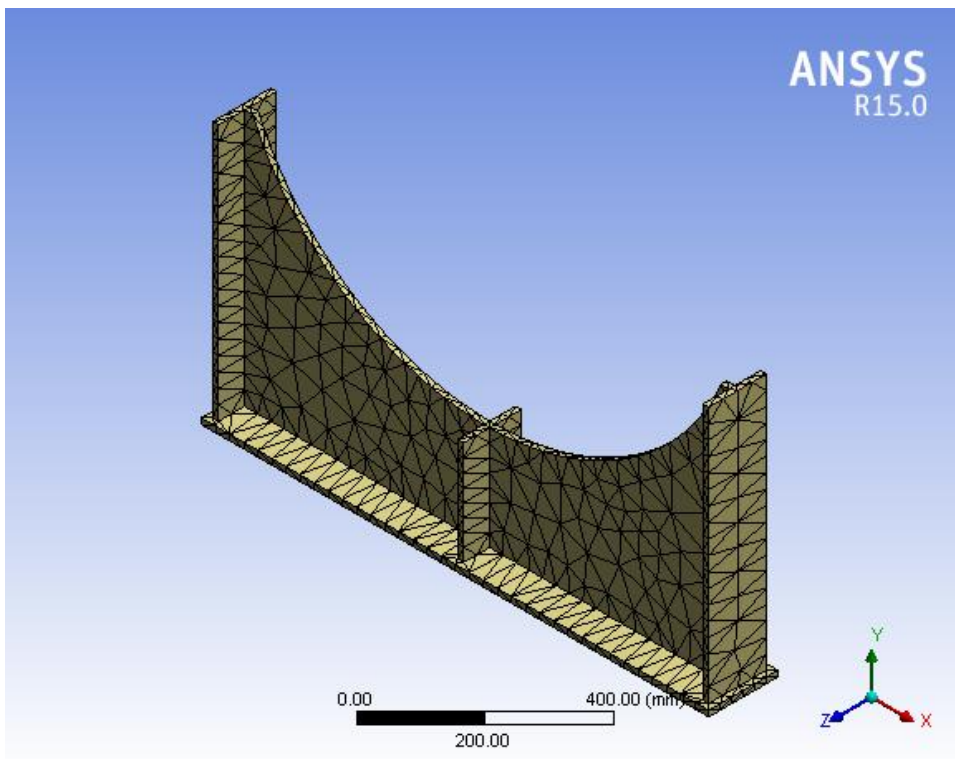
**Figure 7.7 Mesh View of Pressure Vessel**



**Figure 7.8 Mesh View of Shell**



**Figure 7.9 Mesh View of Head**

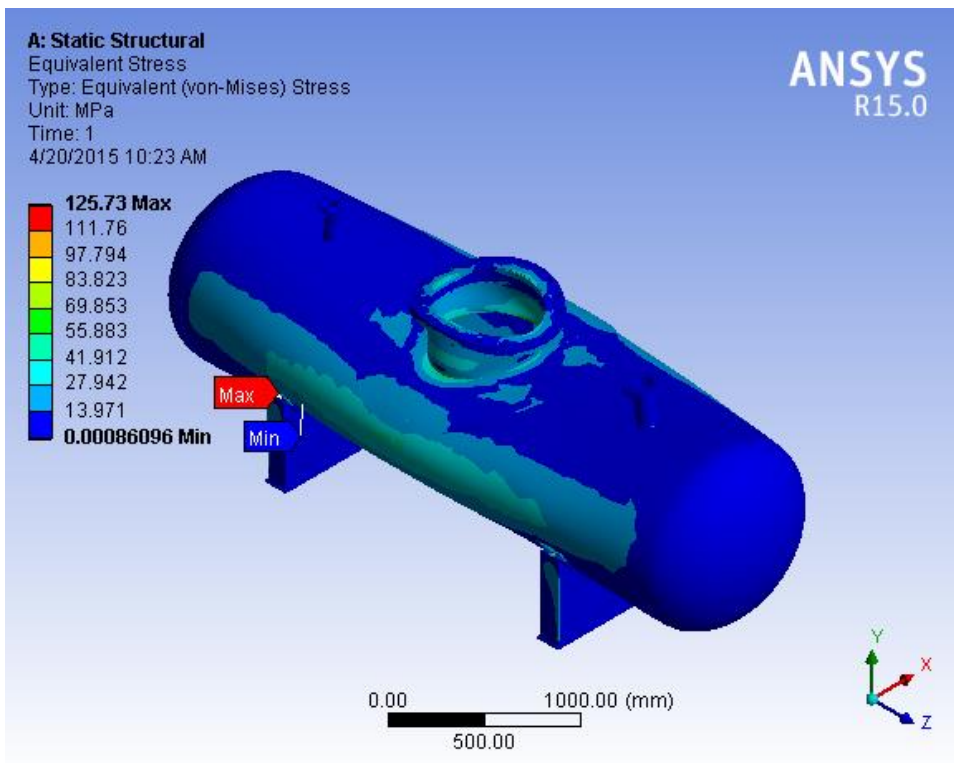


**Figure 7.10 Mesh View of Saddle**

### 7.8.3 Loading Condition And Results

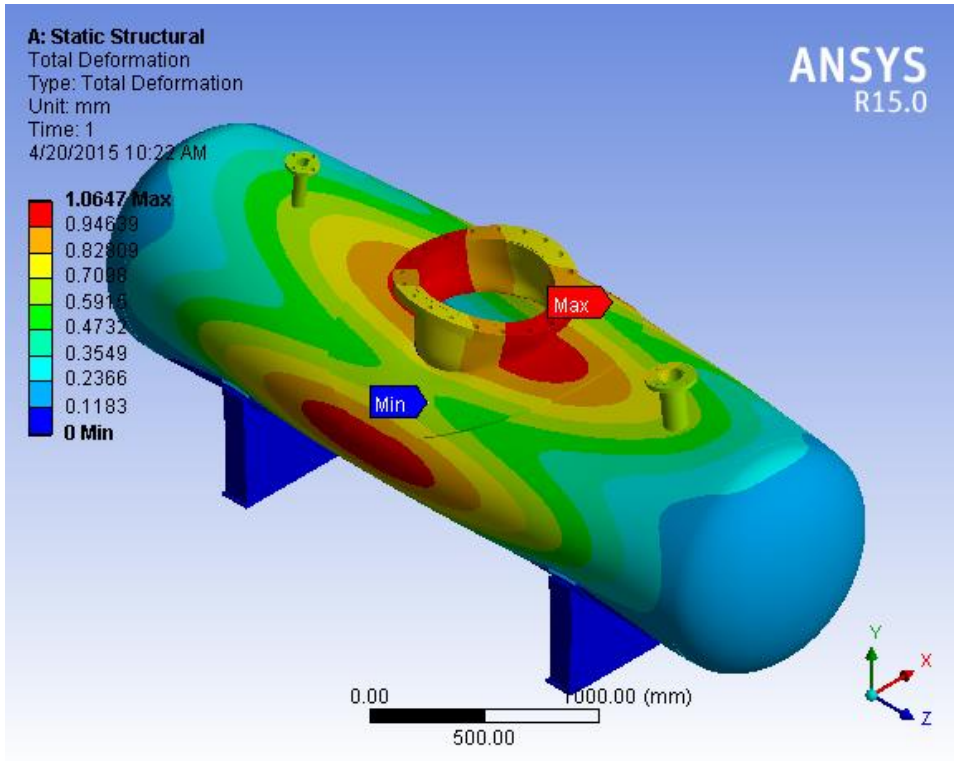
Object Name	<i>Fixed Support</i>	<i>Pressure</i>
State	Fully Defined	
<b>Scope</b>		
Scoping Method	Geometry Selection	
Geometry	2 Faces	8 Faces
<b>Definition</b>		
Type	Fixed Support	Pressure
Suppressed	No	
Define By	Normal To	
Magnitude	0.29 MPa (ramped)	

**Table 7.4 Loading Condition**



**Figure 7.11 Ansys Stress Result**





**Figure 7.12 Ansys Displacement Result**

Object Name	Total Deformation	Equivalent Stress
State	Solved	
<b>Scope</b>		
Scoping Method	Geometry Selection	
Geometry	All Bodies	
<b>Definition</b>		
Type	Total Deformation	Equivalent (von-Mises) Stress
By	Time	
Display Time	Last	
Calculate Time History	Yes	
Suppressed	No	
<b>Results</b>		
Minimum	0. mm	8.6096e-004 MPa
Maximum	1.0647 mm	125.73 MPa
Minimum Occurs On	Saddle:1	FlangeNozz40:1
Maximum Occurs On	Shell2:1	Saddle:1
<b>Minimum Value Over Time</b>		
Minimum	0. mm	8.6096e-004 MPa
Maximum	0. mm	8.6096e-004 MPa
<b>Maximum Value Over Time</b>		
Minimum	1.0647 mm	125.73 MPa
Maximum	1.0647 mm	125.73 MPa

**Table 7.5 Displacement and Maximum Stress Results**

## 7.9 Inventor Report

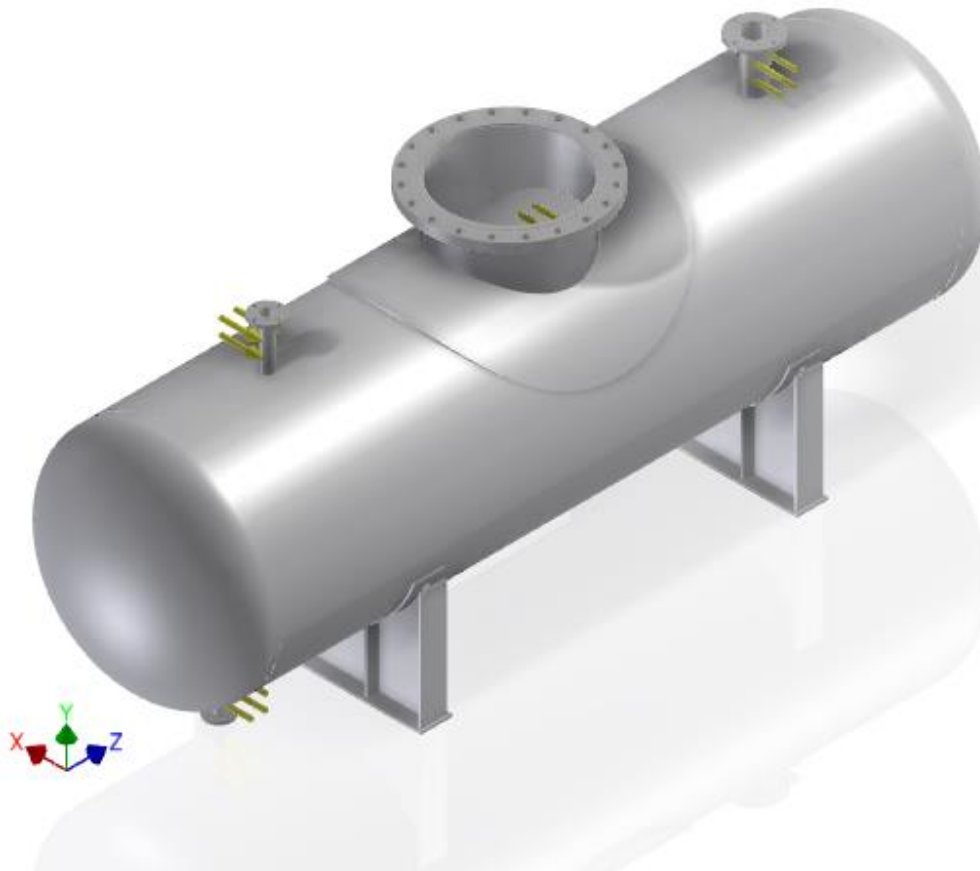
### 7.9.1 Material Properties & Solid Model

Name	Steel ASTM A514	
General	Mass Density	7.85 g/cm <sup>3</sup>
	Yield Strength	372.317 MPa
	Ultimate Tensile Strength	496.423 MPa
Stress	Young's Modulus	204.773 GPa
	Poisson's Ratio	0.29 ul
	Shear Modulus	79.3694 GPa
Part Name(s)	Shell2.ipt ManWay.ipt N40.ipt N50.ipt N100.ipt	
Name	Stainless Steel	
General	Mass Density	8 g/cm <sup>3</sup>
	Yield Strength	250 MPa
	Ultimate Tensile Strength	540 MPa
Stress	Young's Modulus	193 GPa
	Poisson's Ratio	0.3 ul
	Shear Modulus	74.2308 GPa
Part Name(s)	FlangeManWay.ipt FlangeNozz40.ipt FlangeNozz50.ipt FlangeNozz100.ipt	
Name	Stainless Steel	
General	Mass Density	8 g/cm <sup>3</sup>
	Yield Strength	250 MPa
	Ultimate Tensile Strength	540 MPa
Stress	Young's Modulus	193 GPa
	Poisson's Ratio	0.3 ul
	Shear Modulus	74.2308 GPa
Part Name(s)	FlangeManWay.ipt FlangeNozz40.ipt FlangeNozz50.ipt FlangeNozz100.ipt	

**Table 7.6 Material Properties**

Mass	1792.12 kg
Area	36546600 mm <sup>2</sup>
Volume	228206000 mm <sup>3</sup>
Center of Gravity	x=159815 mm y=-16645.1 mm z=-139453 mm

**Table 7.7 Physical Properties and CG**



**Figure 7.13 Solid Model**

## 7.9.2 Mesh

Avg. Element Size (fraction of model diameter)	0.05
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	No
Use part based measure for Assembly mesh	Yes

**Table 7.8 Mesh View**

Nodes:322441  
Elements:164061

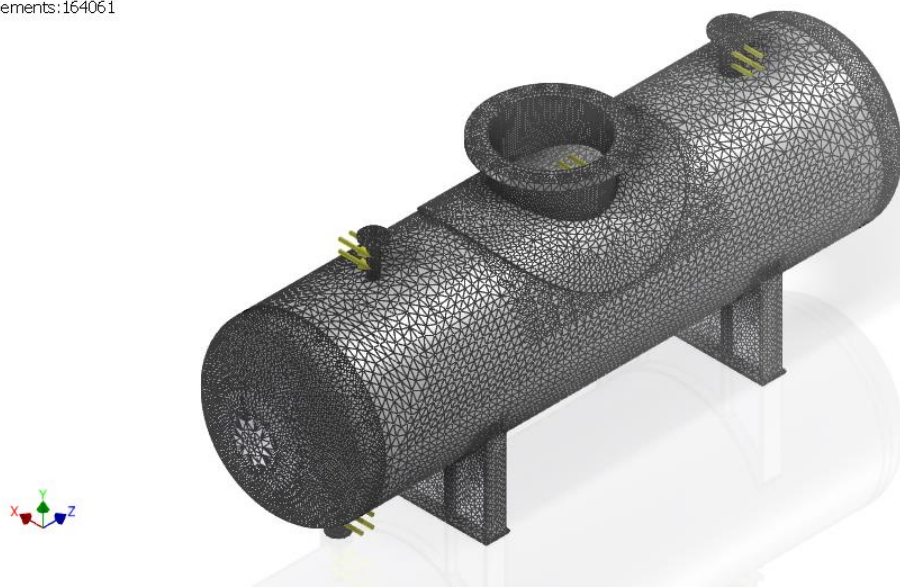


Figure 7.14 Mesh View of Pressure Vessel

Nodes:55797  
Elements:28035



Figure 7.15 Mesh View of Shell

Nodes:29469  
Elements:14588



Figure 7.16 Mesh View of Head

Nodes:8844  
Elements:4269

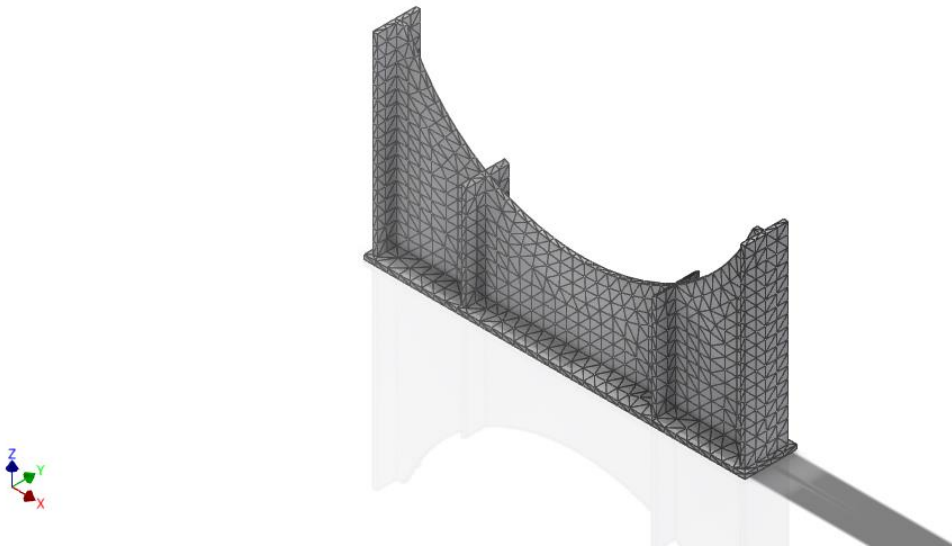


Figure 7.17 Mesh View of Saddle

### 7.9.3 Loading Conditions & Result

Load Type	Pressure
Magnitude	0.269 MPa

Table 7.9 Loading Condition

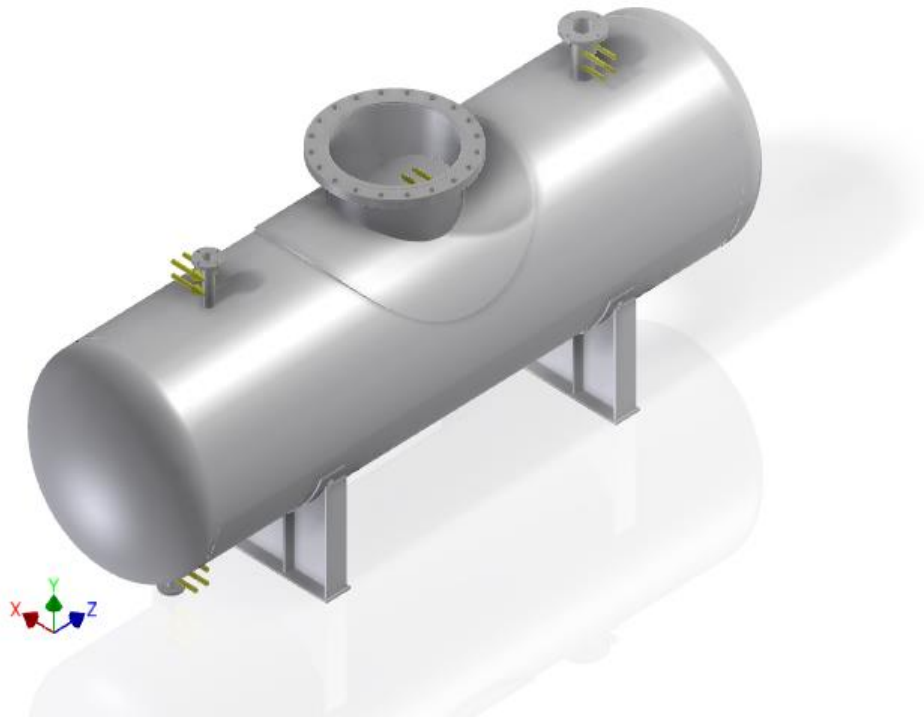


Figure 7.18 Application of Load

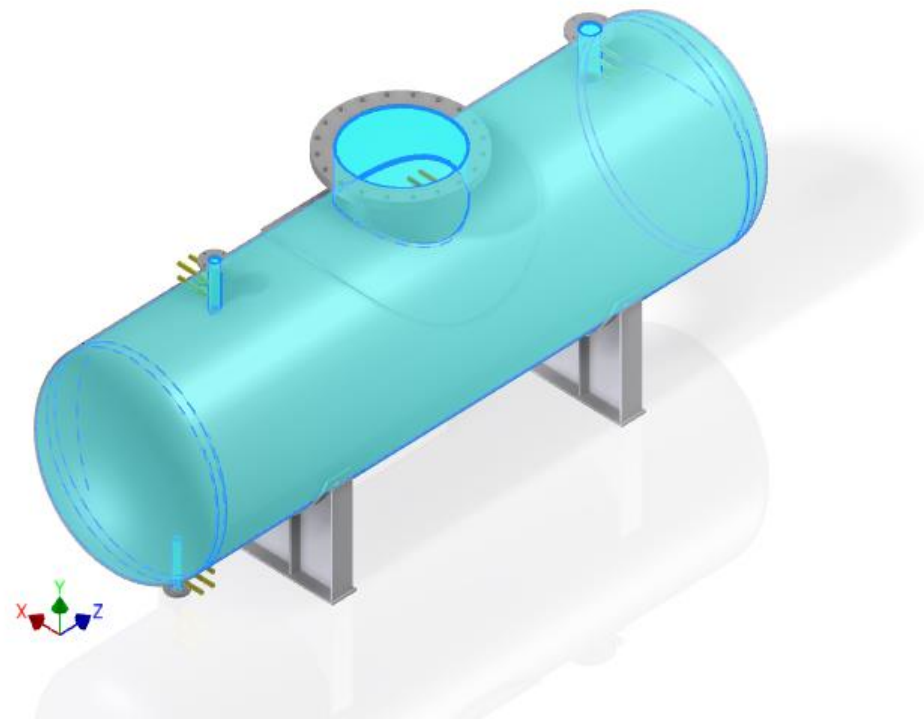


Figure 7.19 Fixed At Walls

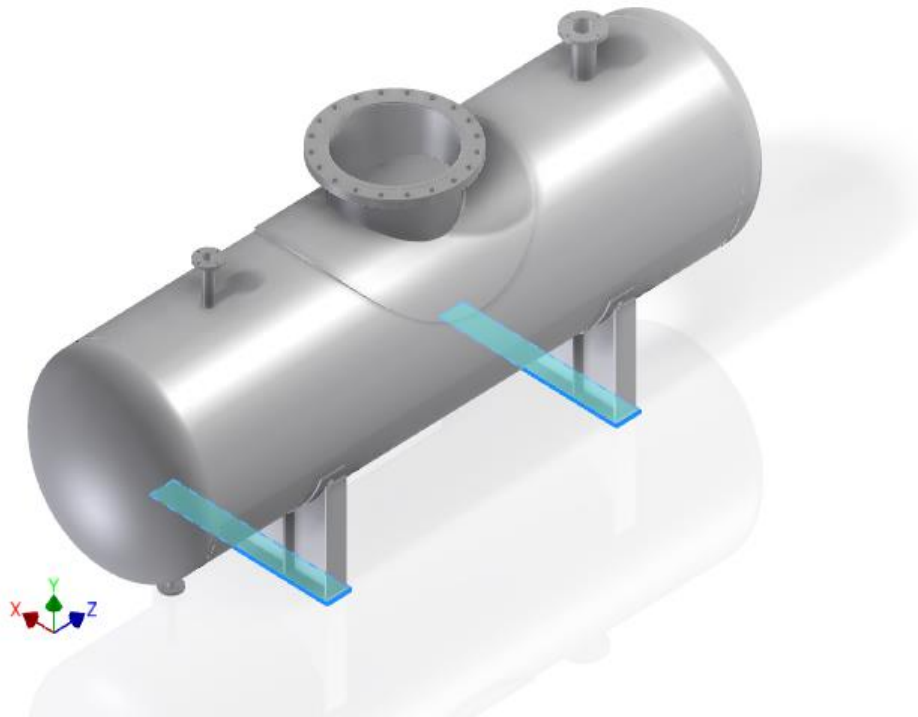


Figure 7.20 Fixed At Base

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint: 1	98689.9 N	0 N	2953.3 N m	-2953.3 N m
		98689.9 N		0 N m
		0 N		0 N m
Von Mises Stress	0.000379358 MPa	85.2419 MPa		
1st Principal Stress	-16.8692 MPa	74.2855 MPa		
3rd Principal Stress	-88.0942 MPa	11.4579 MPa		
Displacement	0 mm	1.09037 mm		
Safety Factor	4.36777 ul	15 ul		

Table 7.10 Results



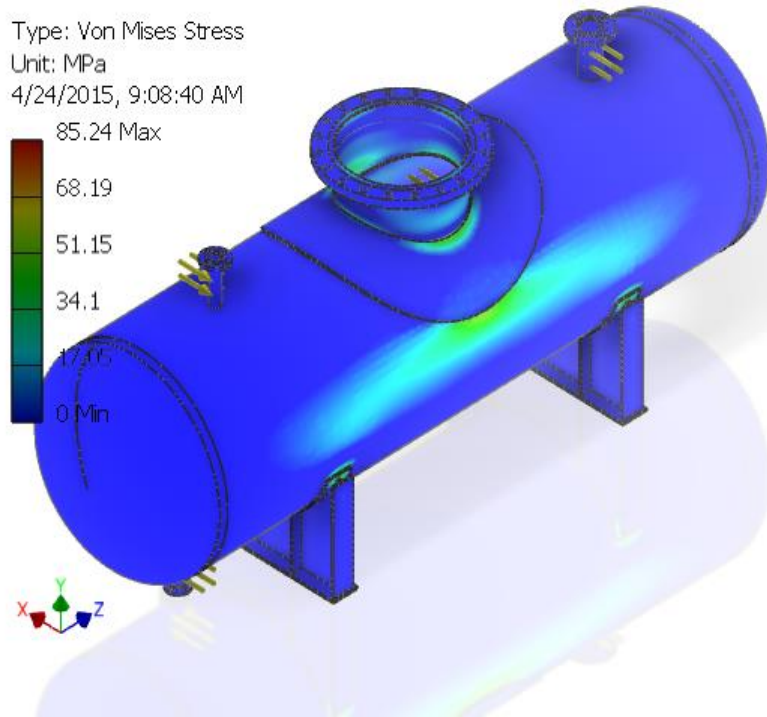


Figure 7.21 Von Mises Stress

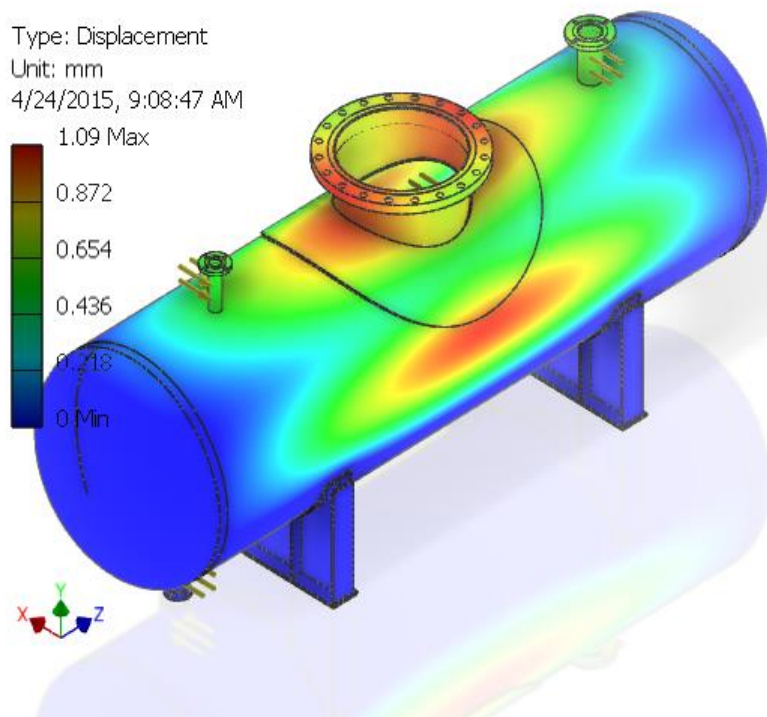


Figure 7.22 Displacement

## **Chapter 8**

### **Result & Conclusion**

## 8.1 RESULT

1. The Pressure Vessel has been designed successfully according to ASME Codes for desired Design Pressure and Temperature.
2. The Saddle has been designed to withstand Seismic and Wind Loads.
3. The Pressure Vessel has been analyzed in ANSYS and Autodesk Inventor software and the results verify that the Pressure Vessel is safe for operation.
4. Different materials and the effects of their yield strength on the thickness of the vessel and the cost of these materials has been calculated and the most appropriate material has been selected for fabrication.

Material	Yield Strength Mpa	Minimum Thickness inch
SA 516-70	260	0.182
SA 515-60	260	0.185
SS 304	177	0.191
SS 316	455	0.162

## 8.2 Conclusion

The Horizontal Pressure Vessel has been designed and analyzed and optimized successfully.

## Acknowledgement

After the completion of this work, we would like to give our sincere thanks to all those who helped us to reach our goal. It's a great pleasure and moment of immense satisfaction for us to express my profound gratitude to our guide **Prof. Momin Mohammad Nafe** whose constant encouragement enabled us to work enthusiastically. His perpetual motivation, patience and excellent expertise in discussion during progress of the project work have benefited us to an extent, which is beyond expression.

We would also like to give our sincere thanks to **Prof. Zakir Ansari**, Head of Department, Project Co-Guide and Project coordinator from Department of Mechanical Engineering, Kalsekar Technical Campus, New Panvel, for their guidance, encouragement and support during the project.

I take this opportunity to give sincere thanks to **Mr. NARAYAN LAXMAN SHETTY**, Owner of "**S.Tech consultancy Services.**" for all the help rendered during the course of this work and their support, motivation, guidance and appreciation.

I am thankful to **Dr. A. R. Honnutagi** Principal, Kalsekar Technical Campus New Panvel, for providing an outstanding academic environment, also for providing the adequate facilities.

Last but not the least I would also like to thank all the staffs of Kalsekar Technical Campus (Mechanical Engineering Department) for their valuable guidance with their interest and valuable suggestions brightened us.

Ansari Mohammad Faiz

Bape Adib Yakub

Bichu Azeeb Irshad

Chaudhary Adnan Ahmed

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## **Appendix**

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- Appendix B Drawing Sheet
- Appendix C ASME 2011a Section II Part A
- Appendix D Flange Properties
- Appendix E ASME Flange Rating ASME B16.5

## **Abstract**

Pressure vessels are one of the most important equipment in the engineering world. Vessels are used in everyday life and on a very large scale. From a simple gas cylinder to huge distillation towers the world is full of Pressure Vessels. We cannot imagine our world without these Pressure Vessels. These Vessels have got great importance and diverse use in the engineering world. We have made an attempt to study these amazing engineering marvels, design, analyze and in a certain way optimize them. Our project is the design of a Horizontal Pressure Vessel. Our motive behind the selection of this project is to enter into the huge world of Pressure Vessels and to explore the various facets of a Pressure Vessel.