Design of Storage Tank for Hydraulic System in Cold Roll Skin Pass Mill

By

Rushang Contractor

12MMED03

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

MAY-2014

Design of Storage Tank for Hydraulic System in Cold Roll Skin Pass Mill

Major Project

Submitted in partial fulfillment of the requirements

for the degree of

Master of Technology in Mechanical Engineering

(Design Engineering)

By

Rushang Contractor

12MMED03

Guided By

Prof M M Chauhan

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

MAY-2014

Declaration

This is to certify that

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

ii) Due acknowledgement has been made in the text to all other material used.

Rushang Contractor

(12MMED03)

Undertaking for Originality of the Work

I, **Rushang Contractor, Roll No. 12MMED03**, give undertaking that the Major Project entitled **"Design of Storage Tank for Hydraulic System in Cold Roll Skin Pass Mill "** submitted by me, towards the partial fulfillment of the requirements for the degree of Master of Technology in Mechanical Engineering (Design Engineering) 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.

Signature of student

Date: ---------------------

Place: Nirma University,Ahmedabad.

Endorsed By

(Signature of Guide)

Certificate

This is to certify that the Major Project entitled **"Design of Storage Tank for Hydraulic System in Cold Roll Skin Pass Mill "** submitted by **Mr. Rushang Contractor (12MMED03)** towards the partial fulfillment of the requirements for the degree of **Master of Technology in Mechanical Engineering (Design Engineering) 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,to the best of my knowledge,haven't been submitted to any other university or institution for award of any degree or diploma.

Prof M M Chauhan
Guide and Assistant Professor,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmedabad-382481.

Dr R N Patel
Professor and Head,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University, Ahmedabad-382481.

Dr K Kotecha
Director,
Institute of Technology,
Nirma University,
Ahmedabad-382481.

Acknowledgement

To begin with, I am thankful to Mr.Vishnu Tripathi (Sr.Manager, Essar Steel Limited, Hazira, Surat) and Mr. Narendra Gupta (Sr.Manager, Essar Steel Limited, Hazira, Surat) for giving me the opportunity to carry out my dissertation in his department and for his constant support and encouragement.

I wish to express my sincerest gratitude to Prof. Mihir Chauhan (Assistant Professor, Mechanical Engineering Department, IT, NU) for his supervision, advice, and guidance. I am very thankful to Prof. S.J.Joshi (PG Co-ordinator/Design, Mechanical Engineering Department, IT, NU) and Dr.R.N.Patel (Head Of Department, Mechanical Engineering Department, IT, NU) who inspired and enriched my growth as a student, a researcher and a engineer want to be.

I am very thankful to Dr K Kotecha (Director, IT, NU) and all the faculty members of Mechanical Engineering Department and all my colleagues who have directly or indirectly helped me during this dissertation work.

I deeply thanks to my departmental group member at Essar Steel Limited, Hazira Mr.Chirag Patel, Mr. Narendran Mohanakrishnan and Mr.Prabhu Govindraj for the outside perspective to my work and their time company.

I would like to thank all the people whose names i did not include here, but they provided me with the necessary help and made it possible for me to write this thesis.

Finally, my parents deserve special mention here. My deep and lasting gratitude to my mother and father for their unconditional love, faith and support.

Rushang Contractor
(12MMED03)

Abstract

Pressure vessels and piping systems have received an extraordinary amount of attention in recent years. Even storage tanks at atmospheric pressure have been more heavily scrutinized by the Occupational Safety and Health Administration (OSHA,United States), the Environmental Protection Association (EPA,United States) and industries in general.

The design engineer is facing a multitude of adverbs when considering the best approach to a pressure vessel design, piping design or system design program to select the appropriate code, material, orientation, shape etc.

This project aims to design a two saddle supported horizontal cylindrical tank for hydraulic system in Cold Roll Skin Pass Mill at Essar Steel, Hazira.Here main function of tank is to, store and supply the oil to the system continuously.ASME section VIII div I is followed considering storage tank as low pressure pressure vessel and design is done using horizontal cylindrical shell with both end closed with ellipsoidal heads.Saddle design is referred from standard design available in pressure vessel handbook and modify the design for material saving.Both the tank and saddle is modeled in Creo Parametric 2.0 and finite element analyse of saddle is done in ANSYS 14.0.

Contents

List of Tables

List of Figures

Nomenclature

V	Volume (litre)
q	Flow rate (m^3/s)
v	Velocity (m/s)
L	Length (mm)
r	Inner radius of cylindrical shell/elliptical head/nozzle (mm)
R	Outer radius of cylindrical shell/elliptical head/nozzle (mm)
d	Inner Diameter of cylindrical shell/elliptical head/nozzle (mm)
D	Outer Diameter of cylindrical shell/elliptical head/nozzle (mm)
ρ	Density of fluid (kg/m^3)
P	Pressure (Mpa)
T	Temperature (°c)
t	Thickness(mm)
S	Allowable stress (Mpa)
E	Joint efficiency
f_{r1}	Allowable stress ratio for nozzle wall material to shell material.
t_n	Nozzle wall thickness(mm)
t_r	Required thickness of seamless shell (mm)
t_{rn}	Required thickness of seamless nozzle (mm)
t_i	Inward projection nozzle thickness (mm)
l	Exterior projection of nozzle in pressure vessel (mm)
h	Interior projection of nozzle in pressure vessel (mm)
A	Area required in reinforcement (mm^2)
A_1	Area available In Shell (mm^2)
A_2	Area available In Nozzle Projecting Outward (mm^2)
A_3	Area available In Inward Nozzle (mm^2)
a	Height of baffle (mm)
b	Width of baffle at free edge (mm)
E	Modulus of elasticity (MPa)
F_y	Allowable yield stress (MPa)
F_b	Allowable bending stress stress (MPa)
F	Force (N)
Q	Load (N)
S_n	Allowable stress for nozzle material (MPa)
S_v	Allowable stress for shell material (MPa)
ISHB	Indian standard column section H beam
ISJC	Indian standard junior channel
ISA	Indian standard angles

Chapter 1

Introduction

Hydraulics is the science of transmitting force and/or motion through the medium of a confined liquid.It is first used to elevating the stages of amphitheaters by the ancient Greeks.Seventeenth century scholars Pascal and Boyle explained the principles of hydraulics scientifically.The laws were discovered by the Pascal and Boyle regarding the effects of temperature and pressure on fluids confined area.

1.1 Basic Principles of Hydraulics

Pascals Law states that "pressure applied to a confined fluid at any point is transmitted undiminished throughout the fluid in all directions and acts upon every part of the confining vessel at right angles to its interior surfaces and equally upon equal areas ".

To better understand Pascals Law, lets use a bottle filled with liquid as an example. Lets consider the bottle has a one-square-inch opening. If 10 pounds of force is applied on the cork at the opening, this 10 pounds would be applied equally to all internal surface of the bottle. which can be expressed as 10 pounds per square inch or 10 psi, here 10 psi is the fluid pressure of the system.

Figure 1.1: Pascal's law statement [1]

First hydraulic press was built by a British mechanic named Joseph Bramah using pressure, force and a confined fluid in a lever-like system. A closed system provides a maximum mechanical advantage similar to that of a simple lever.

Bramah [1] discovered that if in a closed fluid system a small force exerted on a small cylinder it could balance a large force on a large cylinder. If one pound of force is applied to a one-square-inch cylinder it could balance 100 pounds of force on a 100-square-inch cylinder. So by this way it is possible to move a 100 pound weight using only one pound of force.

1.2 Components of Hydraulic System

Fluid: Main function of hydraulic fluid is to provide necessary lubrication and protection against corrosion.

Reservoir: It is a storage tank for the fluid and also act as heat dissipater.

Hydraulic Pump: It converts the mechanical energy into hydraulic energy.

Hydraulic Valves: Valves are used to control the direction, flow rate, pressure of the hydraulic fluid.

Actuator: Actuator converts hydraulic energy into mechanical energy to do work.Rotary hydraulic motor or a hydraulic cylinder is a example of actuator.

Filter: Filter purified the hydraulic oil by removing unwanted contaminants from the fluid.

Hose or Tubing Fluid Lines: It is used to transport hydraulic fluid within hydraulic system.

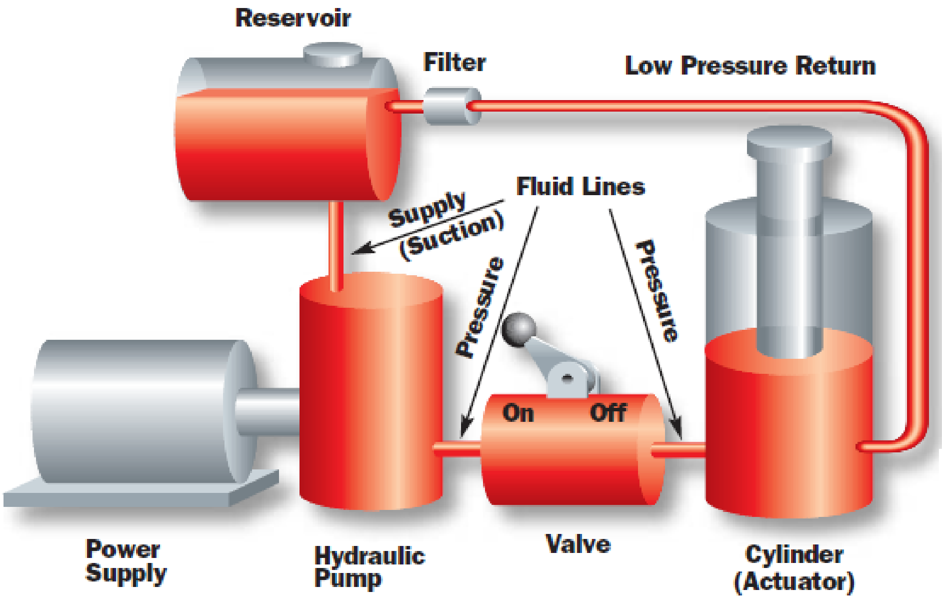

Figure 1.2: Components of hydraulic system [1]

1.3 Cold Roll Mill(CRM) at Essar Steel, Hazira

Steel sheet wound coils transferred from HRM (Hot roll mill) to CRM (Cold rolled mill) which is stands for finishing process on steel sheet. Basically five sections (1.Pickling, 2.Contineous tandem cold roll mill, 3. Batch annealing facility, 4.Skin pass mill, and 5. Galvanizing plant) are there to get the finished steel sheet with customer required properties. Different plants serve different function and as per the customer requirements this coils transferred from one plant to other plant using overhead crane.

1.3.1 Cold Roll Skin Pass Mill (CSPM)

CSPM is also called “work hardening ”intervenes after annealing and consists in giving a light lengthening to the band (from 0.3 to 2 % in general) by combination of an effort of rolling and traction.

Purpose of CSPM Plant

- To improve the mechanical characteristic of metal.
- To obtain correct flatness of sheet by action on the cambering and the choice of convex cylinders.
- To give to sheet its final aspect of surface : polished, glossed and rough.

CSPM Layout

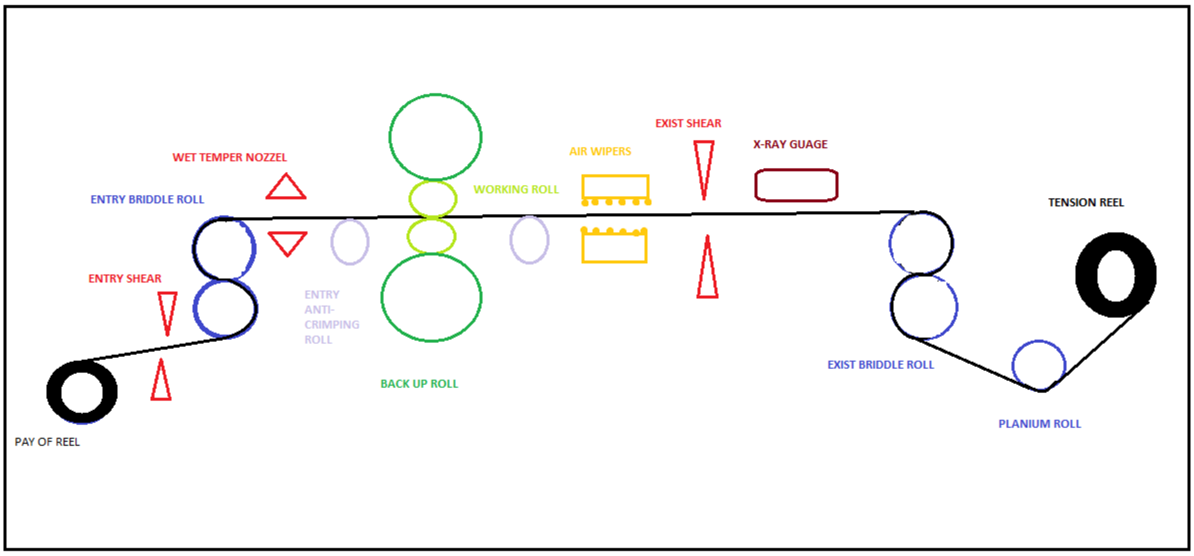

Figure 1.3: CSPM layout

Process

- Coil is transferred on cable drum with the help of entry walking beam.Cable drum is connected to shaft .

- Coil is uncoiling from left hand side and recoiling on the right hand side and in between all processes are done.
- Entry shear and exit shear provided to shear (cutting) the sheet as per the customer require length.
- Entry bridle roll and exit bridle rolls are there to keep tension in sheet .
- Wet temper nozzle is used to supply the oil continuously on the sheet. For reducing the friction between sheet and cylinder.
- The purpose of entry crimping roller and exit crimping roller are to tighten the sheet so that the sheet will not in influence of undulation or wrinkle.
- Working rolls and back rolls are mainly used to press (hydraulic tightening) sheet between them and also secure the flatness of sheet along the wiidth of sheet.
- Air wipers are used to throw away oil from the sheet.
- X-ray gauge is used to measure thickness continuously.

Table 1.1: Existing hydraulic system list

Sr. No.	Equipment	Quantity
1	10,000 Litre Hydraulic Tank	1
2	100 Bar Pump	7
3	200 Bar Pump	2
4	250 Bar Pump	2
5	Filtering Unit (10 Micron With 2 Micron)	1
6	Filtering Unit (25 Micron)	1
7	Accumulator Unit (4nos, Supply 100 Bar, P Max 210 Bar)	1
8	45 Bar Pilot Control System With Two Pumps	1
9	Valve Stands (1 Of 250 Bar, 1 Of 200 Bar, 5 No. 100 Bar)	7

1.4 Problem Faced with Existing System

At present in hydraulic system one 10,000 litre storage tank is there which act as a common reservoir for hydraulic system. Here tank supplies oil to three systems (100 bar system, 200 bar system and 250 bar system),Out of these three systems 100 bar system using maximum oil which does not necessarily required high quality oil.While 200 bar system and 250 bar system require high quality oil but of less amount.So to maintain the purity of oil for 200 bar and 250 bar pressure line, it is advisable to use a separate storage tank.

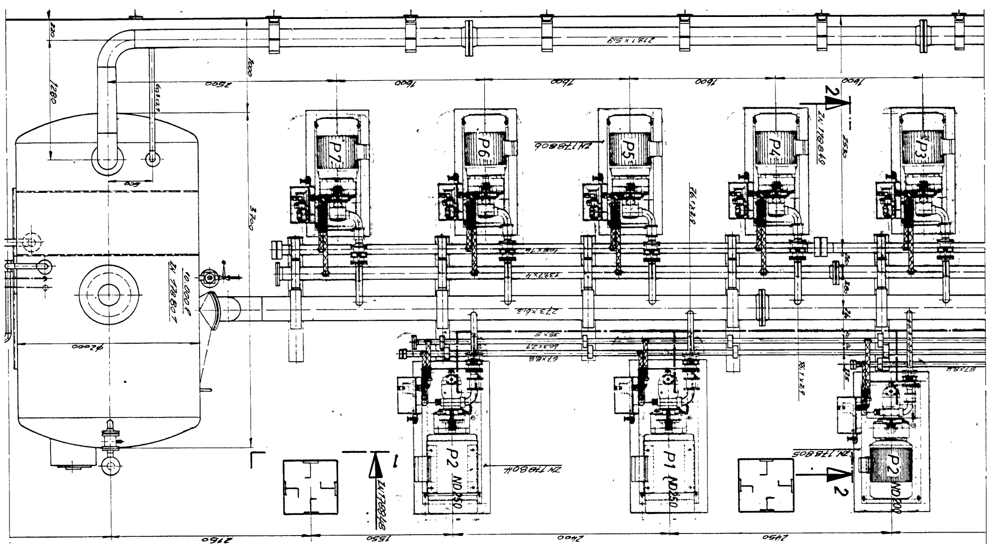

Figure 1.4: Existing system layout

1.5 Objective of Project

Main objective of project is to design new tank and its pipelines. This new tank will supply oil to 200 bar system and 250 bar system. Decision to choose exist tank for 100 bar system is because maximum oil is required to 100 bar system compare to total of 200 bar system and 250 bar system.

1.6 Methodology

Here as per the old tank data available, first volume required for the new tank is calulated.ASME Section VIII Div 1 is followed for designing the tank with cylindrical in shape and both ends closed with ellipsoidal heads.Then saddle is design from the pressure vessel handbook and saddle designed is modified to save the material.Then stand is designed using standard sections available.

1.7 Thesis Organization

Chapter 2 include the overview of the literature which has been reviewed during project work.

Chapter 3 cover the theoretical calculation and 3-D CAD model of pressure vessel and its components.

Chapter 4 cover the design and FEA analysis of saddle.

Chapter 5 cover the design calculation for the stand with 3-D CAD model and analysis results.

Chapter 6 explain the project conclusion and future scope.

Chapter 2

Literature Review

The storage tanks are designed as per the American Petroleum Institute API 650 [2] and API 620 [3]. The standard have specific design procedure for the design of low pressure storage tank but its only applicable when tank is uniformly supported which is possible when tank orientation is in vertical position.API Standard does not have specific design procedure for horizontally oriented tank. Thus designer could not have any design rules for the horizontally oriented tank from the API Standard.

K.Magnucki, J.Lewinskia, P.Stasiewicza [4] formulated optimization criterion for the horizontal cylindrical tank with both ends closed with elliptical head.optimum ratios for basic dimensions of tank with different capacities is found considering allowable space and mass as constraints.

B.S.Thakkar, S.A.Thakkar [5] described that pressure vessel has to withstand different forces while high pressure rise is occur in the pressure vessel. So it is very difficult to take decision on the selection of pressure vessel.The structure of pressure vessel should be designed and fabricated as per ASME standard.Different tests are conducted to measure the performance of a pressure vessel relevant to ASME standard. main objective to present a paper is to design the pressure vessel using ASME codes and standards for legalizing the design. Shaik Abdul Lathuef and K.Chandra Sekhar[6] had discuss some of the potential unintended consequences related to governing thickness of pressure vessel shell as per ASME code.It is mentioned that there is a scope to change the code values as per the desired requirement. Nozzle can be relocate on the pressure vessel for minimizing the stresses in the pressure vessel.It is useful to save the material by taking low value of factor of safety and which lead to reduce the cost with thinner and more flexible pressure vessel.

L. P. Zick [7] developed formulas for the approximate stresses that exist in cylindrical vessels supported on two saddles at various locations with various condition. K. Magnucki, P. Stasiewicz and W.Szyc [8]had done numerical study on the effect of shape and location of supports on the strength of a horizontal cylindrical tank. while Kacperski [9] had done experimental analysis of buckling for a thin walled horizontal cylindrical tank fully filled with a liquid.

Aleksandar Petrovic [10] had done a stress analysis of a cylindrical pressure vessel loaded by axial and transverse force on the free end of a nozzle. Here axis of the nozzle does not cross the axis of the cylindrical shell. After finite element analysis value obtained for stress in the nozzle region is used for finding envelop of maximum stress value, maximum value on these envelop and distance between maximum value on envelop and the outer edge of nozzle. Xue et al. [11] has given the results for the state of shell stress because of the nozzle influence.

Shafique M.A. Khan [12] had modeled quarter portion of pressure vessel and for different load and for different geometries results were investigated for the stress in different parts of the saddle and also recommend the distance between the saddle to end of vessel.

N.El-Abbasi, S.A.Meguid, A.Czekanski [13] has done three-dimensional finite element analysis of a pressure vessel which is a resting on flexible saddle supports.Different pressure vessel configurations are considered and the resulting contact stresses are examined. The effect of saddle radius, saddle width, plate extension and support overhang on the resulting stress field in both vessel and support are evaluated and discussed.

Chapter 3

Pressure Vessel Design

3.1 ASME Section VIII Div 1

The American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section VIII, Division 1 specifies a range of internal pressures from 0.1 MPa to 20 MPa for pressure vessel.For designing pressure vessel first step is to select the appropriate related information and establishment of body for design requirements. Once the first step done, appropriate material is selected and the specified design code will give nominal stress that is used to dimension the main pressure vessel thickness. Additional rules are there to cover the design of various vessel components such as flanges, nozzles and so on.

ASME Section VIII Div 1 Scope and Exclusions

General range

A.15 psig (100 KPa) <Design Pressure (int. or ext.) <3000 psig (20,000 KPa)

B.Internal diameter above 6 inch (152 mm)

Exclusion

A.Those within the scope of other sections and hot water supply storage tanks

B.Fire process tubular heaters and pressure vessels for human occupancy

C.Rotating or reciprocating mechanical devices (pumps, compressors etc.)

D.Piping systems and piping components

E.Vessels containing water up to design pressure of 300 psi (2 MPa) and design temperature of 210O F (99O °C)

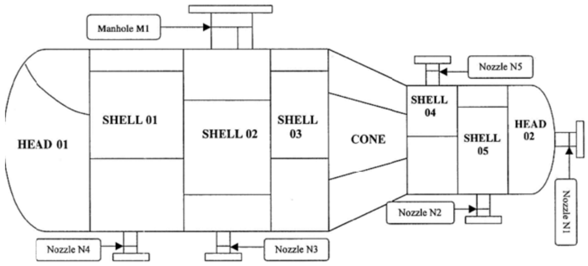

Figure 3.1: Components of pressure vessel[15]

Classification of Pressure Vessel

A.Ratio of diameter to wall thickness of the vessel

1.Thin cylinder $d/t > 20$

2.Thick cylinder $d/t < 20$

B.Geometric shape of the vessel

1.Cylindrical

2.Spherical

3.conical

4.composite

C.End shape of the vessel

1.Flat head

2.Convex head

D.Based on internal pressure

1.Low pressure vessel (L): $0.1 < P < 1.6$ MPa

2.Medium pressure vessel (M): $1.6 < P < 10$ MPa

3.High pressure vessel (H): $10 < P < 100$ MPa

4.Ultra high pressure vessel (U): $P > 100$ MPa

3.2 Tank Volume Calculation

Based on the existing tank volume, a safety factor is calculated for new tank.The volume of tank can be estimated as the total pump supply times safety factor and total pump supply is equal to no. of pump times flow rate through each pump.

Here for 100 bar system no. of pumps is 7 and flowrate through each pump is 190 lpm, for 200 bar system no. of pumps is 2 and flowrate through each pump is 90 lpm, for 250 bar system no. of pumps is 2 and flowrate through each pump is 140 lpm.

Total pump supply = (7×190) + (2×90) + (2×140)

= 1790 lpm

Safety factor = 10,000 / 1790 = 5.59

For new tank volume

Volume required = [(2×90) + (2×140)] × safety factor

= 460 × 5.59

= 2571 litre ≅ 2600 litre

Praposed tank is cylindrical in shape with both ends close with ellipsoidal heads

Volume of tank = (volume of cylindrical shell) + (2 * volume of ellipsoidal heads)

Volume of cylindrical shell $v_1 = \pi \times r^2 \times L$

Volume of elliptical head $v_2 = 2/3 \times \pi \times r^3 \times \beta$

$\beta = 0.5$, 2 : 1 ellipsoidal head

r = Inside radius of cylindrical shell

L = length of cylindrical shell

L = 3.6 * r

Volume of tank

$$V = 2[2/3 \times \pi \times r^3 \times \beta] + [\pi \times r^2 \times L] \tag{3.1}$$

Substituting V=2600 in eq 3.1

r = 580 mm

L = 3.6 * 580 = 2100 mm

3.3 Design of Cylindrical Shell

Design pressure P = 0.2 Mpa

Design temperature T = 65 °C

Inside radius R = 580 mm

Tangent length of shell L = 2100 mm

Allowable stress S = 108 MPa (Material = SA 516 Gr 55)[14]

Joint efficiency E = 0.70[15]

$$t = \frac{PR}{SE - 0.6P} \tag{3.2}$$

t = 1.5447 mm

So Required thickness = t = t + corrosion allowance

= 1.5447 + 3.0

= 4.5447 mm

Next available thickness to order the material plate = 0.1875 inch = 4.7625 mm

3.4 Design of Ellipsoidal Head

Design pressure P = 0.2 Mpa

Design temperature T = 65 °C

Inside diameter D = 1160 mm

Allowable stress S = 108 Mpa (Material = SA 516 Gr 55)[14]

Joint efficiency E = 1 [15]

$$t = \frac{PD}{2SE - 0.2P} \tag{3.3}$$

t = 1.0746 mm

So Required thickness = t = t + corrosion allowance

= 1.0746 + 3.0

= 4.0746 mm

Next available thickness to order the material plate = 0.1875 inch = 4.7625 mm

3.5 Design of Nozzle

Nozzle is designed for outlet from the pressure vessel.Here total 10 nozzles are designed for different purpose.Man hole is designed for the inspection and cleaning purpose.Suction line for supply oil to the pump which further goes to cylinder and return back to vessel by return line.Return drainage line is for empty the tank any time.Heat exchanger and re-filtration line is there for to control the temperature and purify the hydraulic oil.Two outlet is there for the oil level indicator.And air breather is there is to maintain the atmospheric pressure inside the tank and also to purify the air before supplying in to the pressure vessel.

3.5.1 Man-Hole

Design pressure P = 0.2 Mpa

Design temperature T = 65 °C

Outside radius R = 279.4 mm[16]

Allowable stress S = 118 Mpa (Material = SA 106 Gr B)[14]

Joint efficiency E = 1[15]

$$t = \frac{PR}{SE - 0.6P} \tag{3.4}$$

t = 0.4740 mm

Here minimum required thickness is very less because of pressure inside tank is very less and this is very difficult to fabricate this much thickness so here we go for minimum required thickness plus corrosion allowance.

So Required thickness = t = t + corrosion allowance

= 0.4740 + 3.0

= 3.4740 mm

Next available thickness to order the material plate = 4.775 mm

3.5.2 Suction Line

Design pressure P = 0.2 Mpa

Design temperature T = 65 °C

Outside radius R = 109.54 mm[16]

Allowable stress S = 118 Mpa (Material = SA 106 Gr B)[14]

Joint efficiency E = 1[15]

$$t = \frac{PR}{SE - 0.6P} \tag{3.5}$$

t = 0.1858 mm

So Required thickness = t = t + corrosion allowance

= 0.1858 + 3.0

= 3.1858 mm

Next available thickness to order the material plate = 3.759 mm

3.5.3 Return Line

Design pressure P = 0.2 Mpa

Design temperature T = 65 c

Outside radius R = 70.65 mm[16]

Allowable stress S = 118 Mpa (material = SA 106 Gr B)[14]

Joint efficiency E = 1[15]

$$t = \frac{PR}{SE - 0.6P} \tag{3.6}$$

t = 0.1198 mm

So Required thickness = t = t + corrosion allowance

$$= 0.1198 + 3.0$$

$$= 3.1198 \text{ mm}$$

Next available thickness to order the material plate = 3.404 mm

3.6 Area Reinforcement

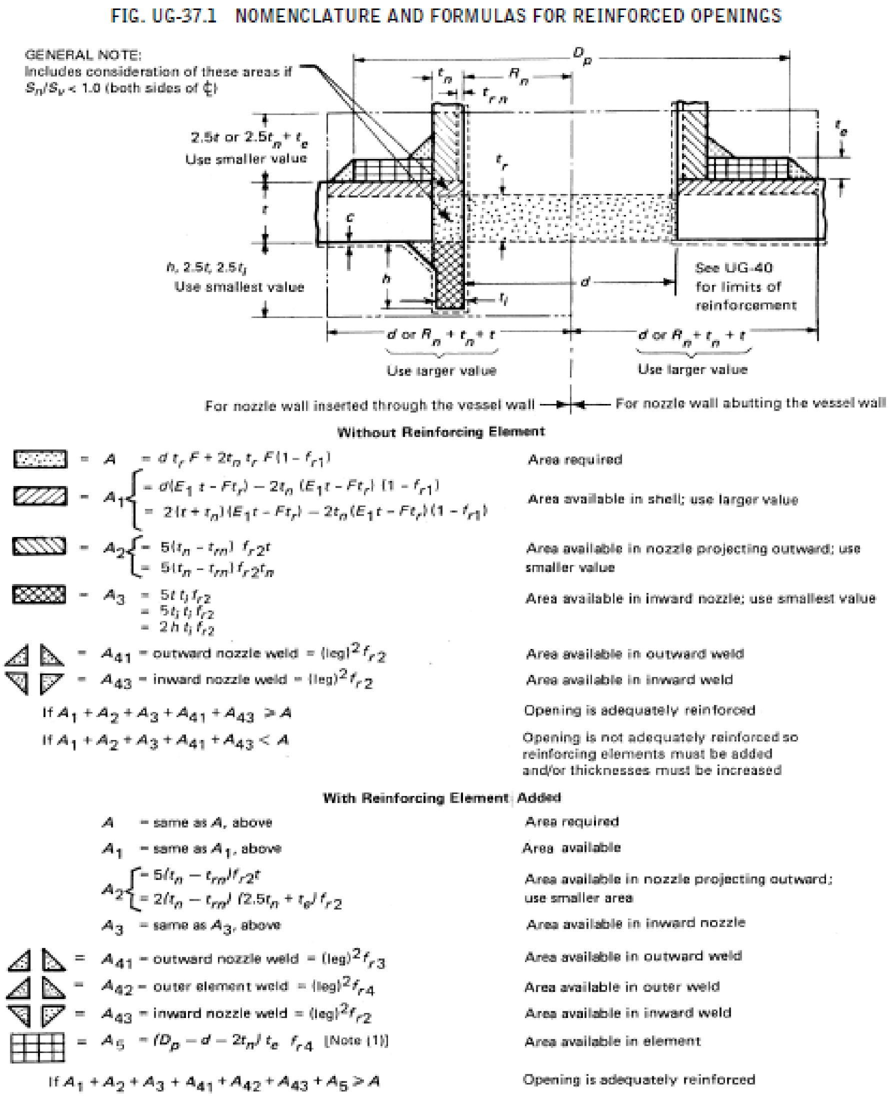

Figure 3.2: Area reinforcement [15]

3.6.1 Man-Hole

f_{r1} = Ratio of allowable stress for nozzle materail to shell material = 118/108 = 1.09

h = Inward projection of nozzle Min(h,2.5t,2.5t_i) = 0

Area required A

$$A = dt_rF + 2t_nt_rF(1 - f_{r1}) \tag{3.7}$$

$$= [(49.25)(1.07)(1)]+2[(4.77)(1.07)(1)(1\text{-}1.09)]$$

A = 589.60 mm^2

Area available in shell A_1

$$A_{11} = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1}) \tag{3.8}$$

$$= [(549.25)(4.76\text{-}1.07)]\text{-}2[(4.77)(4.76\text{-}1.07)(1\text{-}1.09)]$$

A_{11} = 2028.50 mm^2

$$A_{12} = 2(t + t_n)(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1}) \tag{3.9}$$

$$= [(2)(4.76+4.77)(4.76\text{-}1.07)]\text{-}2[(4.77)(4.76\text{-}1.07)(1\text{-}1.09)]$$

A_{12} = 73.59 mm^2

A_1 = Max(A_{11},A_{12})=2028.50 mm^2

Area available in nozzle projecting outward A_2

$$A_{21} = 5(t_n - t_{rn})f_{r2}t \tag{3.10}$$

$$= (5)(4.76\text{-}0.47)(1.09)(4.76)$$

A_{21} = 111.89 mm^2

$$A_{22} = 5(t_n - t_{rn})f_{r2}t_n \tag{3.11}$$

$$= (5)(4.76\text{-}0.47)(1.09)(4.77)$$

$$A_{22} = 112.18\ mm^2$$

A_2 = Min(A_{21},A_{22})=111.89 mm^2

Area available in nozzle projecting inward A_3

$$A_{31} = 5tt_i f_{r2} \tag{3.12}$$

$$= (5)(4.76)(4.77)(1.09)$$

$A_{31} = 124.22\ mm^2$

$$A_{32} = 5t_i t_i f_{r2} \tag{3.13}$$

$$= (5)(4.77)(4.77)(1.09)$$

$A_{32} = 124.54\ mm^2$

$$A_{33} = 2ht_i f_{r2} \tag{3.14}$$

$$= (2)(0)(4.77)(1.09)$$

$A_{33} = 0\ mm^2$

$A_3 = \mathrm{Min}(A_{31}, A_{32}, A_{33}) = 0\ mm^2$

Area available in outward weld A_{41}

$$A_{41} = (leg)^2 f_{r2} \tag{3.15}$$

$$A_{41} = 0$$

Area available in inward weld A_{43}

$$A_{43} = (leg)^2 f_{r2} \tag{3.16}$$

$$A_{43} = 0$$

Here $A_1 + A_2 + A_3 + A_{41} + A_{43} > A$; So reinforcement is not required.(Fig 3.2) [15]

3.6.2 Suction Line

f_{r1} = Ratio of allowable stress for nozzle material to shell material =118/108 = 1.09

h = Inward projection of nozzle = Min(h,2.5t,2.5t_i) = 0

Area required A

$$A = dt_rF + 2t_nt_rF(1 - f_{r1}) \tag{3.17}$$

= [(211.56)(1.07)(1)]+2[(3.75)(1.07)(1)(1-1.09)]

A = 226.72 mm^2

Area available in shell A_1

$$A_{11} = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1}) \tag{3.18}$$

= [(211.562)(4.76-1.07)]-2[(3.75)(4.76-1.07)(1-1.09)]

A_{11} = 782.53 mm^2

$$A_{12} = 2(t + t_n)(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1}) \tag{3.19}$$

= [(2)(4.76+3.75)(4.76-1.07)]-2[(3.75)(4.76-1.07)(1-1.09)]

A_{12} = 65.40 mm^2

A_1 = Max(A_{11},A_{12})=782.53 mm^2

Area available in nozzle projecting outward A_2

$$A_{21} = 5(t_n - t_{rn})f_{r2}t \tag{3.20}$$

= (5)(3.75-0.18)(1.09)(4.76)

A_{21} = 92.95 mm^2

$$A_{22} = 5(t_n - t_{rn})f_{r2}t_n \tag{3.21}$$

= (5)(3.75-0.18)(1.09)(3.75)

A_{22} = 73.37 mm^2

A_2 = Min(A_{21},A_{22})=73.37 mm^2

Area available in nozzle projecting inward A_3

$$A_{31} = 5tt_i f_{r2} \tag{3.22}$$

$$= (5)(4.76)(3.75)(1.09)$$

$A_{31} = 97.79\ mm^2$

$$A_{32} = 5t_i t_i f_{r2} \tag{3.23}$$

$$= (5)(3.75)(3.75)(1.09)$$

$A_{32} = 77.18\ mm^2$

$$A_{33} = 2ht_i f_{r2} \tag{3.24}$$

$$= (2)(0)(3.75)(1.09)$$

$A_{33} = 0\ mm^2$

A_3= Min(A_{31},A_{32},A_{33}) $= 0\ mm^2$

Area available in outward weld A_{41}

$$A_{41} = (leg)^2 f_{r2} \tag{3.25}$$

$$A_{41} = 0$$

Area available in inward weld A_{43}

$$A_{43} = (leg)^2 f_{r2} \tag{3.26}$$

$$A_{43} = 0$$

Here A_1 + A_2 +A_3+A_{41}+A_{43} $> A$; So reinforcement is not required.(Fig 3.2)[15]

3.6.3 Return Line

f_{r1} = Ratio of allowable stress for nozzle material to shell material =118/108 = 1.09

h = Inward projection of nozzle = Min(h,2.5t,2.5t_i) = 8.51 mm

Area required A

$$A = dt_rF + 2t_nt_rF(1 - F_{r1}) \tag{3.27}$$

$$= [(134.49)(1.07)(1)]+2[(3.40)(1.07)(1)(1\text{-}1.09)]$$

$A = 143.60\ mm^2$

Area available in shell A_1

$$A_{11} = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1}) \tag{3.28}$$

$$= [(134.492)(4.76\text{-}1.07)]\text{-}2[(3.40)(4.76\text{-}1.07)(1\text{-}1.09)]$$

$A_{11} = 498.23\ mm^2$

$$A_{12} = 2(t + t_n)(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1}) \tag{3.29}$$

$$= [(2)(4.76\text{+}3.40)(4.76\text{-}1.07)]\text{-}2[(3.40)(4.76\text{-}1.07)(1\text{-}1.09)]$$

$A_{12} = 62.54\ mm^2$

A_1 = Max(A_{11},A_{12})=498.23 mm^2

Area available in nozzle projecting outward A_2

$$A_{21} = 5(t_n - t_{rn})f_{r2}t \tag{3.30}$$

$$= (5)(3.40\text{-}0.12)(1.09)(4.76)$$

$A_{21} = 85.43\ mm^2$

$$A_{22} = 5(t_n - t_{rn})f_{r2}t_n \tag{3.31}$$

$$= (5)(3.40\text{-}0.12)(1.09)(3.40)$$

$A_{22} = 61.06\ mm^2$

A_2 = Min(A_{21},A_{22})=61.06 mm^2

Area available in nozzle projecting inward A_3

$$A_{31} = 5tt_i f_{r2} \tag{3.32}$$

$$= (5)(4.76)(3.40)(1.09)$$

$A_{31} = 97.79\ mm^2$

$$A_{32} = 5t_i t_i f_{r2} \tag{3.33}$$

$$= (5)(3.40)(3.40)(1.09)$$

$A_{32} = 77.18\ mm^2$

$$A_{33} = 2ht_i f_{r2} \tag{3.34}$$

$$= (2)(8.51)(3.40)(1.09)$$

$A_{33} = 63.29\ mm^2$

A_3= Min(A_{31},A_{32},A_{33}) $= 63.29\ mm^2$

Area available in outward weld A_{41}

$$A_{41} = (leg)^2 f_{r2} \tag{3.35}$$

$$A_{41} = 0$$

Area available in inward weld A_{43}

$$A_{43} = (leg)^2 f_{r2} \tag{3.36}$$

$$A_{43} = 0$$

Here $A_1 + A_2 + A_3 + A_{41} + A_{43} > A$; So reinforcement is not required.(fig 3.2)[15]

Table 3.1: Nozzle specification

Nozzle	NPS	OD	Schdule No	Thickness	ID	Length
Man hole	20”	508	Sch 5	4.77	549.25	55
Suction line	8”	219.08	Sch 10	3.75	211.56	55
Return Line	5”	141.30	Sch 10	3.40	134.49	555
Drainage Line	5”	141.30	Sch 10	3.40	134.49	55
H.E. and Fitretion S.	2.5”	73.03	Sch 10	3.04	66.93	55
H.E. and Fitretion R.	1.25”	42.16	Sch 10	2.76	36.62	55
Level indicator	1.25”	42.16	Sch 10	2.76	36.62	200
Level indicator	1.25”	42.16	Sch 10	2.76	36.62	200
Air breather	3”	88.90	Sch 10	3.04	82.80	55

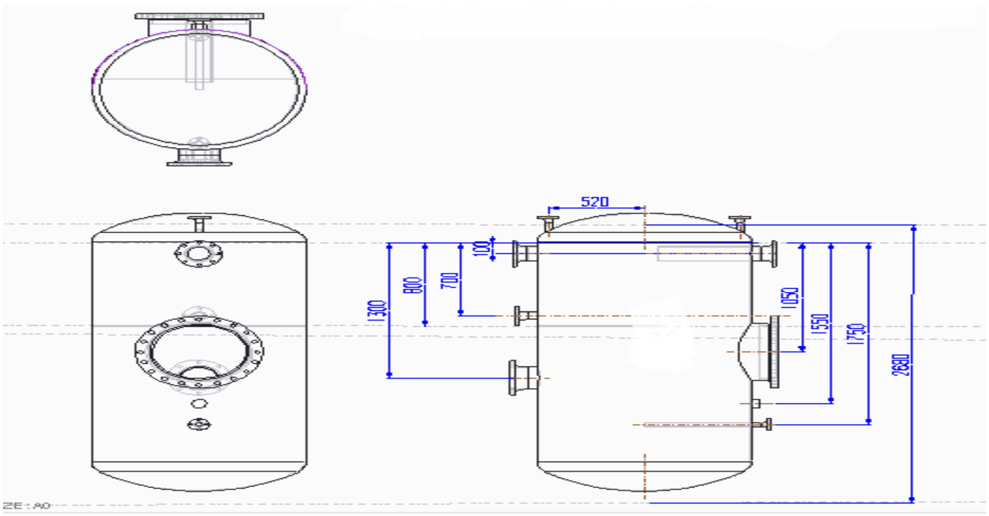

Figure 3.3: Nozzle location

3.7 Flange Design

ASME B16.5 Pipe Flanges and Flanged Fittings is a basic standard use for designing flanges.This standard covers temperature pressure ratings, dimensions, materials, testing, tolerances and methods of designating openings for pipe flanges and flanged fittings.Flanges available with rating class designations 150, 300,400, 600, 900, 1500 and 2500.This Standard also included the requirements and recommendations for flange joints, flange gaskets and flange bolting.

This standard is limited to (1) Flanges and flanged fittings made from cast or forged materials (2) Blind flanges and certain reducing flanges made from cast, forged, or plate materials.

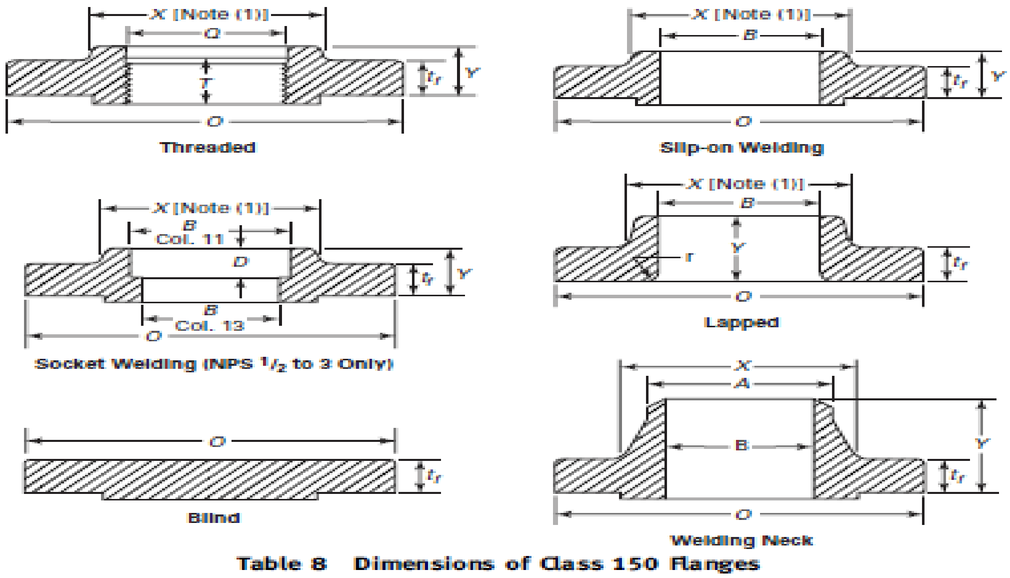

Figure 3.4: Different types of flange [17]

Different types flange of used for different purpose.Here first step to design the flange is to decided the flange rating as per the design pressure and design temperature as per given pressure and temperature class 150 is best suitable and then material is selected carbon steel SA 105 (forged).Here for man hole lapped flange is used which is best suitable for frequent dismantling setup and for other nozzle weld neck flange is used.For man-hole,suction line and delivery line dimension is given below and for other nozzle its dimension is available in ASME B16.5 pipe flanges and flanged fittings [17].

Table 3.2: Standard flange dimension

ASME B16.5 Pipe Flanges and Flanged Fittings			
Flange Class 150 (As per pressure and temperature rating)			
Material = Carbon Steel SA 105 (Forged)			
	Man hole	Suction line	Return line
Type	Lapped	Weld Neck	Weld Neck
NPS	20”	8”	5”
OD of Flange O	700 mm	345 mm	255 mm
Diameter of Bolt Circle W	635 mm	298.5 mm	215.9 mm
Diameter of Bolt Hole	31.75 mm	22.22 mm	22.22 mm
Number of Bolt	20	8	8
Diameter of Bolt	28.57 mm	19.05 mm	19.05 mm
Length of Bolt L	160 mm	110 mm	95 mm
Thickness of Flange t_f	41.3 mm	27 mm	22.3 mm
Diameter of Hub X	559 mm	246 mm	164 mm
Length through Hub Y	1003 mm	100 mm	87 mm
Bore B	514.4 mm	202.7 mm	128.2 mm

3.8 Baffle Design

Baffle plate is designed for dividing the pressure vessel into two compartments.One is suction line and the other one is return line compartment.Materail used for baffle plate is same as pressure vessel material SA 516 Gr 55. Baffle plate is welded near the return line so as oil from return line fall in to return line compartment and impurities with oil gather at the bottom of the tank where heat exchanger and re-filtration suction port is there, so oil with impurities goes to heat exchanger and re-filtration unit from where low temperature and purified oil return back to suction line compartment by heat exchanger and re-filtration return line.so to avoid impurities and lower the temperature of oil which is supplied to pump baffle is designed and located near the return line.

For designing the baffle plate consider one short edge is free and three edges are simply supported, uniformly decreasing load to the free edge and for un-stiffened Baffle plate.

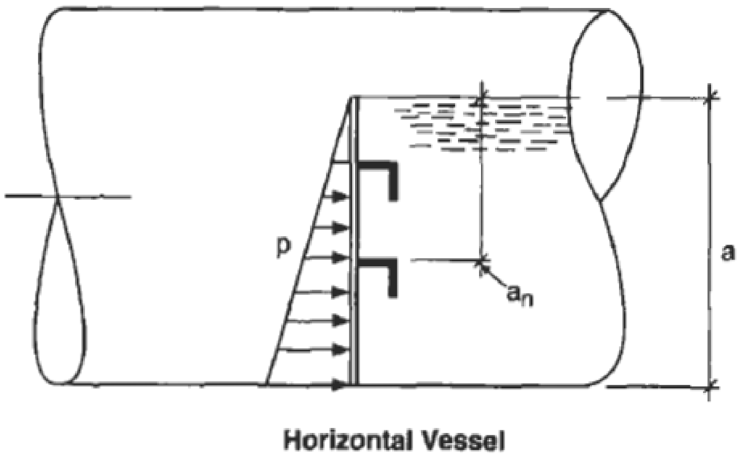

Figure 3.5: Baffle design criteria [18]

a = 680 mm , b = 1142.63 mm

a/b = 0.59 ≅ 0.5

β = 0.11 [18]

γ = 0.02 [18]

ρ =880 kg/m^3

E = 2 × 10^5 MPa

Find Load P

$$P = \rho \times g \times a \tag{3.37}$$

P = 0.0058 MPa

Factor of saftey = 1.5

F_y =205 MPa

F_y = 205/1.5 = 136 MPa

F_b = 0.66 × F_y = 89.76 MPa

Find the Baffle Thickness t_b

$$t_b = \sqrt{\frac{Pb_2\beta}{F_b}} \tag{3.38}$$

$$t_b = 3.06 \text{ mm}$$

Standard thickness available = 4.7625 mm

Baffle deflection δ

$$\delta = \frac{P\gamma b^4}{Et_b^3} \tag{3.39}$$

$$\delta = 3.46 \text{ mm}$$

Which should be minimum of $t_b/2 = 2.38$ mm or b/360 = 3.17 mm

Here deflection is more so there is three ways to avoid failure.

A.Increase baffle thickness

B.Add stiffeners

C.Go to Curved baffle design

So let go for option A and increase the thickness and let the new standard thickness available is 6.35 mm, so after putting new thickness in baffle deflection formula we get 2.25 mm which is less compare to $t_b/2 = 3.17$ mm or b/360 = 3.17 mm. So baffle thickness is 6.35 mm and material is same as shell and head.

3.9 3-D CAD Modeling of Pressure Vessel

Based on the analytical calculation done for pressure vessel as per ASME Section VIII Div I, 3-D CAD Model is modeled in Creo Parametric 2.0.Total length of pressure vessel is 2680 mm and height is 1160 mm.Total weight of pressure vessel is 530 kg and the total weight of 2600 litre hydraulic oil is 2300 kg.So total wight of tank fully filled with hydraulic oil is 2830 kg $\cong$ 3000 kg.

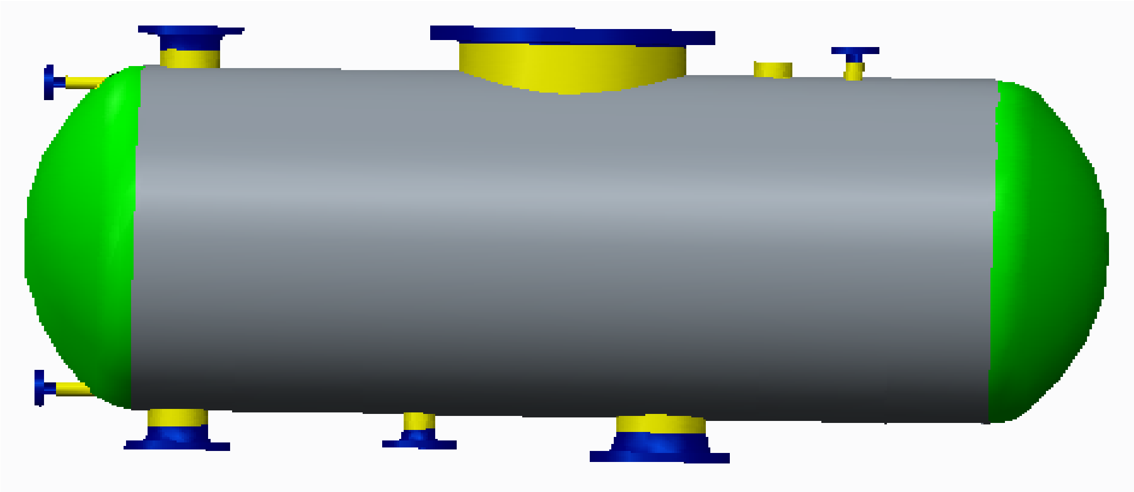

Figure 3.6: Pressure vessel CAD model

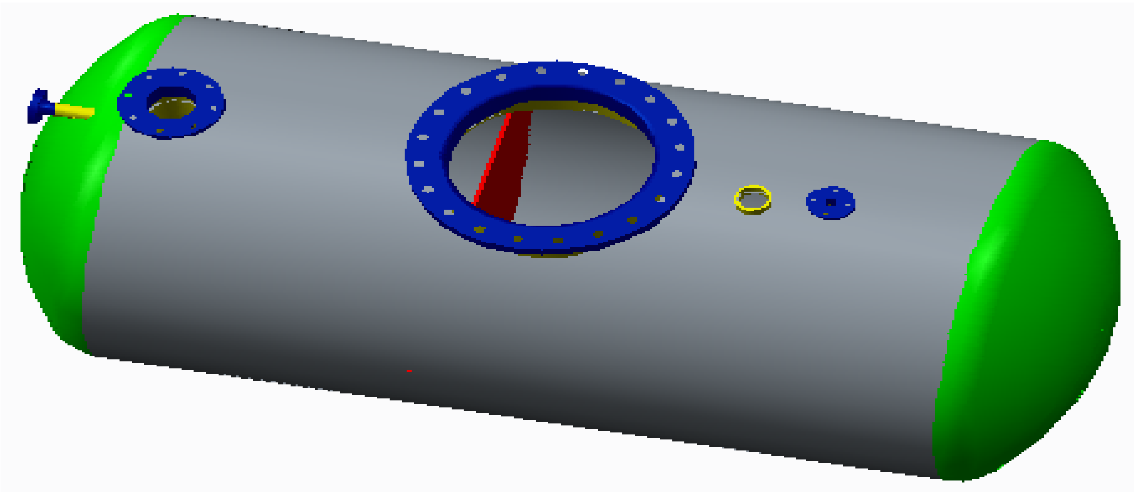

Figure 3.7: Baffle inside pressure vessel

Chapter 4

Support Design

4.1 Saddle Design

The pressure vessels are required to be supported at various stages such as during post weld heat treatment (PWHT), during hydro test, during the transportation and in the process plant permanently.Different types of supports are used to support the pressure vessel like skirt support, lugs support, leg support, saddle support. The selection of the type of support for a pressure vessel is dependent on several factors such as the size of the vessel, its wall thickness, space available and elevation of the vessel in relation to the ground floor and the material of construction.

The pressure vessels in horizontal orientation are commonly supported on the vertical cradles known as saddles.Pressure vessel, when resting on the saddle supports behave like a beam.The size, shape and the position of saddles affect the stresses induced inside the pressure vessel.Here design of saddle is done using pressure vessel handbook by E Megyesy [19] and L P Zick [7] analysis is performed to check the stress at different location of pressure vessel and saddles.

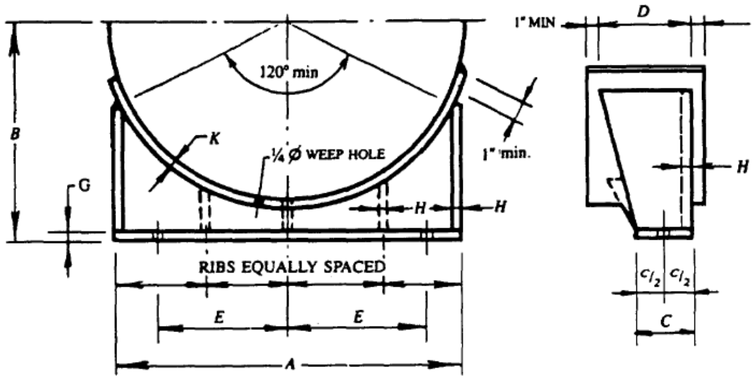

Figure 4.1: Saddle design [19]

The design is based on:

1. The vessel supported by two saddles
2. To resist horizontal force F due to the maximum operating weight of vessel as tabulated.
3. The minimum contact angle of shell and saddle 120°.

Standard dimension and thickness for different plates used in Saddle is given as per the pressure vessel handbook.

Table 4.1: Standard saddle dimension [19]

A	B	C	D	E	Bolt Dia	No of Bolt	G	H	K	Max Weight
mm	mm	mm	mm	mm	mm		mm	mm	mm	kg
1032	800	152	280	400	19.05	2	19.05	9.52	9.52	114305

Material = carbon steel SA 285 Gr B

Tensile strength = 345 MPa

Yield Strength = 185 MPa

Factor of safety = 1.5

Allowable stress = 185/1.5 = 124 MPa

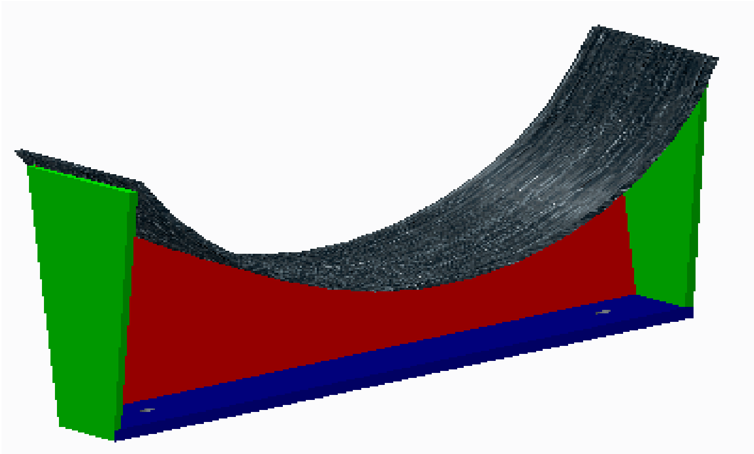

Figure 4.2: Standard saddle CAD model

Total weight of one saddle is 92 kg.3-D CAD modeling is done is Creo Parametric 2.0 and for finite element analysis ANSYS 14.0 is used.Meshing is done using tetrahedron element with 13,856 elements and 27,928 nodes.For analysis total weight of tank is 3000 Kg times 1.5 (factor of saftey) equals to 4500 kg.So Force on one saddle is 22500 N.For analysis Purpose Force act vertically downward on wear plate and base plate is fixed from the bottom.

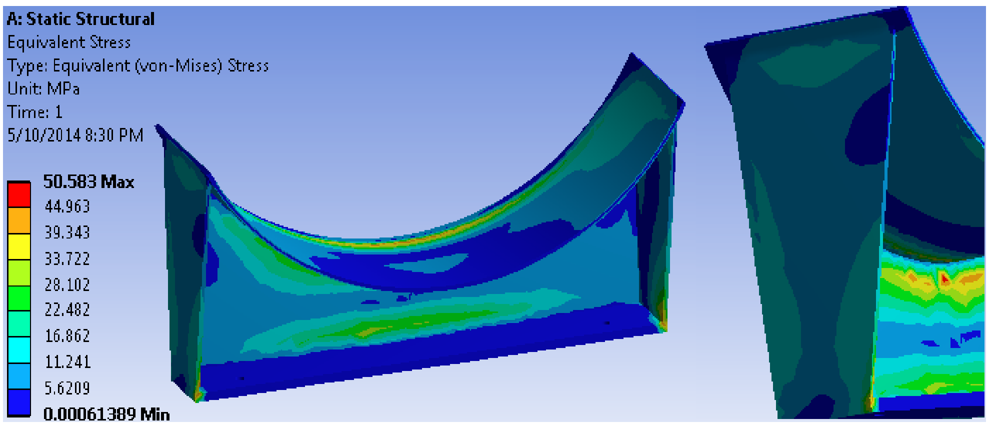

Figure 4.3: Von-Mises stress

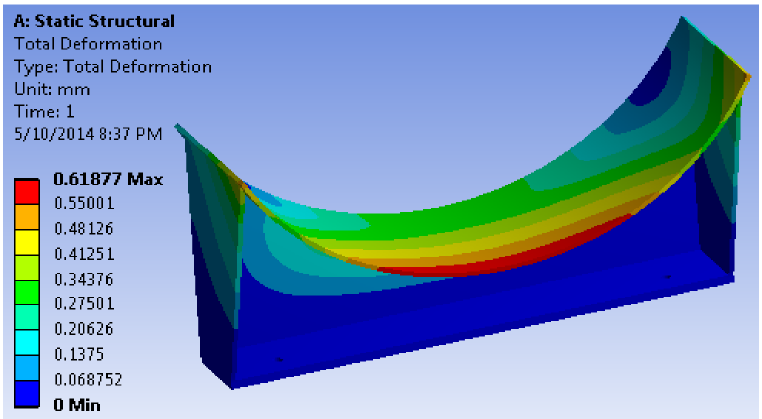

Figure 4.4: Deformation

Here saddle is connected with pressure vessel by welding and this saddles is connected to the stand by two bolts.Here bolts are in under compression, stress inside the bolts ;

F = 45000 N

Material = SA 193 B16

Allowable stress = 482 MPa

No. of bolt = 2

Diameter of Bolt = 19.05 mm

A = $[\pi/4(D)^2]\times$ No. of Bolts = 570.05 mm

$$\sigma = \frac{F}{A} \tag{4.1}$$

$$\sigma = 78.94 \text{ MPa}$$

4.2 Weight Reduction of Saddle

Here stress is very less so it is possible to minimize the thickness of plates to save the material.Trial and Error method is used to get best thickness and design of saddle considering material saving and safety.

Table 4.2: Stress and deformation for different saddle plates

Trial	G	H	K	Weight	Max Stress	Max Deformation
	mm	mm	mm	kg	MPa	mm
1	15.87	7.93	7.93	76	69	0.83
2	9.53	6.35	6.35	57	77	0.88
3	4.76	4.76	4.76	40	97.635	1.12

Here from above table trial 3 is suitable for saddle design but there is scope to reduce the maximum stress and deformation by modifying the saddle design.

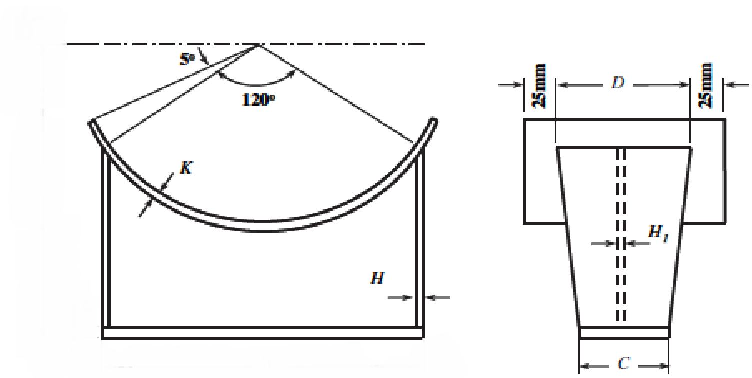

Figure 4.5: Proposed saddle design [12]

Table 4.3: Final saddle dimension

A	B	C	D	E	Bolt Dia	No of Bolt	G	H	K
mm	mm	mm	mm	mm	mm		mm	mm	mm
1024	785	152	280	400	19.05	2	4.76	4.76	4.76

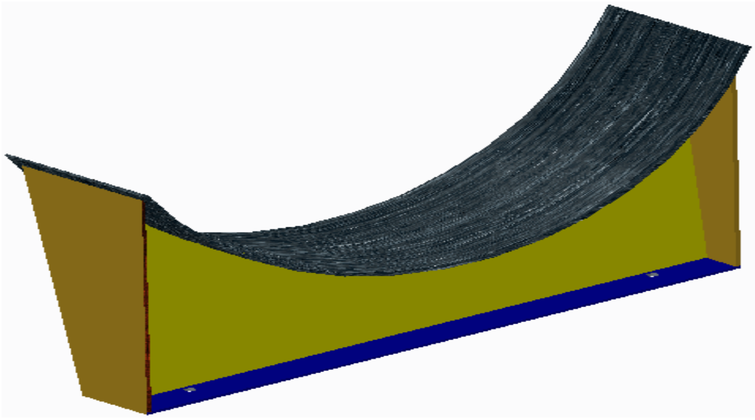

Figure 4.6: Finalized Saddle CAD model

Total weight of saddle is 40 kg.3-D CAD modeling is done is Creo Parametric 2.0 and for finite element analysis ANSYS 14.0 is used.Meshing is done using tetrahedron element with 36,769 elements and 31,679 nodes.For analysis Total Weight of tank is 3000 Kg times 1.5 (factor of saftey) equal to 4500 kg.So Force on one saddle is 22500 N.For analysis Purpose Force act vertically downward on wear plate and base plate is fixed from the bottom.

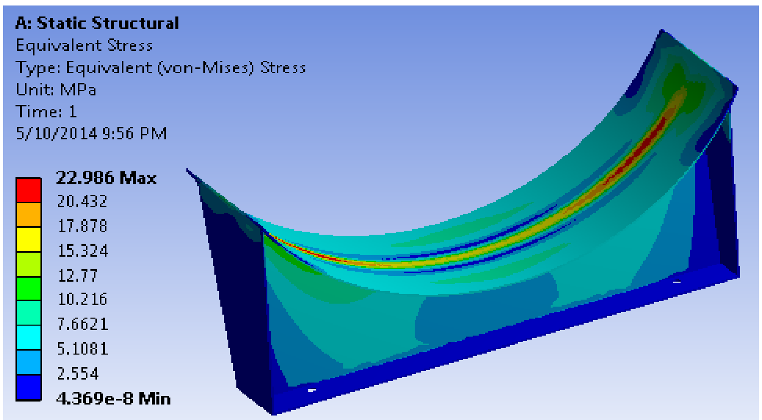

Figure 4.7: Von-Mises stress

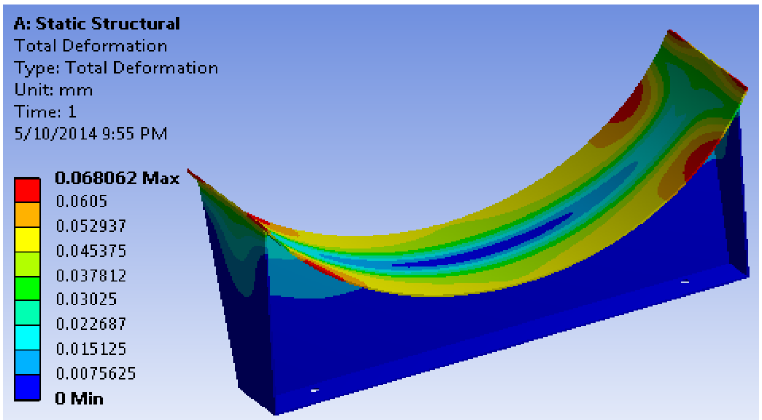

Figure 4.8: Deformation

4.3 L P Zick Analysis

The design methods of supports for horizontal vessels are based on L. P. Zicks [7] analysis presented in 1951.A horizontal vessel on saddle support acts as a beam with the following deviations:

1. The loading conditions are different for a full or partially filled vessel.
2. The stress in the vessel vary according to the angle included by the saddles.
3. The load due to the weight of the vessel is combined with other loads.

Location of Saddle

The location of the saddles is normally decided by the location of openings, sumps etc., which is in the bottom of the vessel. If this is not a matter then the saddles can be placed at the statically optimum point.Large diameter vessel with thin wall are best supported near the heads. Long and thick walled vessels are best supported where the maximum longitudinal bending stress at the saddles is nearly equal to the stress at the mid span.The distance between the shell tangent line and the saddle should not be more than 0.2 times the length of the vessel[19].The minimum contact angle suggested by the ASME Code is 120[19].

Vessels supported by saddles are subject to:

1. Longitudinal bending stress
2. Tangential shear stress
3. Circumferential stress

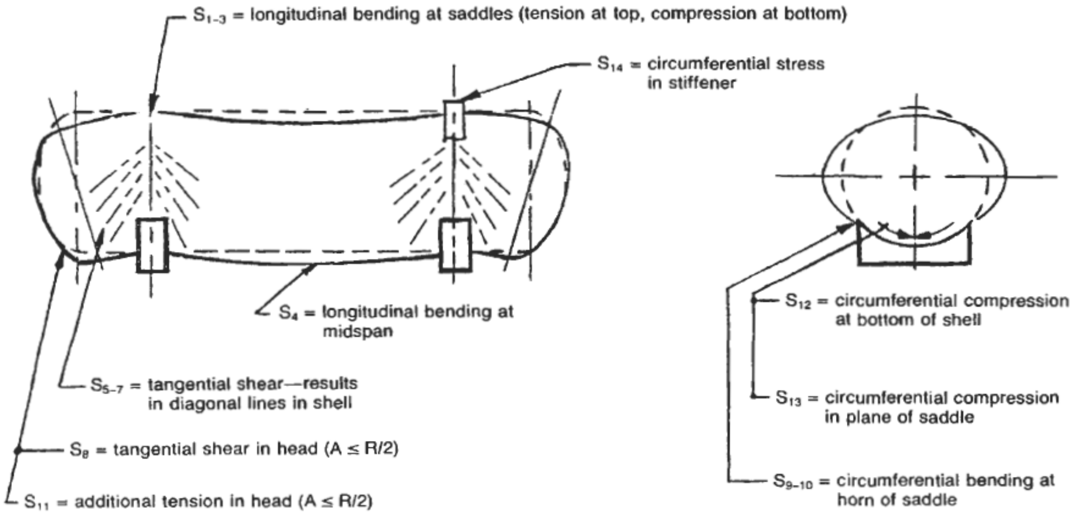

Figure 4.9: Stresses at the different location on saddle and pressure Vessel[18]

Q = 22500 N

R = 584.76 mm

t_s = 4.76 mm

t_h = 4.76 mm

K = Constant [19]

1.Longitudinal Bending

At the saddle

Tension at the top and compression at the bottom

$$S_{11} = \pm \frac{QA(1 - \frac{1-\frac{A}{L}+\frac{R^2-H^2}{2AL}}{1+\frac{4H}{3L}})}{K_1 R^2 t_s} \tag{4.2}$$

$$S_{11} = 3.04 \text{ MPa}$$

The compression stress is not a factor for steel vessel where t/R $\geq$ 0.005.

At Mid span

Tension at the bottom and compression at the top

$$S_{12} = \pm \frac{\frac{QL}{4}(\frac{1+2\frac{R^2-H^2}{L^2}}{1+\frac{4H}{3L}} - \frac{4A}{L})}{\Pi R^2 t_s} \tag{4.3}$$

$$S_{12} = 0.42 \text{ MPa}$$

Maximum stress

Here PR/2t_s = 12.28 MPa

Total tension stresses = S_{11}+S_{12}+PR/2t_s

= 3.04 + 0.42 + 12.28

= 15.74 MPA

Maximum allowable stress = allowable stress $\times$ 0.70 = 108 $\times$ 0.70 = 75.6 MPa

The sum of tension stresses does not exceed the allowable stress value for designed pressure vessel.Hence pressure vessel is safe.

2.Tangential shear

In shell

t_s = thickness of wear plate + thickness of shell = 4.76 +4.76 mm = 9.52 mm

$$S_{21} = \frac{K_2 Q}{Rt_s}(\frac{L-2A}{L+4/3H}) \tag{4.4}$$

$$S_{21} = 2.47 \text{ MPa}$$

In Head

$$S_{22} = \frac{K_3 Q}{Rt_s}(\frac{L-2A}{L+4/3H}) \tag{4.5}$$

$$S_{22} = 1.29 \text{ MPa}$$

Maximum allowable stress = $S_2 < 0.8 \times$ Allowable stress

$= S_2 < 86.4$

So pressure vessel is safe from Tangential shear stress.

3.Circumferential stress

At the horn of saddle

t_s = thickness of wear plate + thickness of shell = 4.76 +4.76 mm = 9.52 mm

$t_s^2 = 4.76^2 + 4.76^2 = 45.31$ mm

b = width of saddle = 152.5 mm

$$S_4 = -\frac{Q}{4t_s(b + 1.56\sqrt{Rt_s})} - \frac{3K_6Q}{2t_s^2} \tag{4.6}$$

S_4 = -90.14 MPa

Maximum allowable stress = $S_4 < 1.50 \times$ allowable stress

$= S_4 < 162$

Pressure vessel is safe from circumferential stress at the horn of saddle.

At the bottom of shell

$$S_5 = -\frac{K_7Q}{4t_s(b + 1.56\sqrt{Rt_s})} \tag{4.7}$$

S_5 = -6.68 MPa

Compression yield point of shell = 205 MPa

Maximum allowable stress = $S_5 < 0.5 \times$ Compression yield point

$= S_5 < 102.5$ MPa

So pressure vessel is safe from circumferential stress at the horn of saddle.

Here all induced stress in the vessel for the given data is bellow the allowable stresses. So design is safe and provided plate thickness is adequate and also location of saddles is at optimum place.

Chapter 5

Stand Design

Stand is designed to install pressure vessel to the height, this is require because oil supplied to pumps are not suction pumps so it is require to put the vessel on some height, so by suction head pressure oil automatically reaches to pump suction line.At present piston pumps are installed on concrete base of 450 mm height and pump suction port is 870 mm above from the ground so we require to put pressure vessel on the height above 870 mm.Here flow control valve also installed in suction line pipe and for ease of operation height of stand selected is 1500 mm.Saddle is attached using bolts to stand.Stand is designed to withstand 50000 N (Total of liquid filled vessel weight plus saddle weight plus safety factor)load.Stand is manufactured from Structural Steel IS 2602 Gr B

Utlimate tensile strength = 410 MPa

Yield strength = 240 MPa

Factor of safety = 1.5

Allowable stress = 160 MPa

Stand is manufacture using two standard I-beams as suitable for carrying bending load on which saddles are resting and four standard channel section are used as a legs of stand and two standard angle section is used to connect legs of the stand.Four holes are drilled in I beams for holding the saddle through four bolts.Base plate is used for easy standing.

5.1 Design Methodology

Selection of I Beam

Total length of I Beam = 1024 mm

Load on one I Beam = 25000 N

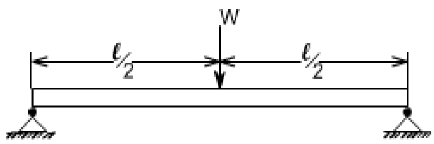

Figure 5.1: Simply supported beam

Here let considering that point load at the mid span = W/2 = w = 12500 N

Half length of beam = L/2 = l = 512 mm

Moment

$$M = \frac{wl}{2} \tag{5.1}$$

M = 3200000 N-mm

Bending stress

$$\sigma_b = \frac{M}{Z} \tag{5.2}$$

Where Z = Section modulus

Allowable bending stress σ_b= 0.66 × allowable yield stress= (0.66)(160)=105.6 MPa

Solving equation 5.2 we get, required section modulus Z = 30303.03 mm^3

Now from standard I Beam available select the I beam whose section modulus is greater than the required section modulus. To accommodate the the saddle base plate and considering material saving in mind best I Beam available is ISHB 150

Table 5.1: Standard I beam dimension [20]

Beam	Height	Width	Thickness of flange	Thickness of web	Moment of intertia
	mm	mm	mm	mm	mm^4
ISHB 150	150	150	9	5.4	14556000

For selected I beam, Section modulus

$$Z = \frac{I_{xx}}{Y} \tag{5.3}$$

Y = h/2 = 150/2 = 75 mm

I_{xx} = 14556000 mm^4

$$Z = 194080\ mm^3$$

Here selected beam section modulus is higher than the required section modulus.So design is safe.

Selection of Channel Section

Total length of channel section = 1350 mm

Load on One Channel section = 50000/4 = 12500 N

Allowable yield stress = 160 MPa

lets go with the smallest Channel section available so let us take ISJC 100

Area of Channel section = 741 mm^2

Table 5.2: Standard Channel section dimension [20]

Channel	Depth	Width	Thickness of Flange	Thickness of web
	mm	mm	mm	mm
ISJC 100	100	45	5.1	3

$$P_{allowable} = 0.60 \times F_y \times A_g \tag{5.4}$$

$$P_{allowable} = 71136 \text{ N}$$

Here allowable load is higher than the design load so design is safe.

Angle Section Selection

Here angle section is welded with Channel section from the distance of 300 mm from the top of channel section.Here angle section is only used to connect the legs of stand so go for ISA 4545

Width = Length = 45 mm

Thickness = 4 mm

Base Plate

Base plate = 150 × 95 × 6.35 mm

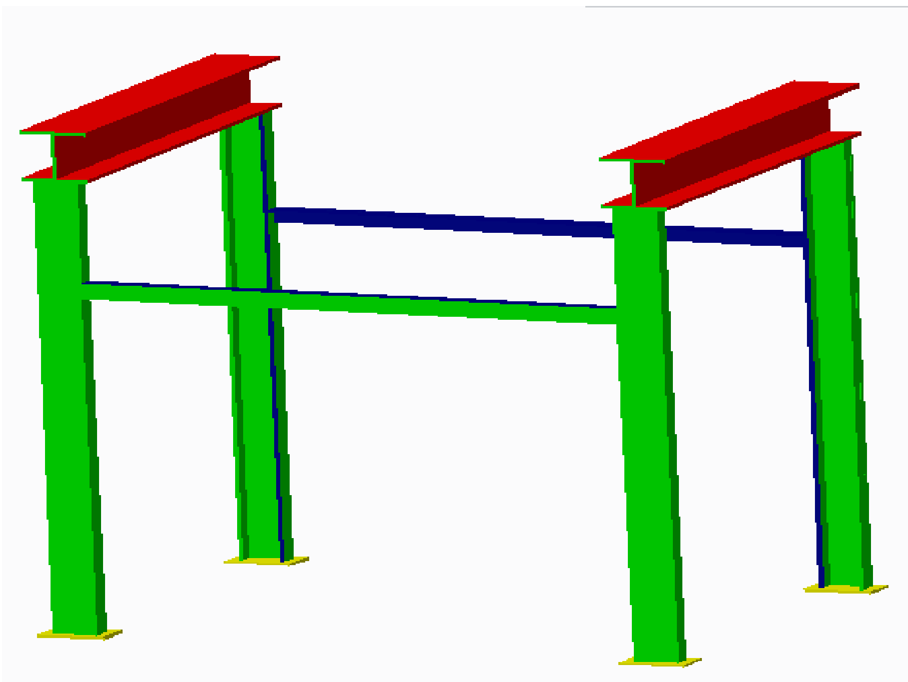

Figure 5.2: Stand CAD model

5.2 Stand Analysis

Total Weight of stand is 95 kg. 3-D CAD modeling is done is Creo parametric 2.0 and for finite element analysis ANSYS 14.0 is used.Meshing is done using tetrahedron element with 22,473 elements and 48,788 nodes.For analysis Total Force on stand is 50000 N.For analysis Purpose Force act vertically downward on top plate and base plates are fixed from the bottom.

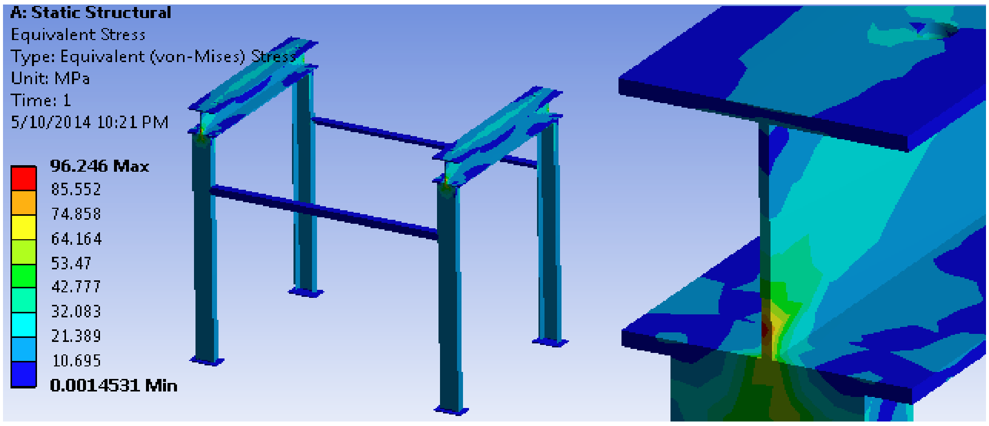

Figure 5.3: Von-Mises stress

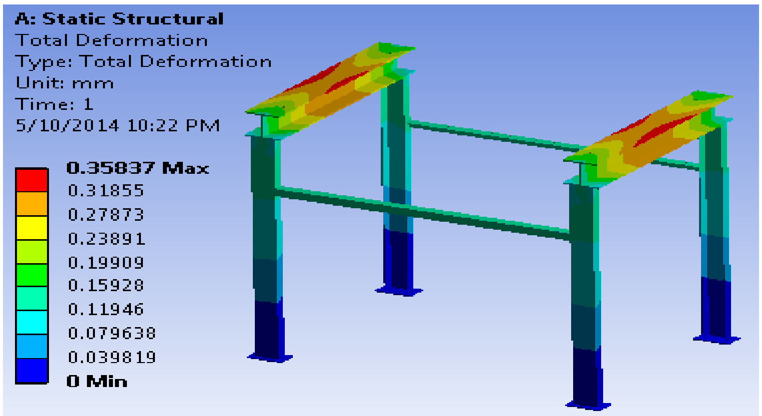

Figure 5.4: Deformation

5.3 Assembly

Figure 5.5: Final assembly

Chapter 6

Conclusion and Future Scope

6.1 Conclusion

A new storage tank is designed for combined 200 bar and 250 bar system using ASME Section VIII Div I. The existing tank is used for 100 bar system, as high pressure line required highly purified oil whereas in low pressure line high amount of purity is not needed.so by designing separate tank for high pressure line, the cost of purification for low pressure line can be eliminate, as it is provided only for high pressure line.

6.2 Future Scope

The project can be extended by designing the supporting piping system for high pressure line.

References

[1] Introduction to Hydraulics by GATES Professional Development Series.

[2] API 650: Welded steel tank for Oil Storage, Washington, D.C. 2007

[3] API 620: Design and Construction of Large Welded Low Pressure Storage Tanks, Washington, D.C. 2005

[4] K Magnucki, J Lewinskia, P Stasiewicz, *Optimal sizes of a ground-based horizontal cylindrical tank under strength and stability constraints* International Journal of Pressure Vessels and Piping,vol.81,pp.913-917,2004.

[5] B S Thakkar, S A Thakkar, *Design of pressure vessel using ASME code, Section viii, division 1* International Journal of Advanced Engineering Research and Studies.

[6] Shaik Abdul Lathuef and K Chandra sekhar, *Design and structural analysis of pressure vessel due to change of nozzle location and shell thickness* International Journal of Advanced Engineering Research and Studies.

[7] L P Zick, *Stresses in large horizontal cylindrical pressure vessel on two saddle supports* welding journal research supplement September,vol.30,pp.435-445,1951

[8] Magnucki K, Stasiewicz P, Szyc W, *Flexible saddle support of a horizontal cylindrical pressure vessel* International Journal of Pressure Vessels and Piping,vol.80,pp.205-210,2003.

[9] Kacperski T, *Experimental study of the phenomenon of stability loss of a horizontal thin-walled cylindrical shell loaded by a liquid* Pol Acad. Sci. Eng. ,vol.42,pp.331-337,1994.

[10] Aleksandar petrovic, *Stress analysis in cylindrical pressure vessels with loads applied to the free end of a nozzle* International journal of pressure vessels and piping, ,vol.78,pp.485-493,2001.

[11] Xue M D, Hwang K C, Lu W, Chen W, *A reinforcement design method based on analysis of large openings in cylindrical pressure vessel* ASME Journal for pressure vessel, ,vol.118,pp.502-506,1996.

[12] Shafique M A Khan, *Stress distributions in a horizontal pressure vessel and the saddle supports* International Journal of Pressure Vessels and Piping, vol.87,pp.239-244,2010.

[13] N El Abbasi, S A Meguid, A Czekanski, *Three-dimensional finite element analysis of saddle supported pressure vessels* International Journal of Mechanical Sciences,vol.43,pp.1229-1242,2001.

[14] ASME Section II Part D: Material Properties (Metric), Newyork, 2010.

[15] ASME Section VIII div 1:Rules for Construction of Pressure Vessel, Newyork, 2010.

[16] Pipe schedule chart By ANSI/ASME B36.10M and API 5L.

[17] ASME B16.5: Pipe Flanges and Flanged Fittings, Newyork, 2003.

[18] Pressure vessel Design Manual by Dennis Moss,Gulf Professional Publication.

[19] Pressure vessel Handbook by E Megyesy,Pressure Vessel Publishing Inc, USA, 2001.

[20] Design Data Book for Engineers By Kalaikathir Achchagam, Coimbatore.