Mechanical Design and Analysis of Guard Drier(Pressure Vessel)

By

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DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May-2013

Mechanical Design and Analysis of Guard Drier(Pressure Vessel)

Major Project

Submitted in Partial Fulfillment of the Requirements For the degree of

MASTER OF TECHNOLOGY

IN

MECHANICAL ENGINEERING

(DESIGN ENGINEERING)

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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 Degree.

II. Due Acknowledgment has been made in the text to all other material used.

-Dhaval H Panchal

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Undertaking for Originality of the Work

I, **Dhaval H Panchal,** Roll No. **11MMED20,** give undertaking that the Major Project entitled **"Mechanical Design and Analysis of Guard Drier(Pressure Vessel) "** submitted by me, towards the partial fullfillment 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: NU, Ahmedabad

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Certificate

This is to certify that the Major Project entitled **"Mechanical Design and Analysis of Guard Drier(Pressure Vessel) "** submitted by **Mr.Dhaval H Panchal (Roll No: 11MMED20)** towards the partial fulfillment of the requirements for the degree of Master of Technology (Mechanical Engineering) in the field of **Design Engineering** of Nirma University is the record of work carried out by him under our supervision and guidance.The work submitted has in our opinion reached a level required for being accepted for examination. The results embodied in this major project work to the best of our knowledge have not been submitted to any other University or Institution for award of any degree or diploma.

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Acknowledgements

My words prove to be insufficient in thanking my thesis supervisors **Mr. Amrish Rami and Prof.Darshita J.Shah** without whose invaluable guidance, the preparation of this report indeed have been a herculean task. The suggestions given by them were invaluable and are sincerely considered by me.They have devoted significant amount of their valuable time to plan and discuss the thesis work.

I am deeply indebted to **HDO Technologies Ltd, Ahmedabad.** For giving me the opportunity to work here and providing me an excellent environment.I would like to express a deep sense of gratitude and thanks profusely to **Mr. L.Venkat Subbarrao** Vice President, at HDO Technologies Ltd, Vatva, Ahmedabad Without the wise counsel and able guidance, it would have been impossible to do the project in this manner.

As a token of my deepest gratitude, I would like to thank **Dr D S Sharma,** PG coordinator of M.Tech Design Engineering, **Dr R N Patel,** the Head of Mechanical Engineering. Dept. and **Dr K Kotecha,** Director, Institute of Technology, Nirma University for his kind support in all respects during my project.I express gratitude to other faculty members of Mechanical Engineering Department,for their intellectual support throughout the course of this work.

Finally, I am indebted to thank all whoever has contributed in this Project.

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Abstract

Guard Drier is a pressure vessel which widely used in process industries.It removes the moisture from the vessel.As these devices subjected to high pressure and temperature, it is necessary to design Guard Drier with special care. Under different loading conditions higher stresses will generate at critical location.These higher stresses regions may affect the efficiency of Guard Drier and may lead to dangerous results sometimes.So,it is highly recommended to design and analyze it with proper precautions.

The aim of the present work is to thoroughly mechanical design of Guard Drier manually as per ASME code section VIII Div I and analyze the dishend to nozzle junction and shell to nozzle junction. Then results of manual calculations are compared with PV Elite and also verified with FEA tool and obtain the stresses at critical location. This task is carried out by doing a theoretical calculation and making model in PV Elite.Thickness of shell, head, nozzle etc. and stresses are obtained which is helpful in making 3D model of dishend to nozzle junction and shell to nozzle junction and subsequent finite element analysis of Guard Drier.Results obtained from theoretical calculations, PV Elite Calculations and analysis performed is within the permissible limits and variations are not exceeding more than five percentage. This shows that the adopted procedure gives the satisfactory results.

Keywords: Guard Drier(Pressure Vessel),Dishend to nozzle junction,Shell to nozzle junction

Contents

List of Figures

List of Tables

Nomenclatures

P	Internal Design Pressure,kg/cm^2
D	Inside Diameter of shell,mm
D_i	Inside Diameter of shell-corroded,mm
E	Joint Efficiency
C	Internal Corrosion Allowance,mm
S	Max. Allowable Stress at Design Temperature,kg/cm^2
t	Nominal thickness,mm
D_o	Outside Diameter of shell,mm
t_r	Required Thickness,mm
t_{rn}	Required Thickness of nozzle wall,mm
d	Finished diameter of circular opening,mm
S_n	Allowable Stress-nozzle material,kg/cm^2
S_v	Allowable Stress-head material,kg/cm^2
S_p	Allowable Stress in reinforcing element,kg/cm^2
D_n	Outside Diameter of nozzle,mm
D_p	Outside Diameter of reinforcig element,mm
A_r	Reinforcement Area Required for Nozzle,mm^2
A_1	Area Available in Shell,mm^2
A_2	Area Available in Nozzle Wall,mm^2
A_3	Area Available in Inward Nozzle,mm^2
A_4	Area Available in Welds,mm^2
A_5	Area Available in Element,mm^2
F	Correction Factor
t_e	Thickness or height of reinforcing element,mm
C_n	Corrosion Allowance for nozzle,mm
W_p	Weld Leg at pad,mm
W_o	Outside Fillet Weld,mm

Chapter 1

Introduction

1.1 Preliminary Remarks

Gaurd Drier is widely used in process industries.It is a pressure vessel only which remove the moisture from the vessel so it is called "Guard Drier".Process industries require the handling and storing of large quantities of materials in containers of varied construction, depending upon the existing state of the material, its physical and chemical properties, and the required operations which are to be performed. For handling such liquids and gases a container or vessel is used. Vessels, tanks, and pipeline that carry, store or receive fluids are called pressure vessels. A pressure vessel is defined as a container with a pressure differential between inside and outside. The inside pressure is usually higher than the outside pressure. Pressure Vessels are used in variety of industries oil and gas, petroleum refining, chemical, fertilizer, food, nuclear etc.

1.2 Objective

In industries people faces many problems regarding the discontinuous junction in Pressure Vessels.Pressure Veseel have many discontinuous regions in their structure such as manhole connections, nozzles, supports, joints etc. They are also subjected to die rent loadings such as internal pressure, external pressure, thermal loads etc. Because of these discontinuous regions there will be chance to failure of the whole structure. So, a reliable and accurate design and analysis is necessary.

1.3 Aim and Scope of Work

The aim of the present work is to thoroughly mechanical design of Guard Drier manually as per ASME sec VIII division I.Results of manual calculations are compared with PV-Elite and also verified with FEA tool and then obtain the stresses at critical location.The model is done with the help of PV Elite. The theoretical calculation is done with the help of ASME (American Society of Mechanical Engineer) code sec VIII div I. The theoretical calculation is verified with PV Elite calculation.The solid model of the dishend to nozzle junction and shell to nozzle junction is prepared in PRO-E and finite element analysis is carried out in ANSYS.

1.4 Thesis Organization

Chapter 2 includes the overview of the literature which has been reviewed during project work.
Chapter 3 covers theoretical calculation and PV Elite calculation of Guard Drier (Pressure Vessel).
Chapter 4 Results and Discussion
Chapter 5 Conclusion and Future Scopes

Chapter 2

Literature Review

2.1 Introduction

A pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. The pressure differential is dangerous and many fatal accidents have occurred in the history of pressure vessel development and operation. Consequently, pressure vessel design, manufacture, and operation are regulated by engineering authorities backed by legislation. For these reasons, the definition of a pressure vessel varies from country to country, but involves parameters such as maximum safe operating pressure and temperature.

Pressure vessels are used in a variety of applications in both industry and the private sector. They appear in these sectors as industrial compressed air receivers and domestic hot water storage tanks. Other examples of pressure vessels are diving cylinders, recompression chambers, distillation towers, pressure reactors, autoclaves, and many other vessels in mining operations, oil refineries and petrochemical plants, nuclear reactor vessels, submarine and space ship habitats, pneumatic reservoirs, hydraulic reservoirs under pressure, rail vehicle airbrake reservoirs, road vehicle airbrake reservoirs, and storage vessels for liquefied gases such as ammonia, chlorine, propane, butane, and LPG.

Theoretically almost any material with good tensile properties that is chemically stable in the chosen application could be employed. However, pressure vessel design codes and application standards (ASME Section VIII Div.1.) contain long lists of approved materials with associated limitations in temperature range.

Many pressure vessels are made of steel. To manufacture a cylindrical or spherical pressure vessel, rolled and possibly forged parts would have to be welded together. Some mechanical properties of steel achieved by rolling or forging, could be adversely affected by welding, unless special precautions are taken. In addition to adequate mechanical strength, current standards dictate the use of steel with a high impact resistance, especially for vessels used in low temperatures. In applications where carbon steel would suffer corrosion, special corrosion resistant material should also be used.

Some pressure vessels are made of composite materials, such as filament wound composite using carbon fiber held in place with a polymer. Due to the very high tensile strength of carbon fiber these vessels can be very light, but are much more difficult to manufacture. The composite material may be wound around a metal liner, forming a composite overwrapped pressure vessel. Other very common materials include polymers such as PET in carbonated beverage containers and copper in plumbing.Pressure vessels may be lined with various metals, ceramics, or polymers to prevent leaking and protect the structure of the vessel from the contained medium. This liner may also carry a significant portion of the pressure load.

Pressure Vessels may also be constructed from concrete (PCV) or other materials which are weak in tension. Cabling, wrapped around the vessel or within the wall or the vessel itself, provides the necessary tension to resist the internal pressure. A

”leak proof steel thin membrane” lines the internal wall of the vessel. Such vessels can be assembled from modular pieces and so have ”no inherent size limitations”. There is also a high order of redundancy thanks to the large number of individual cables resisting the internal pressure.

2.2 Parts of Guard Drier(Pressure Vessel)

There are four main parts of Guard Drier(Pressure Vessel) shown in figure 2.1,

- Shell
- Head
- Support
- Nozzle

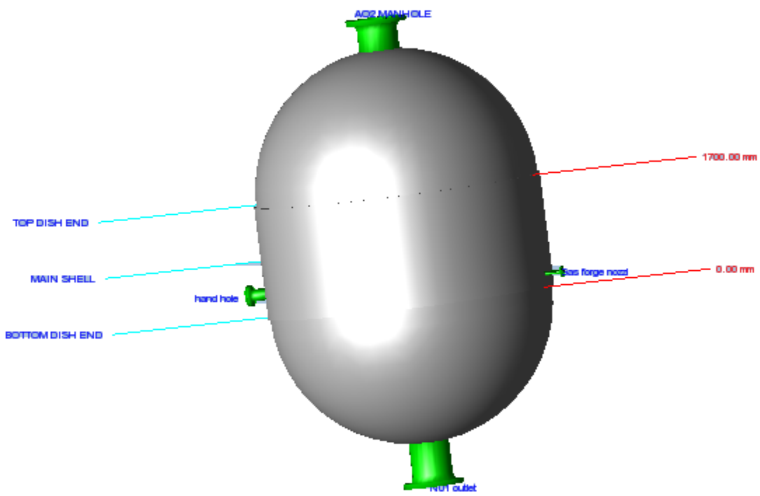

Figure 2.1: Parts of Gaurd Drier(Pressure Vessel)

2.2.1 Shell

It is a major component of Pressure Vessel.There are two types of shell.

- Cylindrical Shell
- Spherical Shell

2.2.2 Head

Head is the end closer of Pressure Vessel. Types of head are shown in figure 2.2, Types of head:

- Hemispherical head
- Torispherical head or Ellipsoidal head
- Conical head
- Flat head

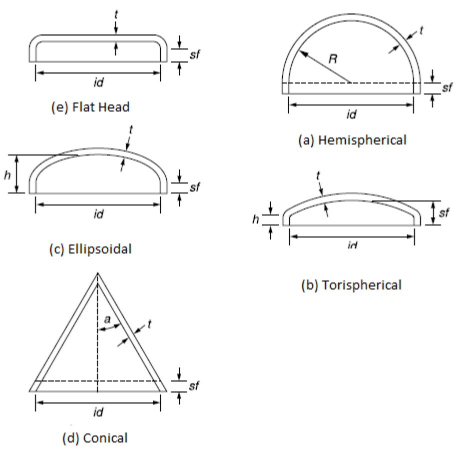

Figure 2.2: Types of Head

2.2.3 Support

Supports are used for vertical or horizontal pressure vessel. Types of supports are Shown in figure 2.3,

- Skirt (straight or conical)
- Saddles (attached or loose)
- Rings
- Lugs

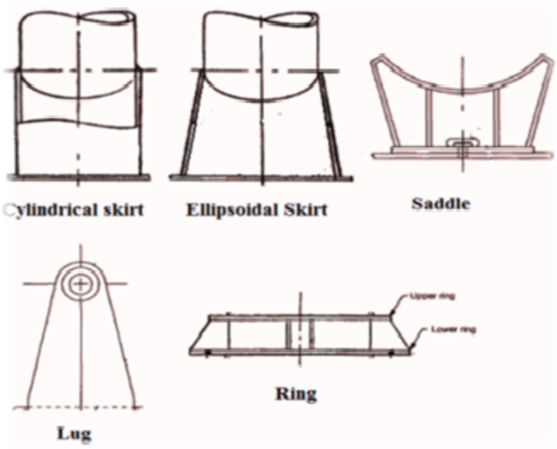

Figure 2.3: Types of Support

2.2.4 Nozzle

A nozzle is often a pipe or tube of varying cross sectional area and it can be used direct or modified the fluid (liquid or gas). Nozzles are frequently used to control throw rate, speed, direction, mass, shape and the pressure of the stream that emerges from them. Types of nozzle are shown in figure 2.4,

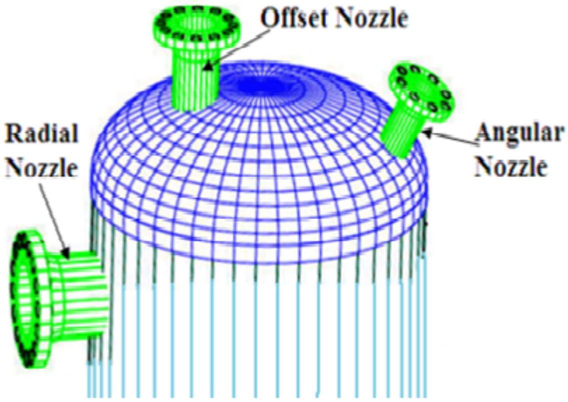

Figure 2.4: Types of Nozzle

- **Radial nozzle:** A nozzle which is perpendicular to orientation line is called radial nozzle.
- **Offset nozzle:** A nozzle which has distance from the orientation line is called offset nozzle.
- **Angular nozzle:** A nozzle which is at some angle from orientation line is called angular nozzle.

Here,**Guard Drier(Pressure Vessel)**has,

- one main cylindrical shell
- top and bottom dish end which is spherical
- two radial nozzle and two offset nozzle

which we can see in the figure 2.1

Q. Zhang, Z.W.Wang, C.Y.Tang, D.P.Hu, P.Q.Liu, L.Z.Xia [**3**] has been reported on determining the thermo-mechanical stresses in a multilayered composite pressure vessel when the influence of its closed ends is considered. In this study, an analytical solution was derived for determining the stress distribution of a multilayered composite pressure vessel subjected to an internal fluid pressure and a thermal load, based on thermo-elasticity theory. In the solution, a pseudo extrusion pressure was proposed to emulate the effect of the closed ends of the pressure vessel. To validate the analytical solution, the stress distribution of the pressure vessel was also computed using finite element (FE) method. It was found that the analytical results were in good agreement with the computational ones, and the effect of thermal load on the stress distribution was discussed in detail. The proposed analytical solution provides an exact means to design multilayered composite pressure vessels.

A.Th.Diamantoudis ,Th.Kermanidis [4] study for design by analysis and design by formula of a cylinder to nozzle intersection has been made using different finite element techniques. The cylinder to nozzle intersection investigated is part of a typical vertical pressure vessel with a skirt support. For the study the commonly used ductile P355 steel alloy and the high strength steel alloy P500 QT were considered. The comparative results clearly show disadvantages in terms of limit load capability when the design-by-formula procedures are used in the design of high strength steel pressure vessels. The FE results also clearly show advantages of the shell to solid sub-modelling technique, as it combines the accuracy of 3D-solid modelling with the affordable computing time of the 3D-shell modelling technique.

R.C.Carbonari,P.A.Munoz-Rojas ,E.Q.Andrade ,G.H.Paulino,K.Nishimoto, E.C.N.Silva [5] the optimization of pressure vessels has considered the optimization of the nozzle independently from the dished end. This approach generates problems such as thickness variation from nozzle to dished end (coupling cylindrical region) and, as a consequence, it reduces the optimality of the final result which may also be influenced by the boundary conditions. Thus, this work discusses shape optimization of axis symmetric pressure vessels considering an integrated approach in which the entire pressure vessel model is used in conjunction with a multi-objective function that aims to minimize the von-Mises mechanical stress from nozzle to head. Representative examples are examined and solutions obtained for the entire vessel considering temperature and pressure loading. It is noteworthy that different shapes from the usual ones are obtained. Even though such different shapes may not be profitable considering present manufacturing processes, they may be competitive for future manufacturing technologies, and contribute to a better understanding of the actual influence of shape in the behaviour of pressure vessels.

H.Darijiani,M.H.Kargarnovin,R.Naghdabadi [**6**] By considering the Bauschinger effect and the yield criterion of Tresca, an exact elasto-plastic analytical solution for a thick-walled cylindrical vessel made of elastic linear-hardening material is derived. Having the working pressure and geometric dimensions of the vessel, the distribution of the hoop and equivalent stresses are optimized in the way that the distribution of stresses becomes smooth in the vessel wall. Based on two optimizing methods of the hoop and equivalent stresses, the best autofrettage pressure is determined. It is shown that this pressure is more than the working pressure and depends on the three following variables: Bauschinger effect, working pressure and geometric dimensions. In the next stage, the main task is to determine the wall thickness having the working pressure. To do this, two different design criteria namely; (i) optimizing the hoop stress distribution and (ii) assuming a suitable percent of yielding in the wall thickness are used. In the last step, for different types of structural materials under different working pressures, a number of different plots are given for the ratio of outer to inner radii and the best autofrettage pressure. It is shown that the design of vessels based on the elasto-plastic methods is much more economic than elastic methods. Also, it is seen that for a non-hardening material, the design of vessel is only done for the working pressure less than unit value.

N. Jiang, L. Zhen, B.P. Xu [**7**] analyzes the limits of secondary stress strength and discusses the influence of the Bauschinger effect (BE) of metallic materialism the analytical design code. It is proved by this research work that the allowance value of secondary stress in the analytical design code is obtained on account of that of the plastic materials in stable condition, and its value is larger than the practical materials which was affected by BE. Therefore the allowance value of secondary stress strength in the analytical design code should take the influence of BE into account, and it can be corrected by PL + Pb + Q $\leq$ 1:5(1 + f) Sm (f stands for the value of BE and ($0 < f < 1$). Otherwise, if the secondary stress is designed on the basis of a limit condition of stress strength PL + Pb + Q $\leq$ 3Sm, there would be some potential inadequacy of safety margin in pressure vessels.

Petrovic Aleksandar [**8**] has discussed stress analysis of cylindrical pressure vessel loaded by axial and transverse forces on the free end of a nozzle. The nozzle is placed such that the axis of the nozzle does not cross the axis of the cylindrical shell. The method of finite elements was applied to determine the state of stress in the cylindrical shell. The value obtained for the stress in the nozzle region was used to determine the following: envelope for maximum stress values; maximum values of this envelope;and distance between maximum values on the envelope and outer edge of the nozzle.

Chapter 3

Theoritical Calculation

3.1 Classification of Stress

3.1.1 Stress Applied

The total elastic stress, which occurs in the vessel shell, is considered to be composed of three different type of stress primary, secondary and peak. In addition primary stress has three specific sub categories. The ASME stress categories and the symbols used to denote them in the code are given below [9],

a. Primary Stress

- General primary membrane stress, Pm
- Local primary membrane stress, PL
- Primary banding stress, Pb

b. Secondary Stress, Q

c. Peak Stress, F

Normal Stress :

The normal stress is the component of the stress normal to the plane of the reference;this is also referred to as a direct stress. Usually the distribution of the stress is not uniform through the thickness of the part, so this stress is considered to be made up in turn of two components one which is uniformly distributed and equal to the average value of stress across the thickness of the section under the consideration, and the other of which varies with the location across the thickness.

Shear Stress :

The shear stress is the component of the stress acting in the plane of the reference.

Membrane Stress :

The membrane stress is the component of stress that is uniformly distributed and equal to the average value of the stress across the thickness of the section under consideration.

Bending Stress :

The bending stress is the component of stress that varies linearly across the thickness of the section under consideration. With this terminology as background, we now can define primary, secondary and peak stress properly.

3.1.2 Stress Categorization

Primary Stress:

A primary stress is a stress produced by the mechanical loading only and it is so distributed in the structure that no redistribution of load occurs as a result of yielding.The basic characteristic of the stress is that it is not self-limiting. Primary stresses that considerably exceed the yield strength will result in failure, or at least in gross

distortion. A thermal stress is not classified as primary stress.
Typical examples of general primary stresses are:

- The average stress due to internal pressure or to distributed live loads;
- The bending stress of a at cover without supporting moment at the periphery due to internal pressure.

Primary stresses are divided in to the 'general' and 'local' categories.

Primary Local Membrane Stress:

Case arise in which the membrane stress produced by pressure or other mechanical loading and associate with a primary together with a discontinuity effect produces excessive distortion in the transfer of load to the other portion of the structure. An example of primary local stress is the membrane stress in a shell produce by external load and moment at a permanent support or at a nozzle connection.

Secondary Stresses :

Secondary stresses are stresses developed by constraints due to geometric discontinuities, by the use of the material of the different elastic modulus under external loads,or by constraints due to differential thermal expansion. The basic characteristic of the stress is that it is self-limiting. Examples are the bending stresses at dished end to shell junction, generally thermal stresses.

Peak Stresses :

Peak stress is that increment of stress which is additive to the primary to the secondary stresses by reason of local discontinuities or local thermal stress including the effect of stress concentration. The basic characteristic of peak stress is that they do not cause any noticeable distortion and only important to fatigue and brittle fracture in conjunction with primary and secondary stress.

3.2 Theoritical Calculation

As per ASME CODE SECTION VII DIVISION-1 EDITION-2007, ADDENDA 2009[9]

Table 3.1: Design Data

Length T.L to T.L	1600mm
Vessel inside diameter	4200mm
Design Pressure	$42.31 kg/cm^2$
Operating Pressure	$37.72 kg/cm^2$
Design Temperature	$250°C$
Operating Temperature	$220°C$
Corrosion Allowance	3mm
Joint Efficiency	1

Material of Constuction :

- Shell, RF Pad, Nozzle Neck : SA 516 GR 70
- Flange : SA 350 GR LF2 CL1

Table 3.2: Material Properties

Material	Young's modulus,MPa	Poisson's ratio	Density kg/m^3	Tensile Ultimate Strength,MPa	Max. Allowable Stress,MPa
SA 516 GR 70	195000	0.3	7850	485	138

3.2.1 Internal Pressure Calculation (Main Shell) :

Design Inputs as per UG-27,

Thickness due to internal pressure t_r:

$$t_r = \frac{P \times Ri}{S \times E - 0.6 P} = 64.3917 mm$$

$$t_d = designthickness = t_r + C = 67.3917 mm$$

Max. Allowable Working Pressure at given thickness (MAWP):

$$MAWP = \frac{S \times E \times (t-c)}{\mathrm{Ri+0.6\times(t-c)}} = 43.9916\ kg/cm^2$$

Max. Allowable Pressure, new and cold (MAPNC):

$$MAPNC = \frac{S \times E \times t}{R+0.6 \times t} = 45.9871\ kg/cm^2$$

Actual Stress at given pressure and thickness (S_a):

$$S_a = \frac{P \times (Ri+0.6 \times (t-c))}{E \times (t-c)} = 1353.4148\ kg/cm^2$$

3.2.2 Internal Pressure Calculation (Spherical Dish End) :

Thickness due to internal pressure t_r:

$$t_r = \frac{P \times Ri}{2 \times S \times E - 0.2 \times P} = 31.7104 \text{ mm}$$

$$t_d = designthickness = t_r + C = 34.7104mm$$

Max. Allowable Working Pressure at given thickness (MAWP):

$$MAWP = \frac{2 \times S \times E \times (t-c)}{Ri+0.2 \times (t-c)} = 49.3429kg/cm^2$$

Max. Allowable Pressure, new and cold (MAPNC):

$$MAPNC = \frac{2 \times S \times E \times t}{R+0.2 \times t} = 53.4044kg/cm^2$$

Actual Stress at given pressure and thickness (S_a):

$$\mathrm{Sa} = \frac{\mathrm{P\times Ri+0.2\times(t-c)}}{\mathrm{2\times E\times(t-c)}} = 1202.5044 kg/cm^2$$

3.2.3 Nozzle Calculation :

Design Inputs per UG-37,

Required thickness of Spherical Head(t_r):

$$t_r = \frac{P\times Ri}{2\times S\times E-0.2\times P} = 31.7104 \text{ mm}$$

Required thickness of Nozzle Wall (t_{rn}):

$$\mathrm{trn}=\frac{\mathrm{P\times Rn}}{\mathrm{S\times E+0.4\times P}}= 9.0554 \text{ mm}$$

Finished diameter of circular opening(d):

$$d = Dn - 2 \times tn = 575.6 \text{ mm}$$

$$f_{r1} = MIN\left(\frac{S_n}{S_v}\ ,\ 1\right) = 1$$

$$f_{r2} = MIN\left(\frac{S_n}{S_v}\ ,\ 1\right) = 1$$

$$f_{r3} = MIN\left(\frac{S_n}{S_v}\ ,\ \frac{S_p}{S_v}\ ,\ 1\right) = 1$$

$$f_{r4} = MIN\left(\frac{S_p}{S_v}\ ,\ 1\right) = 1$$

Reinforcement Area Required for Nozzle (A_r):

$$A_r = d \times \ t_r \ \times F \ + 2 \times \ t_n \ \times \ t_r \ \times F \ \times (1 - \ f_{r1}) = 18252.276 \ mm^2$$

Area Available in Shell (A_1):

$$A_1{}^0 = d \ \times (E \times t - F \times t_r) - 2 \times t_n \times (E \times t - F \times t_r) \times (1 - f_{r1})$$

$=3044.924 mm^2$

$$A_1{}^1 = 2 \times (t + t_n) \times (E \times t - F \times t_r) - 2 \times t_n \times (E \times t - F \times t_r) \times (1 - f_{r1})$$

$=571.32 mm^2$

$$A_1 = MAX \ \left(A_1{}^0 \ , \ A_1{}^1\right) = 3044.924 \ mm^2$$

Area Available in Nozzle Wall (A_2):

$$A_2{}^0 = 5 \times (t_n - t_{rn}) \times f_{r2} \times t = 1470.75 \ mm^2$$

$$\mathrm{A_2{}^1 = 5 \times (t_n - t_{rn}) \times f_{r2} \times t_n = \ 675.75 \ mm^2}$$

$$A_2 = MIN \ \left(A_2{}^0 \ , \ A_2{}^1\right) = 675.75 \ mm^2$$

Area Available in Inward Nozzle (A_3):

A_3=0

Area Available in Welds (A_4):

$$A_{41} = \ {W_o}^2 \times f_{r3} = 81\ mm^2$$

$$A_{42} = \ {W_p}^2 \times f_{r4} = 196\ mm^2$$

$$A_{43} = \ W_i^2 \times f_{r2} = 0$$

$$A_4 = \ A_{41} + A_{42} + A_{43} = 277\ mm^2$$

Area Available in Element(A_5):

$$A_5 = (MIN\,(D_p\ , D_l) - Nozzle\ OD) \times MIN\,(t_p, t_{lwp}) \times f_{r4} = 19616\ mm^2$$

Now,

$A_1 + A_2 + A_3 + A_4 + A_5 = 23613.674\ mm^2$

$A_1 + A_2 + A_3 + A_4 + A_5 > A_r$

$23613.674 > 18252.276$

Therefore Opening is Adequately Reinforced.

Minimum Nozzle Neck Thickness Requirement :

Wall Thickness per UG-45 (a)

$$t_{ra} = \ t_{rn}\ + C_n = 12.05\ mm$$

Wall Thickness per UG-16 (b)

$$t_{r16b} = \ \frac{1}{16}\ in\ + C_n = 4.5875\ mm$$

Wall Thickness per UG-45 (b) (1)

$$t_{rb1} = \ t_r + C = 34.71 \ mm$$

Wall Thickness per UG-45 (b) (2)

$$t_{rb2} = \ t_r \left(Ext.pre\right) + C = 3 \ mm$$

Wall Thickness per UG-45 (b) (3)

$$t_{rb3} = MAX \left(t_{rb1}, t_{rb2}, t_{r16b}\right) = \ 34.71 \ mm$$

Std. Wall Pipe per UG-45 (b)(4)

$$t_{rb4} = 8.334 + C_n = 11.334 \ mm$$

Wall Thickness per UG-45 (b)

$$t_{rb} = MIN \left(t_{rb3}, t_{rb4}\right) = \ 11.334 \ mm$$

Final Required Thickness

$$t_{UG-45} = MAX \left(t_{ra}, t_{rb}\right) = \ 12.05 \ mm$$

Available Nozzle Neck Thickness = 20 mm

Available Nozzle Neck Thickness > Required Thickness

3.2.4 PV Elite Model:

Client gives the specification (i.e. design data) of Guard Drier(Pressure Vessel) to the company as shown in figure 3.1,

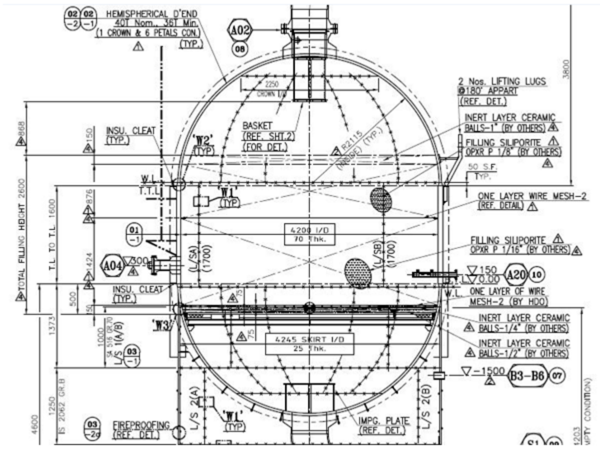

Figure 3.1: Data Sheet

According to the specification and with the help of ASME code, I prepared the PV Elite model of Guard Drier(Pressure Vessel) as shown in figure 3.2,

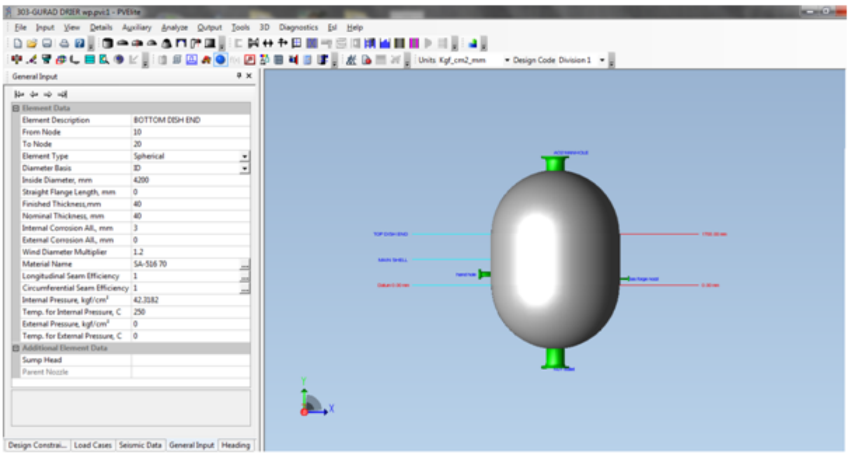

Figure 3.2: PV Elite Model

PV Elite Calculation :

For Guard Drier(Pressure Vessel),internal pressure calculations, nozzle calculation are shown in respectively table 3.3 and 3.4,

Table 3.3: Internal Pressure Calculation

	Thickness mm	MAWP kg/cm^2	MAPNC kg/cm^2	Stress kg/cm^2
Cylindrical Shell	67.45	43.95	45.95	1353.67
Spherical Head	34.74	49.30	53.36	1206.87

Table 3.4: Nozzle Calculation

	Thickness-cylindrical shell,mm	Thickness-nozzle wall,mm
Nozzle	31.74	9.06

Chapter 4

Results and Discussion

4.1 Comparison of PV Elite and Theoretical Calculation :

The calculation of the thickness, pressure, stresses etc. are done by using PV Elite software and ASME code. The comparisons are shown in the following tables 4.1,4.2 and 4.3.
Comparing the manual calculations with the PV Elite calculations, the results are shown in the table. The variations of thickness, pressure and stress of the internal pressure, nozzle are much lower and it is within the limit. There isn't any part that is fail due to that.

Table 4.1: Comparison of Internal pressure (Main Shell)

	PV Elite Calculation	Theoretical Calculation	Variation(%)
Thickness (mm)	67.4543	67.3917	0.0009280
MAWP(kg/cm^2)	43.958	43.9916	0.0007643
MAPNC(kg/cm^2)	45.952	45.9871	0.0007638
Stress(kg/cm^2)	1353.678	1353.414	0.0001950

Table 4.2: Comparison of Internal pressure (Spherical Head)

	PV Elite Calculation	Theoretical Calculation	Variation,(%)
Thickness (mm)	34.7407	34.7104	0.0008721
MAWP(kg/cm^2)	49.306	49.3429	0.0007483
MAPNC(kg/cm^2)	53.364	53.4044	0.0007570
Stress(kg/cm^2)	1206.870	1202.5044	0.003617

Table 4.3: Comparisson of Nozzle

	PV Elite Calculation	Theoretical Calculation	Variation,(%)
Thickness-cylindrical shell(mm)	31.7407	31.7104	0.0009546
Thickness-nozzle wall(mm)	9.0639	9.0554	0.0009377

4.2 Finite Element Analysis:

4.2.1 Finite Element Model

- **Computer Program**

  The modelling is performed in PRO-E version 5,and analysis is performed using finite element analysis computer program ANSYS version 11.

- Design Data,Material of construction and Material properties are given in chapter 3,all the values of taken from there.

- **Geometric Model Of Dishend To Nozzle Junction**

  A 3D model of dishend to nozzle junction is considered with weld.The model comprises intersections dishend,shell and nozzle.A geometric model is shown in figure The dimensions are as follows:

  ID of Dishend = 4230 mm

  Thickness of dishend = 40 mm

ID of Nozzle = 609.6 mm

Thickness of nozzle = 20 mm

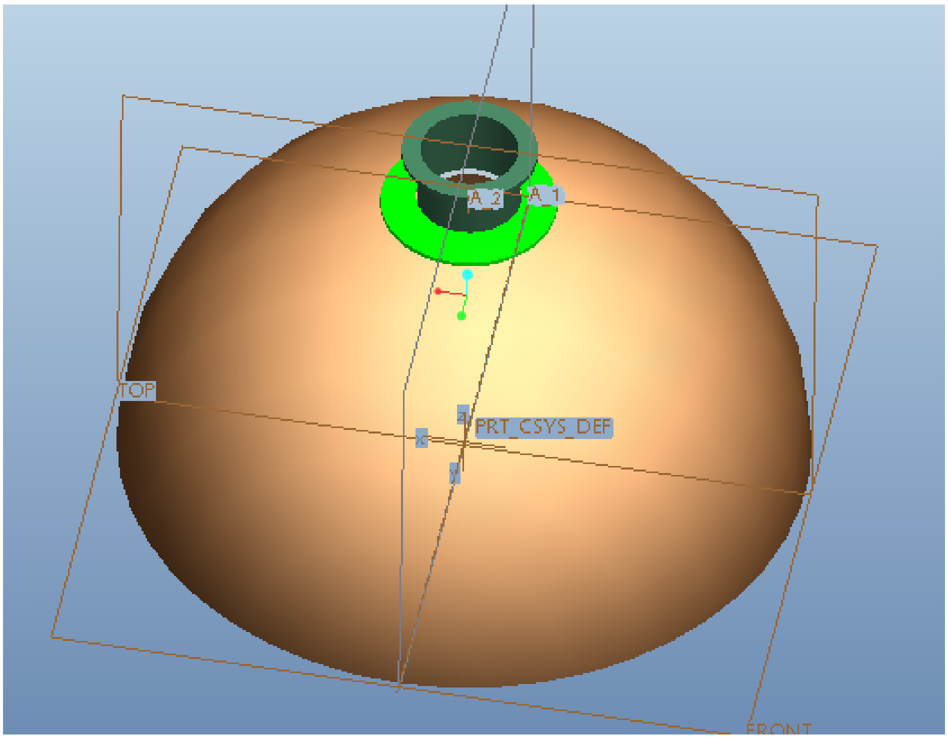

Figure 4.1: Geometric Model

- **Finite Element Mesh**

  The characteristics of the elements used for finite element analysis as follows: SOLID187 element is a higher order 3-D, 10-node element. SOLID187 has a quadratic displacement behavior and is well suited to modeling irregular meshes (such as those produced from various CAD/CAM systems). The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions. The element has plasticity, hyperelasticity, creep, stress stiffening, large deection, and large strain capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elastoplastic materials, and fully incompressible hyperelastic materials.

Type of element : 3-D 10-Node Tetrahedral Structural Solid (Solid 187)

Total no. of elements in the model :16424

Total no.of nodes in the model :32394

Finite Element Meshing figure 4.2 is as follows:

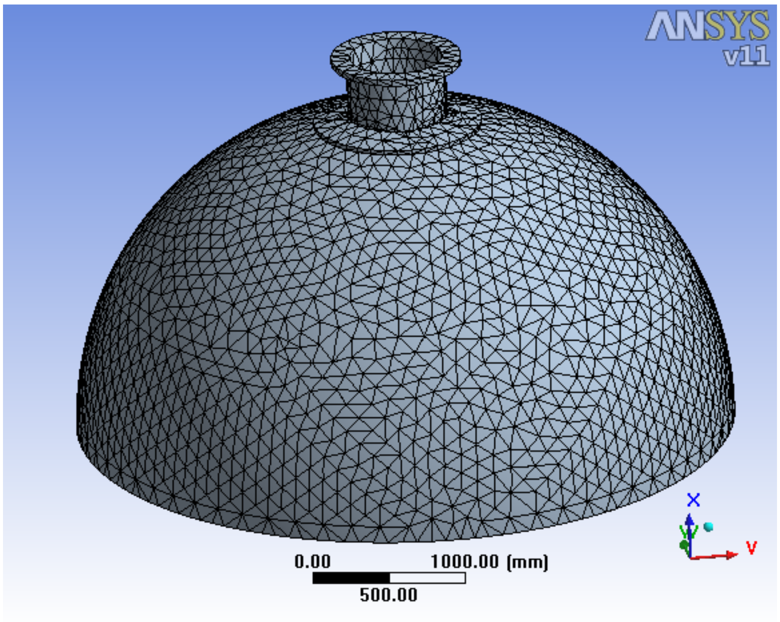

Figure 4.2: Finite Element Meshing

- **Mechanical Boundary Condition**

  For evaluating the stresses due to the structural loads, the periphery of the dishend is fixed and 4.15 MPa pressure is given inside and it is normal to the dishend inner diameter.

- **Static Structural Analysis**

  A static analysis calculates the effects of steady loading conditions on a structure,while ignoring inertia and damping effects, such as those caused by time-varying loads. A static analysis can, however, include steady inertia loads (such as gravity and rotational velocity), and time-varying loads that can be approximated as static equivalent loads (such as the static equivalent wind and seismic loads commonly defined in many building codes). Static analysis determines the displacements, stresses,strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time. The types of loading that can be applied in a static analysis include:

1.Externally applied forces and pressures

2.Steady-state inertial forces (such as gravity or rotational velocity)

3.Imposed (nonzero) displacements

4.Temperatures (for thermal strain)

According to that stresses are generated,von-mises stress figure 4.3,maximum principal stress figure 4.4 are as follows.

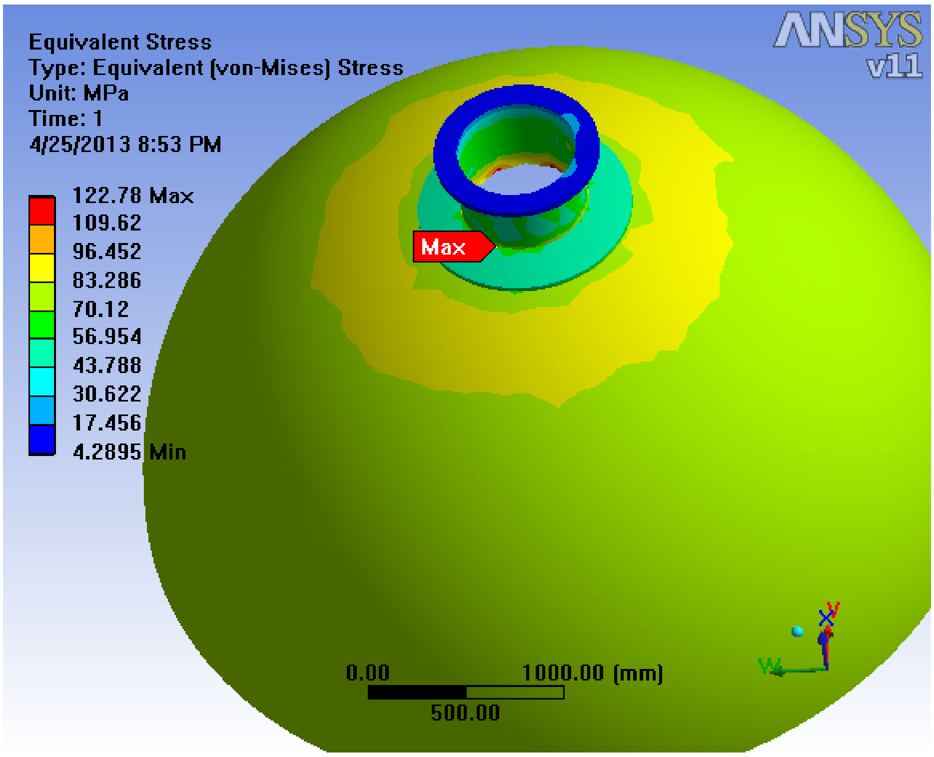

Figure 4.3: Von-Mises Stress

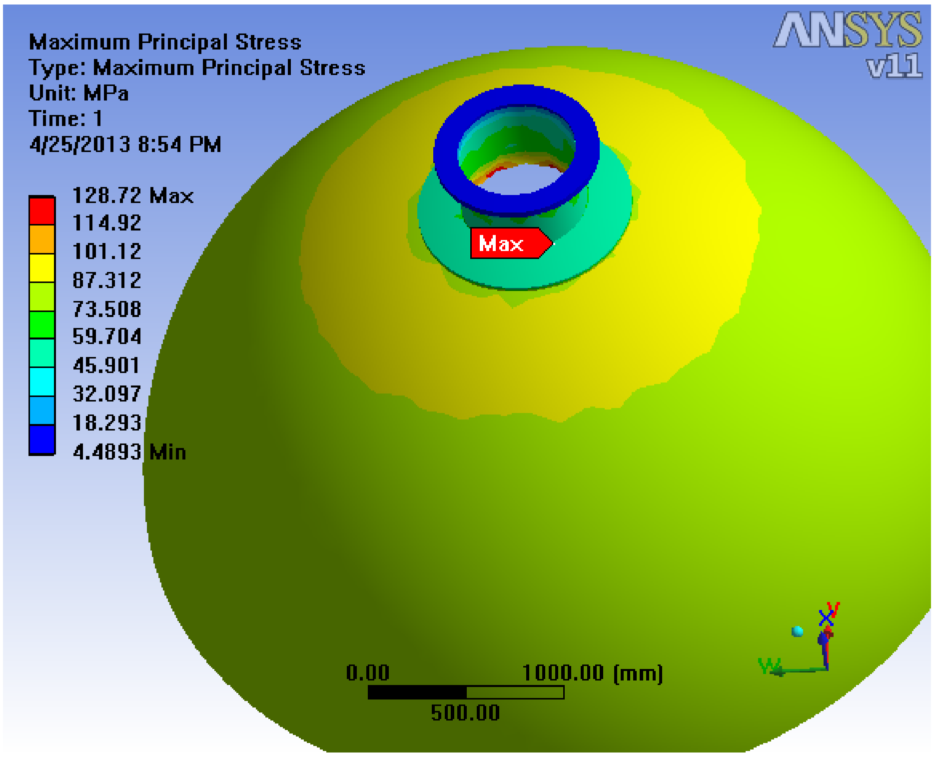

Figure 4.4: Max.Principal Stress

Results obtained from finite element analysis are shown in the following table,

Table 4.4: FEA Results for dishend to nozzle junction

Stresses	Obtained MPa	Permissible MPa
Von-Mises	122	138
Max.Principal	128	138

- **Geometric Model Of Shell To Nozzle Junction**

  A 3D model of shell to nozzle junction is considered with weld.The model comprises intersections dishend,shell and nozzle.A geometric model is shown in figure 4.5,The dimensions are as follows:

  ID of Shell = 4200 mm

  Thickness of Shell = 70 mm

  ID of Nozzle = 150 mm

  Thickness of nozzle = 12 mm

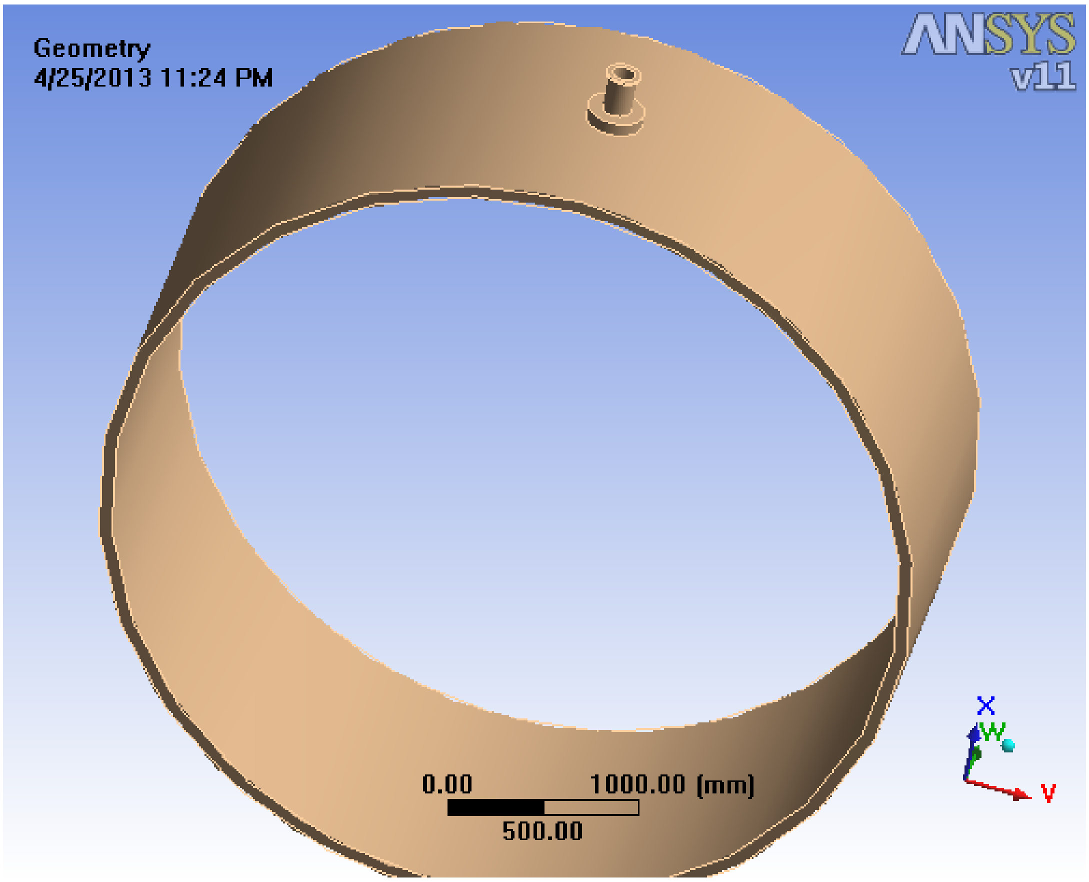

Figure 4.5: Geometric Model

Type of element : 3-D 10-Node Tetrahedral Structural Solid (Solid 187)

Total no. of elements in the model :5247

Total no.of nodes in the model :10521

Finite Element Meshing figure4.6 is as follows:

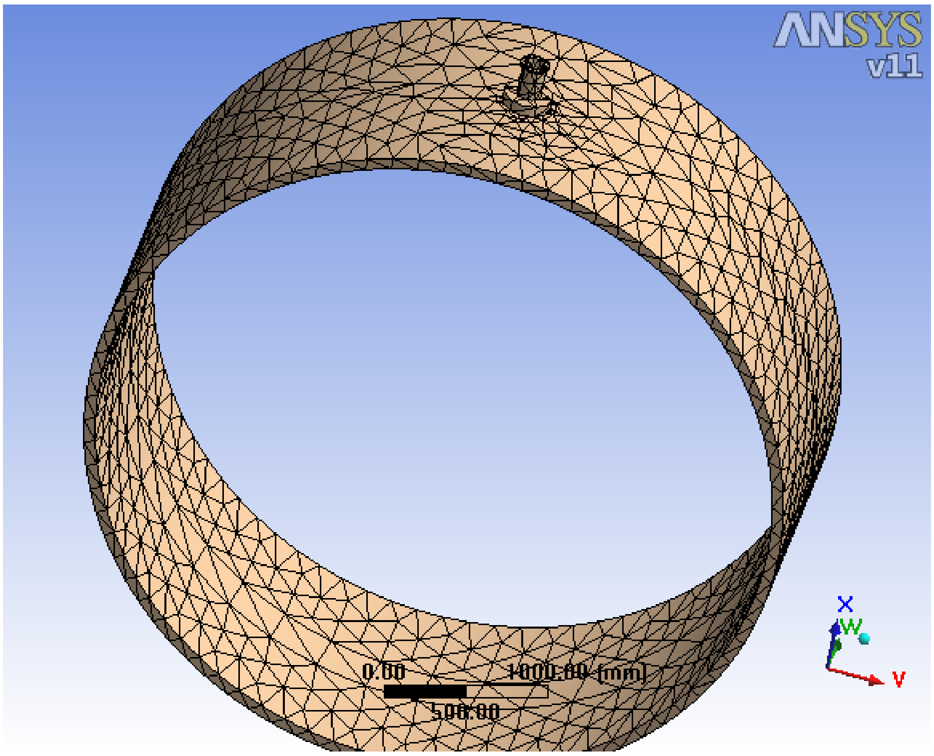

Figure 4.6: Finite Element Meshing

- **Mechanical Boundary Condition**

  For evaluating the stresses due to the structural loads, the left and right periphery of the shell are fixed and 4.15 MPa pressure is given inside and it is normal to the shell inner diameter.

  From the static structural analysis stresses are generated,von-mises stress figure 4.7 and max.principal stress figure 4.8 are as follows:

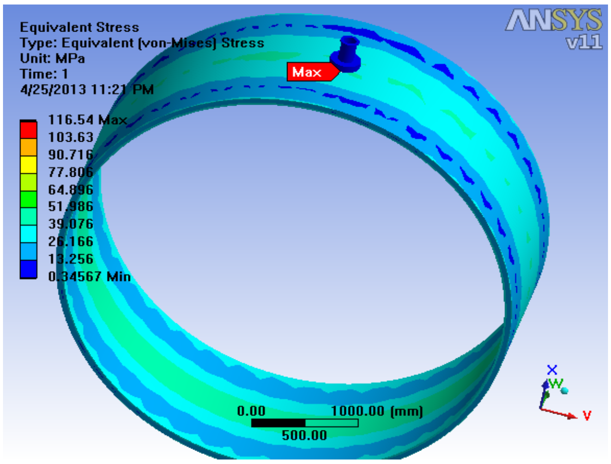

Figure 4.7: Von-Mises Stress

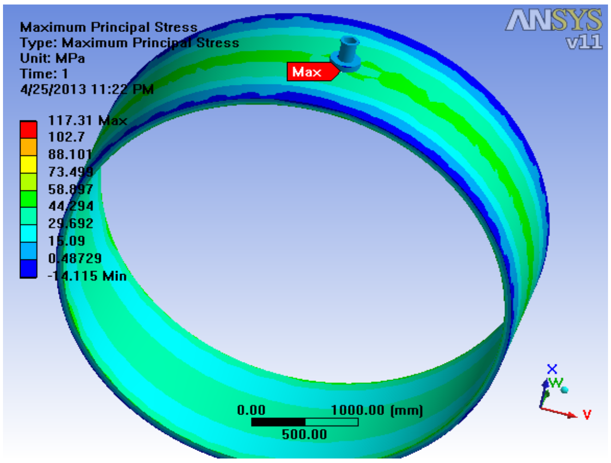

Figure 4.8: Max.Principal Stress

Results obtained from finite element analysis are shown in the following table,

Table 4.5: FEA Results for shell to nozzle junction

Stresses	Obtained MPa	Permissible MPa
Von-Mises	116	138
Max.Principal	117	138

Here,we can say that from table 4.4 and 4.5,stresses obtained from the static structural analysis are quite less than the permissible limit.So the design and analysis of dishend to nozzle junction and shell to nozzle junction of Guard Drier is safe for the given working conditions.Also it is verified that adopted procedure gives satisfactory results within the safe working limits.

Chapter 5

Conclusion and Future Scopes

5.1 Conclusion

- The calculation of the thickness,stresses etc. has been done by using ASME code and PV Elite software. After comparing theoretical calculation and PV Elite calculation, the variation of the thickness and stresses are much lower and it is within the limit.

- The solid model of the dishend to nozzle junction and shell to nozzle junction is prepared by considering weld at the junction using PRO-E. The weak section of Guard Drier(Pressure Vessel) i.e nozzle to dish-end junction and nozzle to shell junction is analyzed with finite element analysis method using ANSYS.

- Results obtained from theoretical calculations and analysis performed is within the permissible limits and variations are not exceeding more than five percentage.

5.2 Future Scopes

- Here,the complete static structural analysis of Guard Drier(pressure vessel) at dishend to nozzle junction and shell to nozzle junction has analyzed.But also other junctions like shell to dishend and shell to saddle can be done.

- Work can also be extend to design and analysis of pressure vessel under external loading conditions.

- Zick analysis of supported saddle can be done.

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