DISSERTATION

Design and Analysis of a Hyperbaric Chamber

By

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the

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Approved by,

____

(Ir. Idris B. Ibrahim)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

September 2012

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and the original work contained herein have not been undertaken or done by unspecified sources or persons.

MOHAMMAD FARHAN MOHD ZAIN

ABSTRACT

Hyperbaric is defined as an act of operating at pressure higher than atmospheric pressure. The hyperbaric chamber is an improvised version of pressure vessel where a pressure vessel is designed in a way that the vessel can be excess via door, in order for a components to be inserted and tested at high pressure. The high pressure acted in the vessel can be maneuvered or controlled by a fluid that is pumped to the vessel. The main purpose of designing this hyperbaric chamber is to support the UTP Deep-water Research for testing a deep-water Oil and Gas instrument at high pressure. Hence, the aim of this project is to design and analyze the Hyperbaric Chamber according to the pressure vessel design guideline drawn by ASME VIII standards with the defined parameter taken by previous researcher in deep-water oil and gas field.

There are four major parts designed in this hyperbaric chamber which are the vessel main body, vessel heads, nozzle instrumentation attached to the vessel and the vessel supports. The designed simulation is carried out by two mechanical engineering software which is CATIA V5R and ANSYS 14. CATIA V5R is used to drawn the virtual designed of hyperbaric chamber and then it is transferred to ANSYS for further analysis. Based on ANSYS simulation results, there are 7 critical stress points defined in the designed hyperbaric chamber. Compared with the theoretical stress calculated to the results analyzed by ANSYS, proved that the stress acted in the hyperbaric chamber wall does not failed due to over pressure at operating condition. Hence, the design is acceptable as per required design specification.

ACKNOWLEDGEMENT

In the Name of Allah, The Most Gracious and Merciful, praise to Allah, He is without doubt the Almighty. Eternal blessing and peace upon the Glory of the Universe, our beloved Prophet Muhammad (S.A.W) his family and companions. First and foremost, thanks to the Almighty for giving me the strength to complete the Final Year Project for Mechanical Engineering course without casualties. A deep gratitude goes to my supportive family that given me the motivation on completing this project throughout two semesters.

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NOMENCLATURE

3-D	Three Dimension
ANSI	American National Standard Institute
ASTM	American Society for Testing and Materials
ASME	American Society of Mechanical Engineering
UTP	Universiti Teknologi PETRONAS
HBOT	Hyperbaric Oxygen Therapy
YS	Yield Strength
UTS	Ultimate Tensile Strength
P	Force Applied
A_o	Initial Area
σ	Engineering Stress
e	Engineering Stress
l_o	Initial length
E	Modulus of Elasticity
FMEA	Failure Mode Analysis
R	Radius
K_{1C}	Fracture Toughness
a	Initial Crack length
Y	Yield Strength
ISO	International Organization for Standard

CHAPTER 1

INTRODUCTION

1.1 Background of study

The main topic for this project is to design and analyze a hyperbaric chamber for deep-water high pressure application. The hyperbaric chamber is an application of high pressure vessel where the main function is to test components at a high pressure conditions. This chamber has been developed back in the 18th centuries where at first it was established for a health treatment. Nowadays, the chambers have served many purposes in many industries apart from medical; it was also used in underwater divers as in Figure 1.1 [1], and in oil and gas industries. This project is focused on designing and analyzing the hyperbaric chamber based on the defined parameter.

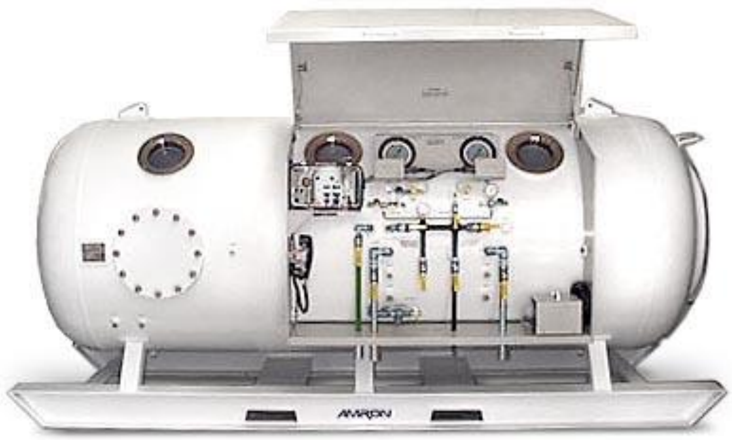


Figure 1.1: Hyperbaric Chamber

As for a fact, a drilling underwater currently has reached to a level where the equipment could not withstand the pressure acted underwater. Hence over the years, a simulation and test were conducted for the equipment in order to simulate the event of a real situation underwater. Generally, the pressure increases as the depth of the sea-level underwater increases. In other words, the deeper the equipment operates underwater, the higher the pressure acted towards it. Due to this event, it has affected the integrity or the deep-water equipment life span faster than expected. Other than that, deep-water equipment also experiences a slightly low temperature condition. However, this temperature does not reach to an extreme level where the water would freeze. By knowing the required temperature and pressure acted, the hyperbaric chamber is designed.

The main objective of this project is divided into two parts which are designing the hyperbaric test chamber according to standard, analyzed the designed stress and deformation profile of the design. The design requires literature review to determine the specification details of the suitable test component used in deep-water application and then identify the possible critical point of failure in the design.

One of the most important aspects in designing the high pressure chamber is identifying the material type for the chamber itself. The selected material physical properties are required for the correlation with the design pressures during simulation process. The physical properties will limit the pressure of the chamber's capability of holding. Furthermore, the material will determine the strength of the design based on its yield strength and its mechanical properties.

Other than material, the design of hyperbaric chamber depends greatly on the geometry size and the design shape. These design factors play an important role in stress analysis. The geometry of the design will determined the theory involves in project design. For instant, circumferential and longitudinal stresses are the two major theory of cylindrical thin-walled pressure vessel. Based on the researched, the most suitable design shape for hyperbaric chamber is cylindrical shaped because of its advantages such as user friendly and easy to be designed. Then, the diameter or size of the vessel also determined the value of stress acted on the chambers wall. A higher diameter of cylindrical vessel will have a lower stress acted on its body.

As for the stress analysis, the authors will relate the theory of failure available to the design simulation. Based on the theory of failure, the critical point is where the potential failure of the design might occur. When pressure is introduced into the chamber, it pressurizes the wall of the chamber evenly and hence, bringing weak point diligently to failed.

1.2 Problem Statement

Development of system component / part / or machine for deep-water application need to be properly tested before they are actually installed in the field. This is due to high pressure and low temperature operating condition. The testing is normally done in the hyperbaric test chamber in the laboratory. The test chamber is normally developed by the testing service provider and therefore data related to the hyperbaric testing chamber characteristics is not easily accessible. The UTP deep-water Mission Oriented Research group in its roadmap is looking at developing hyperbaric testing facilities in campus to enhance deep-water technology research. Thus, this project is carried out to produce a hyperbaric test chamber to support UTP deep-water research.

1.3 Objective of the study

1- Design a hyperbaric test chamber suitable for underwater component testing at high pressure deep-water environmental condition.
2- Analyze the design capability by performing a stress analysis at the chamber critical point such as the vessel opening, support and etc.

1.4 Scope of Study

The main purpose of this project is to execute the objective stated. The process starts with the designing of hyperbaric chamber where all the criteria involved to design a normal pressure vessel are investigated. The designs have to be relevant with the factor approach and theory involves in order to understand the behavior of the chamber failure during analysis. The methodology section showed the design approach of the hyperbaric chamber in detailed especially on the flow of the process involving design and stress analysis.

The stress analysis of the hyperbaric chamber is based on the simulation of the chamber using well-developed engineering software excluding the fabrication works. The design is simulated and analyzed through software simulation solution such as ANSYS Static Structural and CATIA V5. ANSYS software shows the results of the pressure flow through system during actual operation of the design chamber. The values were recorded, analyzed and compared with the allowable stress and yield stress of the structure. Meanwhile, the CATIA software expresses the whole design as well as its parts through 3-Dimensional images and dimensions.

1.5 Relevancy of project

In relevance to the project basis is to apply the knowledge of designing a pressure vessel based on the mechanical theory of selecting the appropriate parameter involving safety measures and design variation. This report also contains the design blueprint that helps the reader to understand the methods to design the most suitable hyperbaric chamber according to design specification required based on ASME standard. Oil and Gas Industries or any other industrial that uses high pressure equipment would have been benefit on this project by obtain the relevant knowledge to design their own hyperbaric chamber. Besides designing, they could also be able to understand the theoretical failure of stress membrane within the pressure vessel and thus, integrate the relationship of this theoretical failure to the stress behavior of the pressure vessel. By knowing the mention knowledge, they could apply this knowledge to improvise the necessary design available in the market for better quality applications.

CHAPTER 2

LITERATURE REVIEW

2.1. Historical Background of Hyperbaric Chamber

Hyperbaric means "relating to, producing, operating, or occurring at pressures higher than normal atmospheric pressure" [2]. Hyperbaric literally means a pressure condition acting is higher that the atmospheric pressure. Meanwhile hyperbaric chamber is chamber that capable of handling a substantial amount of pressure required for certain proposed application which is course higher than the atmospheric pressure. Previously, the hyperbaric chamber is extensively used in medical field for patient oxygen treatment.

Henshaw was the first person who developed the concept of hyperbaric. He first built a pressurized chamber in 1662 and used for health treatment [3]. The concept of his chamber is pressurizing a room where a patient will absorb more oxygen higher than at normal person at atmospheric level. In 1928, Harvard Medical School built a hyperbaric chamber for health researched and developments of ill patient. Not long after that, around 1960, people started to take interest and discovered HBOT (Hyperbaric Oxygen Therapy) for treatment [4]. The HBOT is made mainly for a patient or even a healthy person that requires an oxygen therapy. The chamber is pressurized above atmospheric while pure oxygen is pumped inside the chamber to enhance the absorption of oxygen through patient's body.

In 1788, the first pressurized room was created; high pressure chamber was developed and put to a high scale development in diving bell for underwater industrial repair of an English bridge underwater. Diving bell is structured and rigid chambers used to transport a diver underwater. This chamber applies the concept of hyperbaric because when it used to transport a diver, who will experience a high pressure environment [3] There are many other development of a hyperbaric concept over the history, likewise the first ever deep sea diving suit was invented by August Siebe in 1819, which used a compressed air and supplied directly to "the helmet" which mobilize the diver underwater [3]. The compressed air supplied act as an oxygen that the diver/individual breathe underwater for certain purposes. Nowadays, a diver used a hyperbaric chamber (high pressure chamber) to familiarize their body to sustain pressure higher than the atmospheric before diving underwater. Over the years, the systems and theory evolve, enhance as well as improvise by many companies worldwide to solve many problem related to testing of equipment operates and exposes high pressure environment.

2.2. Challenge and Design Specification Involves

The hyperbaric chamber primarily designed to mitigate an oil and gas industries component failure. The oil and gas underwater component in platform such as BOP (Blowout Preventer) is exposes to a very high pressure underwater and the only ways to test these components capability withstanding such pressure is through hyperbaric chamber. The environment now is very challenging where the depth of the drilling operation has now increased over *10000ft* or over 3000m below sea level. Due to this event, the pressure acted has reached to a certain level where it exceeded the pressure rating of much equipment design. The pressure acted is approximately over *4500psi* with the evacuation of internal pressure inside the riser pipe or in any such component that are exposed to the external pressure enough to reach the equipment's design specification. Hence, increase the tendency of the equipment to fail and damage abruptly [4].

Based on the research, the hyperbaric chamber specifications deducted from the reliability control of Deep-water testing [4] stated as follows:

- *Operating Fluid: Fresh water*
- *Operating Temperature: 50° to 100° F (10° to 38° C).*
- *Hydrostatic pressure test capability: 0 - 2,050bar (0 - 30,000psi)*
- *Hyperbaric test chamber dimensions: 280mm Ø x 1,620mm (11in x 63.8in)*
- *Hyperbaric pressure test capability: 0 - 340bar (0 - 5,000psi)*
- *Material: Stainless steel 316L*

Since, the Hyperbaric Chamber will be operating at room condition; the pressure gauge installed will not indicate the actual pressure exerted by the system. Most pressure gauges are designed only to measure and indicate the pressure of a fluid and neglecting the atmospheric pressure. Normally, pressure gauge will calibrate to zero reading once in room condition. However the absolute pressure can be calculated by the following equation;

$$Absolute\ Pressure = Gauge\ Pressure + atmospheric\ Pressure \qquad \text{Eq. 2.1}$$

The standard value of the atmospheric pressure is 101.325 kPa.

2.3. Material and Structural Considerations

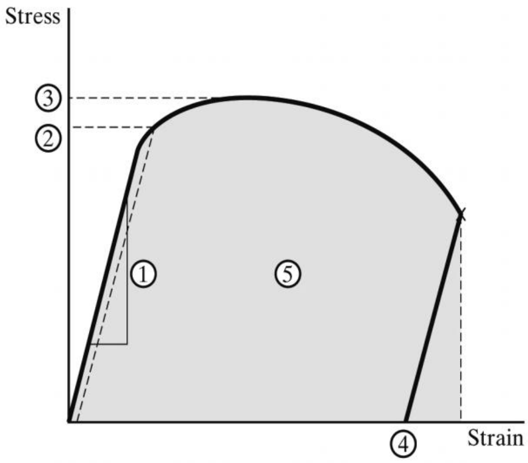


Figure 2.1: Tensile test of ductile material

The prior criteria for material selection depend on the mechanical properties of the material itself. The significance of these mechanical properties can be clearly understood under tensile test as shown in the Figure 2.1. The mechanical properties shown in Figure 2.1 are *Modulus of Elasticity (1), yield strength (2), the ultimate tensile strength (3)* of the material, *Ductility (4), Reduction in area under stress strain diagram (5), fracture toughness* and *resistance to crack*. However the fracture toughness and its resistance to crack are neglected. There are varieties of materials suitable for the design of hyperbaric chamber, but compared with the required design criteria such as physical property, whichever able to withstand the required amount of pressure is selected.

Below are the basic equations for nominal stress and strain for most material. It indicates the amount of pressure that a material can exerted for theoretical data calculation of the design later on.

$$\text{Engineerin g Stress}, \sigma = \frac{P}{A_o} \qquad \text{Eq. 2.2}$$

$$\text{Modulus of elasticity, } E = \frac{\sigma}{e} \qquad \text{Eq. 2.3}$$

$$\text{Engineering Strain, } e = \frac{(l - l_o)}{l_o} \qquad \text{Eq. 2.4}$$

While, P is the pressure exerted, A is the cross sectional area of the specimen for the design, l is the instantaneous length of the specimen [5].

The failure in designing the hyperbaric chamber mainly depends on the stresses that act upon the chamber's wall. As for this reason, material stress analysis plays an important role in designing the hyperbaric chamber. Materials with the easiest obtainable in the market and cost effective are selected for the design. For instant, a carbon fiber based material is highly durable and ductile material compared to other potential material, the cost for obtaining such material is high. Hence, the famous pressure vessel fabricators in the world, Mersen France Pagny-sur Moselle have listed several material suitable for the high pressure design that can be considered as good based material for a very high pressure conditions [6] such as;

a) Graphite
b) Titanium
c) High Yield Carbon Steels
d) Carbon Fiber
e) Aluminum Alloys
f) Other potential alloys and reinforce fiber

The mechanical and physical properties of the metal are extremely significant in terms of selecting the design. However, only several materials have good mechanical characteristic and easily obtained in the markets such as carbon steel. A carbon steel material is preferable because of its well-known application in piping whether in low or high pressure and it is much cheaper compare to other materials. There are many types of carbon steels in the market but what make the differences between them are the compositions of the materials. For instant, a normal ASTM 105 carbon steel used in most fabrication and a high yield API 5L carbon steel, both are in the same carbon steel group but their hydrostatic pressure capability are highly different. Thus, for designing this high pressure chamber, high yield carbon steel is preferred.

2.4. Pressure Vessel Design

A normal pressure vessel consists of 4 main sections; the main body which includes the main cylindrical vessel to hold the volume of the fluid or gas, the nozzle attachments for an instrument installed in the vessel with the flange attached to each nozzle, the pressure vessel supports, and the end cap or the heads to enclose the pressure vessel. Since hyperbaric chamber is an improvement application of a pressure vessel, an opening door is required for excess in setting up the specimens. Before proceeding with the design, the following sections are thoroughly researched in accordance to the ASME codes of pressure vessels.

2.4.1. Main Body Shell

Pressure vessel shells are welded together to form a structure that has common rotational axis. Based on the research done by Donald M. Fryer and John F. Harvey by the book of High Pressure Vessel, a shape of a vessel usually comes in a form of spheres, cylinders, ellipsoids and many more with a basic curve of a circle [7]. The most common shape for a pressure built over the centuries are cylindrical and spherical shape. However, for accessing via doors, the suitable structure is cylindrical shape. This type of vessel has the lowest stress point compared to others.

In designing the shape of the chamber, it is essential to know the stress exerted against the wall of the chamber. The stress can be calculated using the formula guided by ASME. The purpose of knowing the stress is to make sure that the design does not fail due to over-stress at certain points when the value is higher than the allowable material stress. Below are the theory and the internal stress calculation of Thin-walled pressure vessel theory for Thin Walled Cylinder [8].

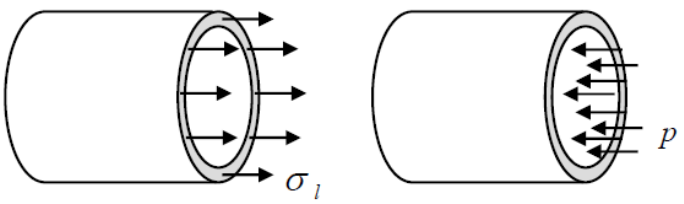


Figure 2.2: Circumferential stress in a thin-walled cylindrical vessel

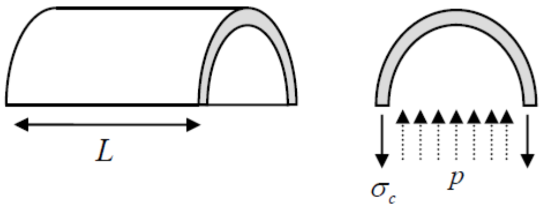


Figure 2.3: Longitudinal stress in a thin-walled cylindrical vessel

The longitudinal stress calculations for cylindrical vessel;

$$\boxed{\sigma_l = \frac{pr}{2t}} \qquad \text{Eq. 2.5}$$

The circumferential stress calculation for cylindrical vessel;

$$\boxed{\sigma_c = \frac{pr}{t}} \qquad \text{Eq. 2.6}$$

Where,

P is the internal pressure, r is the radius and t is the thickness

According to ASME code of pressure vessel, under the code UG-27b [9], the Equation 2.6 is derived in order to find the suitable thickness;

$$t = \frac{PR}{SE - 0.6P} \qquad \text{Eq. 2.7}$$

Whereas;

t = minimum thickness

p = Allowable pressure

S = Allowable stress

E = joint efficiency, in percent

R = inside Radius

The equations is applied if the thickness or less than half of the radius and the internal pressure is less than **0.385SE**

Since the main body shell is cylindrical, the easiest way to fabricates it is using an ASTM standard pipe line. Pipe line has its own schedule and label based on the materials and thickness. Since, the design basis of the cylindrical shells is according to the piping

ASME standard, thus the dimension follows the ASME B36.10 Carbon Steel standard. The standard dimension of pipe range is between 1/2mm to 600mm. This dimension can also be applied to other types of standardized piping materials.

2.4.2. Heads and closures

Heads are typically curved rather than flat. Usually, curve configurations are stronger in shape and thus allow the structure to be thinner in size, lighter, and much less expensive than the flat heads [9]. Heads are categorized by the shapes; such as ellipsoidal, hemispherical, tori spherical, conical, flat head, and tori conical. Since the main shell used the ASME cylinder pipe standard also can be applied to the heads or a pipe cap. A pipe cap usually takes on the shape of ellipsoidal head and mostly it is welded together to the main cylindrical shells. The configurations of ellipsoidal head are shown in figure 2.4 below [10].

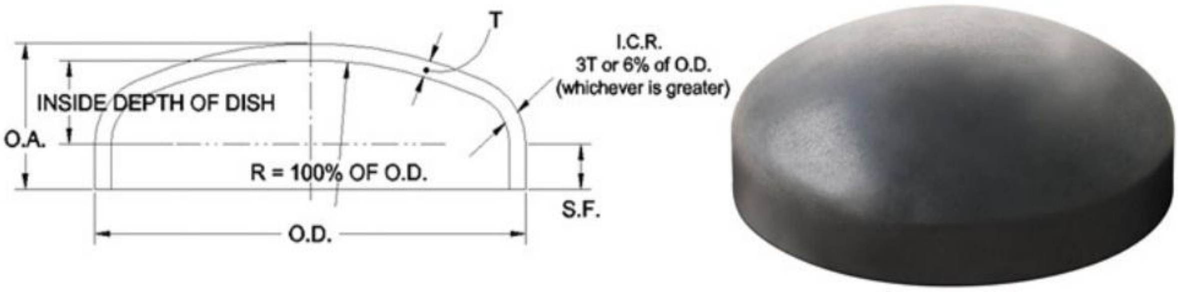


Figure 2.4: Design of pipe cap in accordance to ANSI B16.9 or ASME B16.9

To ensure that the internal pressure acted upon the heads is compatible to the design, the ASME has provided the equation (code reference UG-32) based on the ellipsoidal Heads under Internal pressure [11].

$$t = \frac{PR}{2SE - 0.2P} \quad \text{Eq. 2.8}$$

However this equations is only applicable for D/2h = 2,

If, the ratios are between 1.0 to 3.0;

$$t = \frac{PRK}{2SE - 0.2P} \quad \text{Eq. 2.9}$$

Where,

$$K = \frac{1}{6}\left[2 + \left(\frac{D}{2h}\right)^2\right] \quad \text{Eq. 2.10}$$

The minimum thickness is compared with the cylindrical thickness shell calculated earlier and noted whichever higher is taken [11].

Nevertheless, a hyperbaric chamber must have an opening door for inserting a specimen or component to be tested. Thus a flat end with a blind flange bolted and welded to the main shell is applied. Since the structure of the cylindrical shell is constructed based on the ASME pipe, thus the bolted flange follows the same procedure. Flange can be categorized by their pressure class and material types. However there are also special types of flanges which covers a higher in pressure rating, using higher yield carbon steel material such as is API6A (6B) flanges. A flat end flanges head design can be divided into few parts which are;

a) Weld neck flange
b) Bolt and Stud
c) Gaskets
d) Blind Flange

2.4.3. Nozzle opening and compensation

A nozzle is a spout at the cylindrical vessel body installed to provide an opening for instrumentation and vessel drains via opening. Instrumentation installed such as pressure gauge and thermometer; are the two common basic compartment designs together with a pressure vessel. The main purpose of this compartment is to monitor and calibrate the amount of medium inserted through the vessel input up to its required level. Other than instrumentation, the nozzles also function as a main source for input and output medium such as fluids into the vessel. Installing this nozzle requires an opening of the main vessel.

There are guidelines or procedure shown by ASME codes for nozzle installation. ASME codes stated that the area removed for the opening must be replace by an equivalent area in order for the opening to be adequately reinforced [12]. However, an opening could weaken the structure of the shells structure. To overcome this problem, a reinforcement pad is installed at the area of shell removed. ASME Code has simplified the rules and equations for the dimension of nozzle attached to ensure that during operating the shell membrane stresses of the shell are kept within acceptable limits even though the opening is made as Figure 2.5 below [12].

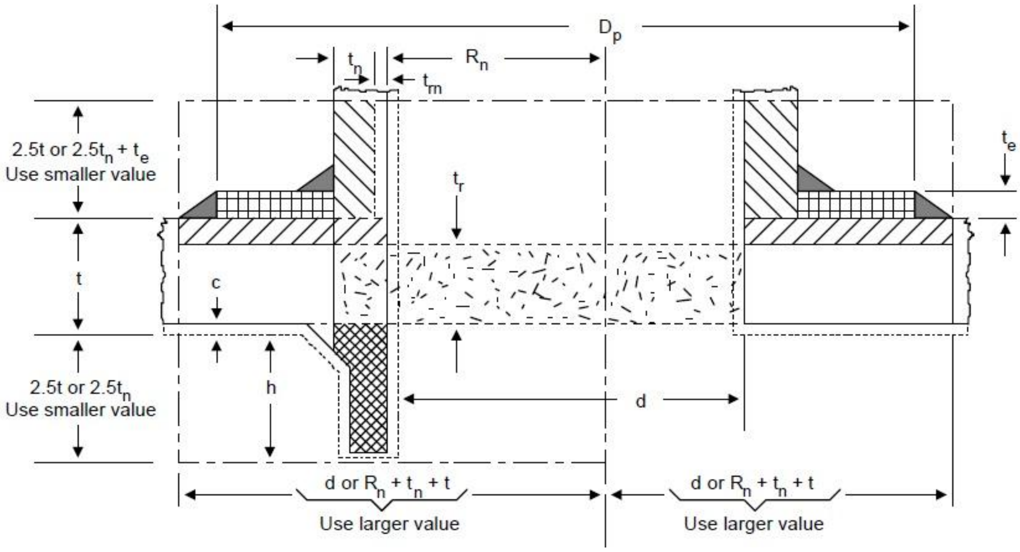


Figure 2.5: Cross-Sectional View of Nozzle Opening

Where,

D_p = Diameter of the reinforce pad

D = diameter of the openings

tr = thickness of the shells

trn = thickness of nozzle shell

te = thickness of reinforce pad

Usually nozzle attachments installed are welded together with a flange at the other ends. This is to ease the instrumentation installation such as pressure gauge welded to the flanges.

2.4.4. <u>Pressure Vessel Supports</u>

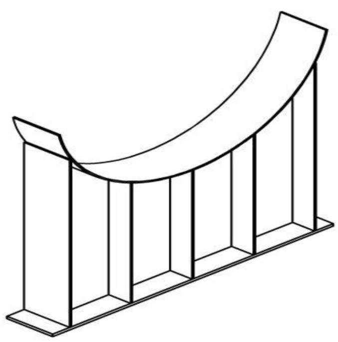


Figure 2.6: Typical scheme of Saddle Support

This type of pressure vessel support depends primarily on its size and the orientation of the pressure vessel. There are few types of support for pressure vessels such as skirts, leg, and saddle for horizontal pressure vessel. The most suitable support to install on a small scale horizontal pressure vessel are saddle type supports as shown in Figure 2.6 [11]. A saddle support spreads the weight load over a large area of the shell in order to prevent an excessive local stress at shell within the support points. The width of the saddle is determined by the specific size of the design conditions of the pressure vessel.

2.5. Factor of Safety and Theory of Failure

Based on the table in ASME VIII-1, the allowable stress governing equation values for welded pipe for ferrous or non-ferrous is [11];

$$\mathbf{2/3 \times 0.85\, S_y,} \qquad \text{Eq. 2.11}$$

S_y indicates the yield strength of the material. This is to ensure the safety of the design from failure due to plastic deformation during operation. Because of the diversity of pressure chamber design variation, the allowable stresses are not based on the single material property alone, but a combination of properties such as tensile strength, the yield strength, elongation, and many more. The hyperbaric chamber concepts are literally almost equal to the concept of a pressure vessel, thus the ASME, American Society of Mechanical Engineers Boiler and Pressure Vessel Code, Section VIII Division 1: indicated the rules for Construction of Pressure Vessel, the allowable design stress are will be based on the following [13];

a) 25 percent reduce of the specified minimum tensile strength at room temperature
b) 62.5 percent reduce of the specified minimum yield strength at room temperature
c) 80 percent reduce of the minimum stress required to produced material rupture, at the end of 100 0000 hours at design temperature.

In additions, the welded pipe dimension also considers the corrosion allowance as an additional thickness in the event of corrosion, erosion, abrasion or scaling lost on the shells. As for the design pressure, a safety measure of 10% to the operating pressure is applied. Besides safety measures, the theory behind the failure have to be known in order for understanding the behavior of membrane stress.

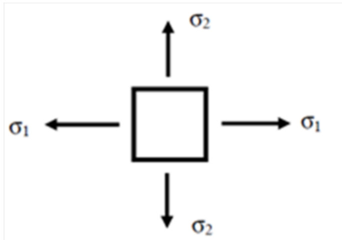


Figure 2.7: Principal Stress

Based on the Pressure Vessel, Design and Practice Book by Somnath Chattopadhyay [13], in element stress analysis, there are three theories of failure which is; Maximum Principal Stress Theory as shown in Figure 2.7, Maximum Shear Stress Theory, and Maximum Distortion Energy Theory. In this hyperbaric stress analysis, the test design simulation is run by ANSYS Software. The principal behind the stress analysis of the software is the Von Mises theories of Failure. Thus, it is essential on knowing the theory of Maximum Distortion Energy (Von Misses) Criterion. The von Mises Criterion (1913), is often used to estimate the yield strength of the ductile material.

Hypothetically, the Von-Mises theory of failure shown when the distortion energy accumulated in the component under stress of elastic limit or in other word the energy reaches the same as the yield stress as determined by uniaxial tension [13]. The simple uniaxial tensions are determined by the membrane stress to principal stress, σ_1, σ_2 as shown Figured 2.8. Mathematically, this von Mises theory is expressed and shown in figure below;

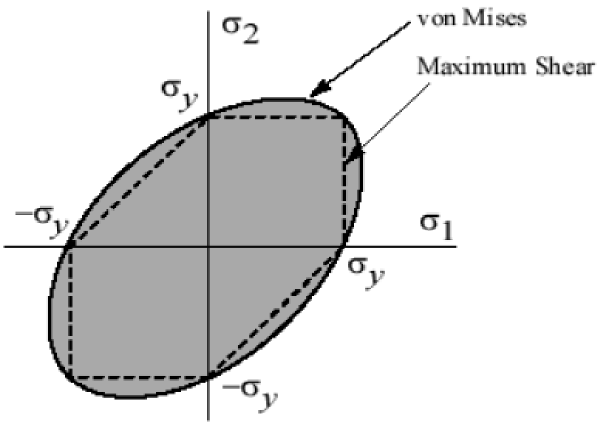


Figure 2.8: Von Mises Stress theory of Failure

$$\sqrt{\frac{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2}{2}} \leq \sigma_y \qquad \text{Eq. 2.12}$$

With the assumption of plane stress,
$\sigma_3 = 0$, derives;

$$\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 \leq \sigma_y^2 \qquad \text{Eq. 2.13}$$

The Equation 2.13 is plotted in the Figure 2.8 above. According to von Mises Criterion, the stress point falls outside of the eclipse will cause yielding or failed.

2.6. Reliability Stress Analysis of Hyperbaric Chamber

Based on the research by Dr Tae Bo Jeon, Industrial Engineering Kangwon National University, the reliability of Failure Modes on Hyperbaric Chamber of Oxygen Treatment is studied. In his study, he performed a Failure Mode and Effect Analysis, FMEA to find out the potential failure in the system at certain points. Based on his researched, the failure analysis consisted of three parts which are the chamber itself, the door and the openings [14]. Most of the failures are due to the shock and poor manufacturing of the design fabrication during high pressure simulation. Throughout the list of the failure recorded, critical points were located and guided for hyperbaric design chamber analysis later on.

The door openings of flanges are usually failed due to excessive amount of pressure applied to the flange end. Besides that, the failure may also come from the wrong selection of pressure rating of the flange. By applying a lower pressure rating to the design will cause the flange to exert more pressure that the allowable, hence it started to burst. Moreover, there is also failure due to the bolted losses its grip from the flanges. The theory below indicates the stress calculation for bolted flange acting upon a cylindrical tank as shown in Figure 2.9 below [15].

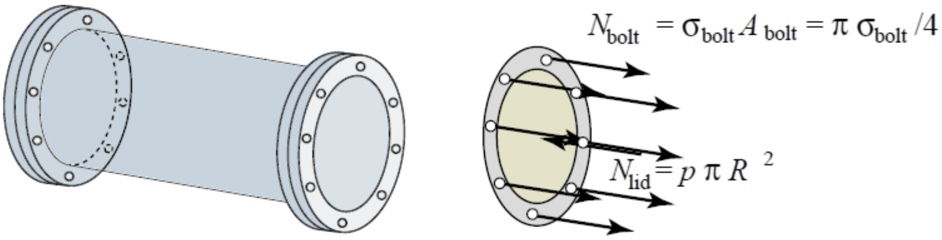


Figure 2.9: Bolted failure analysis

Where;

N_{bolt} is the force acted of each of the bolted flange.

N_{lid} is the force acted on the lid in negatives x-direction

n is the number of lid

Derived from the circumferential stress acted in a cylindrical vessel equation 2.5; the bolted stress value can be known by the following equation;

$$n \; x \; N_{bolt} = N_{lid} \qquad \text{Eq. 2.14}$$
$$\sigma_{bolt} = 4pR^2/n$$

The stress acted in each of the bolts determines whether the design can withstand certain amount of pressure in actual test.

Besides the head failure, the weakest and common critical point in a high pressure chamber is at the welded joint. For example a study of Fast Fracture Failure of Ammonia Tank as has shown in figure 2.10 which forms a crack through heat affected zone or the welded parts of the tanks [16]. The shape of the pressure vessel as shown in the figure was cylindrical shell with hemispherical cap at the edge and was welded. This case study shows that a crack about 2.5mm deep form in the heat-affected zone in between shell and circumferential weld will cause the tank to burst.

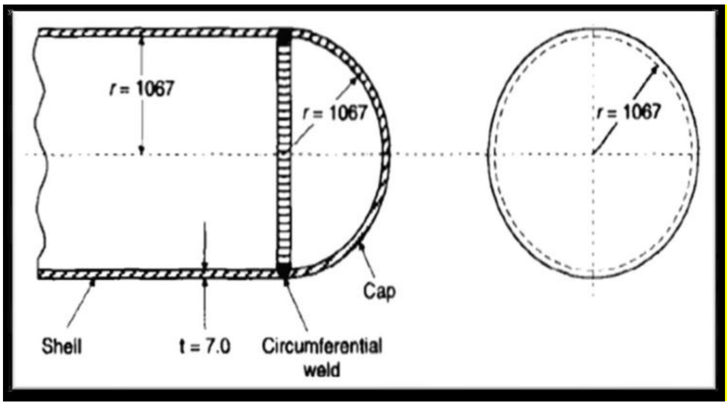


Figure 2.10: Ammonia Tank Parameter

The purpose of the study is to determine the behavior of the vessel or tank with the crack form in the welded area of the Heat-Affected zone whether it would be fast fracture or just leaking in the area. By knowing the behavior, the potential problem can be mitigated with a better design in the future and prevent from any inadequate failure in the design. The value of fracture toughness parameter in the system is determined by;

$$K_{1C} = Y\sigma\sqrt{\pi a} \qquad \text{Eq. 2.15}$$

The equation above measures the fracture toughness of a design. Y is the constant – depending on the geometry of the cracked body while 'a' is the allowable crack depth. The fracture toughness is measured by experimenting and producing small fatigue crack in a standardized test specimen [16]. Then, the specimen undergoes a constant cyclic stressing while the rate of crack growth is measured. The faster the cracks form, the lower the value of the fracture toughness.

The fracture toughness defines the ductility of the material due to crack. The lower value of its fracture toughness the higher probability for the system to deform and failed. To know the relation between the stress deform at the crack section and the allowable stress, is by comparing the longitudinal stress of the material exerted by the cylinder calculated using equation 2.6 and the critical stress for fast fracture at equation 2.15. However, the applications of fracture toughness in the design is neglected due to shortage of time, nevertheless, it is put under recommendation of design verification before fabrication.

CHAPTER 3

METHODOLOGY

3.1 Research Methodology

A numerous background studies and understanding have been done to complete this project. Gantt chart can be referred in the Appendix H for further understanding. Below are the steps taken and project execution in order to achieve the objectives stated;

3.1.1 Problem Identification

Most project studies were based on finding the solution for the problem statement. In order to proceed with the project understanding, it is essential to understand thoroughly the problem statement by meeting with certain people and do a slight background researched before proceed with the literature review. On this project, the problems arise with the short facilities of UTP to simulate a hyperbaric test. Thus, an objective is formed in order to cater the problem.

3.1.2 Background Research

Background research or literature review is one of the most important parts of project based studies. It is one of the major phases where the past works is reviewed and all the potential solution from the problem statement is analyzed. An amount of parameters for hyperbaric chamber is noted and contributed to the project designed. In designing the hyperbaric chamber, the design specification based on researched done by previous researcher is detailed. These parameters are the type of working fluid, hyperbaric and hydrostatic pressure. Other than that, information for design criteria is significantly analyses and understands. Hence, the theoretical analysis is reviewed.

3.1.3 Designing the Concept

The concept is design with a proper instrumentation and parameter included as accordance to the design ASME standard. A various design concept is analyses and compared to achieve better results. After the understanding of the standard guide by ASME Pressure Vessel codes, the designed is modeled using CATIA or 3-Dimensional image software. This parts involved calculations and decision making for the suitable material and selections of part according to the required design specification. There are 4 main parts designed involved in this section, which are designing the main cylindrical vessel, the end heads, supports and nozzle installation. Then, the design concept is modeled on 3-Dimensional software, CATIA.

3.1.4 Analyzing the design

Before analyzing the design, the parameter or the boundary conditions are defined for each part Hyperbaric Chamber in the ANSYS software. ANSYS is simulation software used to simulate a solution of an operating pressure to the designed pressure chamber via structural analysis section. After defining the boundary, there are 5 different results analyzed in the ANSYS such as; maximum principal stress and strain, Von-Mises equivalent stress and strain and last but not least the total linear deformation.

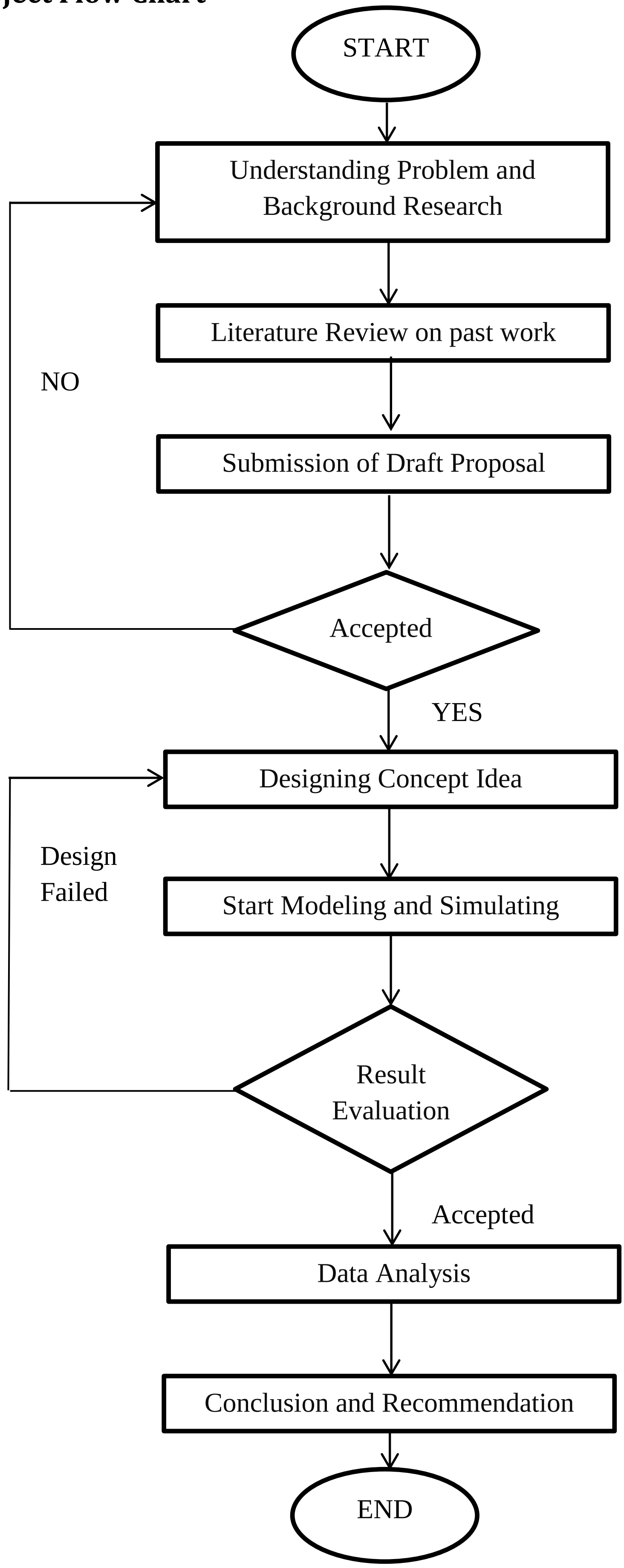
3.1.5 Post-processing of results

The simulation result is compared with the theoretical value calculated. If the design could withstand the required pressure or stress, the maximum design potential is examined using the ANSYS 14. Then, the hyperbaric chamber is simulated to find out the critical point or the highest stress acted if parts are separated. Other than that, the welding point and all possible critical point are noted as well. With all the data obtained, a table and graph is plotted. A discussion is made in comparison between the results obtained to the theoretical value calculated.

3.1.6 Conclusion and Recommendation

This project is concluded and recommendations for further improvement in the future design were made. The relevant data, results and discussion were recorded and documented in the final reports.

3.2 Project Flow Chart



3.2.1 Tools and Software Required

A) CATIA Version 5R,

This software is used to design 3 dimensional image structures according to the dimension and parameter defined. In this project, CATIA is used to drawn the 3 dimensional hyperbaric chamber parts before transferring it to ANSYS Software to simulate the design condition.

B) ANSYS Version 14

ANSYS is an engineering simulation software that offers a simulation solution sets providing access to virtually any field of engineering simulations that requires design process. In this project, it is used to simulate the pressure acted inside the chamber to verify and analyzed the designed capability.

3.3 Designing the Hyperbaric Chamber

The Figure 3.1 below (not to scale) is the 3 dimensional design of the hyperbaric chambers based on pressure vessel guided by ASME under section VIII pressure vessel design. The of pressure vessel design are divided into four (4) main parts, which are (as labeled);

Table 3.1: Legend for the Design Pressure Vessel

No.	Parts of the Design			Quantity
1	Cylindrical Vessel			1
2	The Heads	2.1	Ellipsoidal Head	1
		2.2	Flange Head – Weldneck Flange	1
		2.3	Flange Head – Blind Flange	1
		2.4	Flange Head – Bolts and Studs	16
		2.5	Flange Head – Ring Type Joint	1
3	Nozzle Attachments	3.1	Compensation Pad	7
		3.2	Two inch Weldneck Flange	3
		3.3	One inch Lap Joint Flange	1
		3.4	Three quarter inch Lap Joint Flange	1
		3.5	Two inch elbow	2
4	Pressure Vessel Support			1

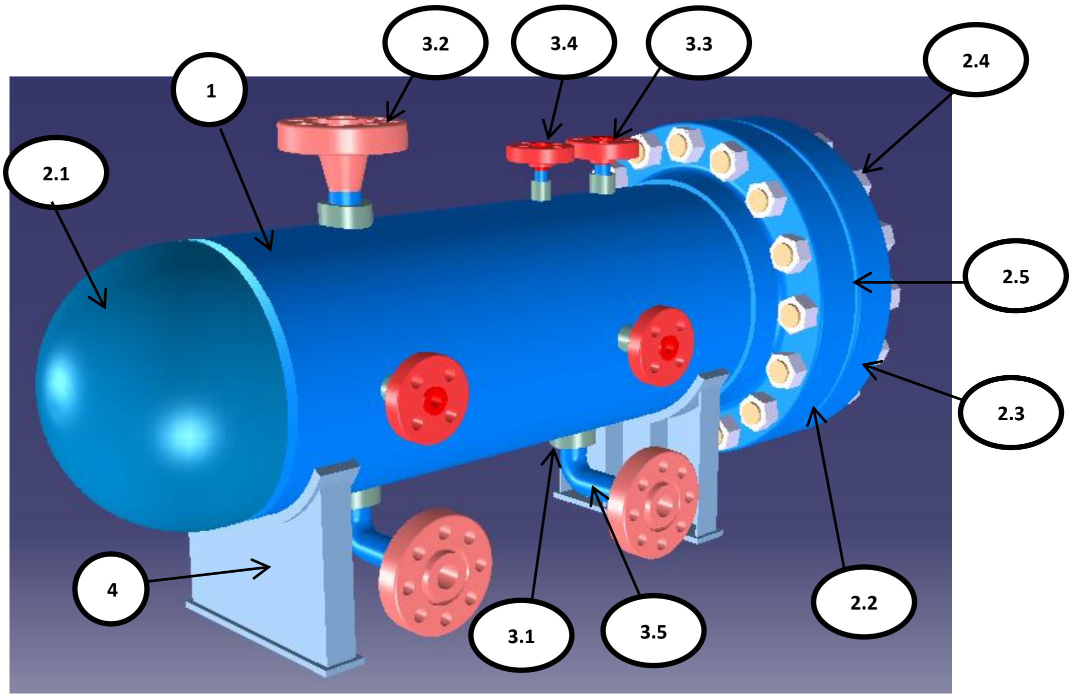


Figure 3.1: Design Pressure Vessel

3.4 Numerical Analysis

3.4.1 Cylindrical Vessel

The main vessel for pressure vessel usually came in two shapes which are cylindrical and spherical. Both shapes have the most efficient in balancing pressure throughout the shell. Nevertheless, cylindrical is chosen to ease of application suits for its own purpose which to input an instrument or specimen inside for pressure and stress testing. As stated before, the cylindrical vessel will be selected from pipe ASTM standard in accordance to its yield strength. Below are the criteria that have been considered to choose the appropriate design;

1- Material Selection,

Since the pressure to be tested is significantly about 5000psi (~34Mpa) hyperbaric pressure. Thus, the author has select a **API 5LX line Pipe, Grade X70, Seamless, ERW, Butt-weld** [17] with significantly high yield strength of 485Mpa and tensile strength of 570Mpa, taken from Chemical Composition & Mechanical Properties by Jindal Group Ltd, Piping Fabricator Company in India.

2- Calculation for suitable operating shell thickness,

Take the dimension given by ASME B36.10 for pipes design [18];

20" NPS or 500mm Nominal Diameter, Internal Diameter, ID = 431.8mm for t = 38.10mm with Schedule Number 120.

- Calculating allowable stress, S and design pressure, P_{design} using equation 2.11

$$S = \frac{2}{3} x 0.85 x \sigma'_y$$

$$S = \frac{2}{3} x\ 0.85\ x\ 485Mpa$$

$$S = 278.33Mpa$$

E = 0.95 for API 5L grade pipe

$$P_{design} = P_{operating}\ x\ 110\%$$

$$P_{design} = 34Mpa\ x\ 110\% = 37.4Mpa$$

- Cylindrical Shell under Internal Pressure

$$P_{design} < 0.385\ X\ SE$$

$$= 0.385\ X\ 278.33\ x\ 0.95 = 101.8Mpa$$

Since P_{design} < 0.385SE, the following equation is applicable,

Assume corrosion Allowance 2mm at normal conditions in accordance to ASME.

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

$$\frac{37.3Mpa\ X\ \left(\frac{431.8mm}{2} + 2mm\right)}{278.33Mpa\ X\ 0.95 - 0.6\ X\ 37.4Mpa}$$

$$t = 34.15mm$$

- Spherical Shells under Internal Pressure

$$t = \frac{PR}{2SE - 0.2P}$$

$$t = \frac{37.4Mpa\ X\ \left(\frac{431.8}{2} + 2mm\right)}{2\ X\ 278.33Mpa\ X\ 0.95 - 0.2\ X\ 37.4Mpa}$$

$$t = 15.63mm$$

Since t = 34.15mm < 38.10mm, thus the thickness is acceptable. The Dimension of the cylindrical vessel is shown in the appendix A.1

3.4.2 Pressure Vessel Heads

The heads plays an important role in pressure vessel. For this hyperbaric chamber, the author has chosen ellipsoidal head and blind by flange at both ends.

1- In order to ease fabrication, Pipe cap is used as mention in literature review and the dimension for one end, Ellipsoidal head is taken based on end Cap [18] – ASME B16.9. The material base is similar to main body vessel, which is Seamless and Butt-weld. Calculate whether the thickness is suitable, using the equation 2.7 from ASME for Spherical shell under internal pressure.

$$t = \frac{PR}{2SE - 0.2P}$$

The above equation may applied, if $\frac{\boldsymbol{D}}{\boldsymbol{2h}} = \mathbf{2,}$ or the ration of major axis (D) to twice the minor axis, (2h) is 2:1

$$\text{Since,}\quad \frac{500}{2\,x\,229} = 1.11 < 2,$$

Thus, noted under the second criteria according to ASME code for Pressure Vessel if D/2h in in between 1.0 to 3.0, the thickness under internal stress is multiplied by K using equation 2.8 and 2.7

$$t = \frac{PRK}{2SE - 0.2P}$$

Where,

$$K = \frac{1}{6}\,X\left[2 + \left(\frac{D}{2h}\right)^2\right]$$

$$K = \frac{1}{6}\,X\,[2 + (1.11)^2]$$

$$K = 0.539$$

$$t = 15.63mm\ X\ 0.539 = 8.42mm$$

Thus, t = 15.63mm < t = 31.8mm acceptable. The Ellipsoidal head dimension is shown in the Appendix A.2

2- The other cylindrical end of the pressure vessel would be a weld-neck flange attached to the main pressure vessel. The use of this weld-neck is to have open-closed doors which enable the user to input a specimen at the end. The Weld-neck is then attached to a blind flange connected with bolts and stud as shown in the appendix B.2.

- Since the pressure applied is 5000psi, the most appropriate selection for Weld-neck flange that attached to the main cylindrical vessel is Flange type API-6A-6B [19] with a pressure rating of 5000psi and may connected with BX ring gasket type [20] as shown in Appendix B.3 (as specified by AT Engineering Solution Ltd, UK company majoring in flanges, pipe and fittings fabrication and supplies. These types of flanges holds relatively high yield of stress for Onshore or Offshore piping systems.

- Select **Weld-Neck Flange and Blind Flange, with pressure rating 5000psi, Nominal Size and Bore of Flange, 16 ¾ Inch or 425.45mm**. The drawn dimension of the Blind Flange and Main Weld-neck flange is shown in the Appendix A.3 and Appendix A.4.

- Bolts and Stud

  The bolts and Stud are chosen based on **ASME B1818.2.2 for Heavy Hex Nuts** used with a stud bolts in accordance to ASME B16.5 [21]. The Nominal Size of the bolts is 1 7/8 inch with a spanner at 75mm types for turning. The bolts joint together the main weld-neck Flange and the Blind flange, locked by the stud at both ends as shown in Appendix B.2 and the detailed dimension is as shown in the Appendix A.5.

- Ring Type Joint Gasket

  The Rings type follows the required standards with the flange specify by AT engineering Ltd. Nevertheless, the dimension is taken from Novus Sealing Ltd [20], a manufacturer and distributors of sealing and jointing materials as shown in the appendix B.1. The design selection is RTJ, or **Ring type joint BX162 with net weight 2.26kg and 16 ¾ inch, 5000psi pressure class, API 6A type flanges.**

3.4.3 Nozzle Compensation

The following design is based on ASME calculation [12] for nozzle installation in the pressure vessel design. The design criteria are as follows;

Table 3.2: Nozzle Installation Types

Nozzle attachment for Water and Drains for Both Sides	**Instrument attachments**
2" NPS X 3 (Nominal Diameter ;50mm)	1" NPS X 3 (Nominal Diameter ;25mm) ¾" NPS X 1 (Nominal Diameter ;20mm)

1- Design Information;
   - Design Pressure = 37.4Mpa
   - Nozzle and Shell Material API Pipe 5LX – 70X (L485)
   - Corrosion Allowance for Shell = 2mm
   - Corrosion Allowance for Nozzle = 1mm
   - Nozzle does not pass through Vessel Weld Seam
2- Calculation for 2inch Pipe, Nominal Diameter = 50mm
   - OD = 60.3mm, Schedule Number = 180
   - t = 5.54mm
   - ID = 49.2mm

<u>Reinforce Area,</u>

$$A = dt_rF \quad \text{Eq. 4.1}$$

Where,

d = Finished diameter of circular opening

t_r = Minimum required thickness of the shell using appropriate ASME Code formula and a weld joint efficiency of 0.95 (API 5LX)

F = Correction factor, = 1

Diameter, d = ID - 2(Corrosion Allowance)

d = 49.2mm – 2 x 1mm = 51.2mm

Reinforce Area, A

$A = (51.2 \times 34.15) = 1748.5mm^2$

Available reinforcement area in the shell, A_1 whichever larger between A_{11} or A_{12}

$$A_{11} = (E_1t-Ft_r)d$$
$$A_{12} = 2\,(E_1t-Ft_r)(t+t_n) \qquad \text{Eq. 4.1, 4.2}$$

Where;

$E_1 = 1.0$

F = 1 for all cases except integrally reinforce nozzles that are inserted into a shell

t_n = Nominal thickness of the nozzle in the corroded conditions

thus,

$A_{11} = (38.10mm – 2mm - 34.15mm)\,(51.2mm) = 99.84mm^2$

$A_{12} = 2\,(38.10mm -2mm - 34.15mm)\,(38.10 – 2mm +5.54mm) = 162.4mm^2$

Thus $A_1 = 162.4mm^2$ is chosen

Calculate thickness of the nozzle, t_{rn}

$$t_{rn} = \frac{37.4Mpa\left(\frac{51.2mm}{2}+1mm\right)}{(274.83Mpa)(0.95)-0.6\,(37.4Mpa)} = 4.17mm$$

Reinforcement area that is available in the nozzle wall, A_2. Whichever smaller between A_{21} or A_{22}

$$A_{21} = (t_n – t_{rn})\,5t$$
$$A_{22} = 2(t_n-t_{rn})\,(2.5\,t_n + t_e) \qquad \text{Eq. 4.3, 4.4}$$

Where,

t_{rn} = Required thickness of the nozzle wall

r = radius of the nozzle

t_e = thickness of the reinforce pad, assume there is no reinforce pad = 0

Thus,

$A_{21} = (5.54mm – 1mm – 4.17mm) \times 5(38.10mm – 2mm) = 66.79mm^2$

$A_{22} = 2\,(5.54mm – 1mm – 4.17mm) \times [2.5 \times (5.54mm – 1mm) + 0] = 8.4mm^2$

Thus $A_2 = 8.4mm^2$ is chosen

Total available reinforcement area, A_T

$A_T = A_1 + A_2 = 162.4mm^2 + 8.4mm^2 = 170.8mm^2$

Since $A_T < A_1$, the nozzle is not adequately reinforce, thus reinforcement pad is required.

Calculate Area of requirement pad area A_5, and the pad diameter, D_p

$$A_5 = A - A_T \qquad \text{Eq. 4.5}$$

Thus,

$A_5 = [D_p - (d+2t_n0] \, t_e$

Choose t_e = 40mm

$1748.5mm^2 - 170.8mm^2 = [D_p - (51.2mm + 2 (5.54mm - 1)] \times 40mm$

$1577.7mm^2 / 40mm = D_p - 60.28mm^2$

$D_p = 99.7mm$

To confirm that the diameter does not extend the outer limit permitted reinforcement zone in the shell, 2d

$2d = 2 \times 53.2mm = 106.4mm$

Thus, D_p = 100mm is accepted.

Applied with the same steps and calculations for 1" pipe and 3/4 "pipe, the results are tabulated (table 4.3) and the dimension drawn shown in the Appendix A.6 as follows; the completed nozzle compensation 3D drawing attached with the main vessel is shown in the Appendix B.5.

Table 3.3: Nozzle Compensation Details for 1 and 3/4 inch

Nozzle attachment pipe size, (mm)	**1" NPS, 25mm Nominal**	**¾' NPS, 20mm Nominal**
Outside Diameter, OD, (mm)	33.4	26.7
Schedule Number, SH	80	80
Inside Diameter, ID, (mm)	24.3	18.9
Diameter of opening, d, (mm)	26.3	20.9
Thickness of Nozzle pipe, t_{rm} ,(mm)	4.55	3.91
Thickness of reinforce pad, t_e , (mm)	40	40
Pad Diameter, D_p, (mm)	56	42

3- Minor flange welded after the nozzle compensation installation at the main vessel shell.

- For **2 inch nozzle**, the author have chosen weld-neck flange. Weld-Neck Flange size 50mm, according to ASME B16.5, Material, ASTM A105 (Carbon Steel), Class 2500, maximum allowable pressure are 42.5Mpa at 38Degree Celsius as stated ASME for Temperature/pressure rating table for carbon steel ASME B16.5 (BS 1560) [18] the dimension of the 2inch weld-neck flange welded together with the elbow are shown in the Appendix A.7

- At the drain section on below the pressure vessel, there are elbow which connected to the weld-neck flange for a better adjustment and lower maintenance. The Elbows are accordance to ASME, equivalence to the main vessel, with a short Radius elbow types, 2inch in nominal diameter, Butt-weld and Seamless pipe fittings.

- For 1 inch (25mm Nominal Diameter) and ¾ inch (20mm Nominal Diameter) flange. The criterion is similar to the 2in weld-neck flange but the only difference is the type of flanges uses, the lap joint Flange which connected to the stub end as shown in the appendix 4.1. The difference between these two flanges is the stub ends may easily modified and maintain for a smaller size instruments.

- The 2inch weld-neck flange will be connected to a valve which will control the water flows in and out for operating procedure. While the smaller flanges are connected to a fix pressure or temperature instrument for calibrations.

- The lap Joint Stub ends are design according to ANSI B16.9 with the net weight 1kg per pieces and the dimension of the stud and the lap joint flange are shown in the appendix A.8 and A.9;

3.4.4 Supports

1- As mention in the previous, a saddle support is design with the guide by ASME for pressure vessel support design. The dimension and information are subtracted from Pressure Vessel Design Manual by Dennis Moss, Third Edition, Gulf Professional Published in 2003. The design is taken on table 3.26, typical Saddle Diameter with an outside diameter of 24 inch and a weight of 80lb (36.2874kg) per sets. However the maximum operating weight of 15400lb (6985kg) [22].

2- Typical dimension is alters using interpolation for 20 inch vessel.
Calculation for total weight of pressure vessel,

- Main Vessel

  Cylindrical Shell weight, 572.21kg + Nozzle compensation 2inch, pipe 200mm, (3,4 kg x 3) + elbow, (0.6kg x 2) + 1 inch compensation, pipe 100mm (1.7kg x 3) + 3/4inch (1.07kg) = 590kg

  Flange weight, 2inch (16.2kg x 3) + 1inch with stub [(4.77kg + 0.3kg) x 3) + ¾ inch with stub (3.4kg + 0.18kg) = 67.4kg

  Water weight, total volume of cylinder (π(ID)h) x density of water, 1000kg/m^3

  Π x 0.432m x 1.2m x 1000kg/m^3 = 1628kg

Total weight 590kg + 67.7kg + 1628kg = 2285.7kg

- Heads

  End Ellipsoidal Cap , weight 113.4kg

  Water weight, volume (π x ID^2 x h/6) x water density, 1000kg/m^3,

  Π x 0.432^2 x 1.2m/6 x 1000kg/m^3 = 117.3kg

Total weight = 113.4kg + 117.3kg = 230.7kg

  Flat end, (weight taken from API Weight Reference Chart 1,Woodco,USA, pg.5)

  Weld-neck flange weight, 810.2 lbs ~ 367.5kg

  Blind Flange weight, 1085.8 lbs ~ 492.5kg

  Weight of 1 set of all thread stud bolts with 2 nuts each, 245 lbs ~ 111.1 x 16sets = 1777.6kg

  Ring Gasket weight, type BX162, 4.96 lbs ~ 2.25kg

  Water Weight, 0.12m height x π x 0.432m x 1000kg/m^3 = 51.84 kg

Total Weight = 367.5kg + 492.5kg + 1777.6kg + 2.25kg + 51.8kg = 2691.7kg

Total all weight, Heads + Main vessel,

2285.7 kg + 2691.7 kg = 4977.4kg

3- Thus, since the operating weight is 4977.4kg < 6985kg, lower than the maximum operating weight by design, thus the design is suitable and implemented. The detailed dimension of the supports is shown in the Appendix A.10.

3.5 Applying Boundary Condition

Table below shows the summarize type or material used/selected to design the hyperbaric chamber. The types are defined and label as ASME standard as shown in table below;

Table 3.4: Hyperbaric Chamber Part Design Types

Parts of the Design		**ASME Size**	**ASME Type**
Cylindrical Vessel		20” NPS (500mm), 1.2 m,	API 5LX line Pipe, Grade X70, Seamless, ERW, Butt-weld
The Heads	Ellipsoidal Head	20” NPS (500mm)	API 5LX Pipe Cap, Grade X70, Seamless, ERW, Butt-weld
	Flange Head – Weldneck Flange	16 ¾ “ (425.45mm)	API 6A-6BX, Weld-neck Flange, 5000psi
	Flange Head – Blind Flange	16 ¾ “ (425.45mm)	API 6A-6BX, Blind Flange, 5000psi
	Flange Head – Bolts and Studs	1 7/8” NPS,	Heavy Hex Nuts
	Flange Head – Ring Type Joint	16 ¾ “ NPS,	Ring Type BX162, 5000psi, API 6A type flanges
Nozzle Attachments	Two inch Weldneck Flange	2” NPS	PN 420, Class 2500, ASTM 105, Butt-weld, Seamless
	One inch Lap Joint Flange	1” NPS (25mm),	PN 420, Class 2500, ASTM 105, Butt-weld, Seamless
	Three quarter inch Lap Joint Flange	¾” NPS (20mm)	Class 2500, PN 420, ASTM 105, Butt-weld, Seamless
	Two inch elbow	2” NPS (50mm)	PN 420, 90D, SR, Class 2500, ASTM 105, Butt-weld, Seamless
Pressure Vessel Support		OD 24in, 122D, 7” x 4”	Saddle Type,

*ERW: Electric Resistance Welding NPS: Nominal Pipe Standard, PN : Nominal Pressure, OD : Outside Diameter

Before analyzing the Hyperbaric Chamber parts using ANSYS 14 software, the material properties and other parameters are defined as follows;

Table 3.5: Defined Boundary Conditions in ANSYS

Parameters	**Material Properties**	**Value Defined**
Main Cylindrical Vessel and Ellipsoidal Head	Density, Kg/m^3	8000
	Poisson's Ratio	0.3
	Tensile Yield Strength, Mpa	485
	Tensile Ultimate Steel Stress, Mpa	570
Main Flanges	Density, Kg/m^3	8000
	Poisson's Ratio	0.3
	Tensile Yield Strength, Mpa	414
	Tensile Ultimate Steel Stress, Mpa	586
Small Flanges and 2inch Elbow	Density, Kg/m^3	7850
	Poisson's Ratio	0.3
	Tensile Yield Strength, Mpa	250
	Tensile Ultimate Steel Stress, Mpa	485
Meshing (Fine)		Default
Hyperbaric Pressure, Mpa		34
Environment Temperature, ^{o}C		22
Calculated Allowable Stress, Mpa		279

The defined parameter will determined the stress acted towards the wall of the Chamber. The results of the simulation are analyzed in the next chapter.

CHAPTER 4

RESULT AND DISCUSSION

4.1 Simulation of Hyperbaric Chamber separated by parts

The objective of the simulation is to analyze whether it is functional according to the operating conditions by design and understand the potential maximum stress and its critical region. The detailed 3D modeling of the hyperbaric chamber was done using CATIA V5R17, with actual scale drawn as shown in the blueprint in appendix I. The CATIA 3 dimensional figure according to dimension calculated is shown in the Appendix B and it has been analyzed by ANSYS software. There are three sections for the simulation results of this hyperbaric chamber; first, the vessel parts is simulated based on the operating condition which is 34Mpa then followed by simulation of the vessel welded together at operating pressure and until its allowable stress limits, last but not least the diameter deformation of the vessels.

By applying the boundary condition, welding and contact points as shown in Appendix C, the results simulation is recorded. There are 5 results recorded during simulations which are, the total deformation or linear deformations, the maximum principal stress and strains, and the equivalent stress and strain. The results of the structural steel ANSYS stress usually came with aa figure and digram of stress region in different color as shown in Figure 4.4. Red color indicates the highest distortion and highest stress acted in the region and the maxumum value of this reqion is recorded. There are several parts in the hyperbaric chamber that exert a significantly high stress based on the Figure 4.4. Nevertherless, the author has simplified the list of stress acted and tabulated as shown in the table 4.1. The results of the simulations is devided into 4 parts as shown in the Appendix D, E , F and G which are;

a) The main vessel
b) The Heads include both ellipsoidal head and main flanges
c) The Chamber Support
d) At Nozzle Compensation

Table 4.1: Critical Region of Hyperbaric Chamber

Critical Point	At Design Hyperbaric Pressure, 34Mpa				
	Maximum Principal Strain, (mm/mm)	**Maximum Equivalent Elastic Strain (Mpa)**	**Total Deformat ion(mm)**	**Maximum Principal Stress, (Mpa)**	**Maximum Equivalent Elastic Stress (Mpa)**
Support Weld, (Refer Figure 4.4)	0.0008705	0.00089815	0.067615	225.14	265.09
At Nozzle Weld (Refer Figure 4.4)	0.0007210	0.00074846	0.0854519	248.25	358.74
Nozzle Compensation to Small Flange (Refer Appen. G.1)	0.001132	0.0011549	0.08429	271.70	230.56
Flange Head (Refer Appen. F.3)	0.0013578	0.0014654	0.14535	243.29	237.5
Ellipsoidal Head (Refer Appen. E)	0.0011462	0.0011891	0.2483	219.09	237.82
2inch Elbow at nozzle (Refer Appen. G.4)	0.001209	0.001196	0.2785	271.76	319.13
Nozzle Opening (Refer Appen. D.1)	0.0011503	0.0012096	0.14998	224.83	360.96

The table 4.3 above shows the maximum amount of deformation, stress and strain taken from the simulations results as shown in the appendix D, E, F and G and Figure 4.4. The equivalent stress and strains indicates the von-misses stress and strain acted in the Hyperbaric Chamber, while the Maximum Principal Stress and Strains indicates the maximum amount of allowable stress exert in the system. Comparatively, the highest equivalent elastic stress by all defines critical points would be at the nozzle opening inside the pressure vessel which is 361Mpa. This indicates the von-misses stress in the systems; nevertheless the amount of this stress still does not surpass the amount of yield stress of the material which is 485Mpa, thus it would not starts to yield. Thus, the design is acceptable. The highest distortion is at 2in elbow where the linear deformation is nearly reached 0.3mm which is still a lot less for the systems to fail. Besides that, the highest amount of principal stress is the 2inch elbow which is 271Mpa and its still does not surpass the maximum amount of allowable stress calculated, 279Mpa. Hence, the designed hyperbaric chamber is acceptable.

4.2 Simulation of Hyperbaric Chamber at its limit

Table 4.2: Simulation at Maximum Pressure before Fracture, 34Mpa

Maximum Principal Strain (mm/mm)	Maximum Equivalent Strain (mm/mm)	Maximum Principal Stress (Mpa)	Maximum Equivalent Stress (Mpa)
0.00012313	0.0001487	-36.645	29.791
0.0002726	0.0002994	0.54272	59.581
0.00042208	0.0004491	37.73	89.371
0.00057155	0.0005988	74.917	119.16
0.00072102	0.0007485	112.1	148.95
0.0008705	0.0008982	149.29	178.74
0.00102	0.0010478	186.48	208.53
0.0011694	0.0011975	223.67	238.32
0.0013189	0.0013472	260.85	268.11

One of the objectives of the analyzing the hyperbaric chamber is by knowing the pressure limits of the chamber vessel. However, before analyzing the hyperbaric chamber designed to its pressure limit, the chamber will be first simulated at operating condition as shown in Figure 4.1. The results of the simulations are tabulated in the table 4.3.

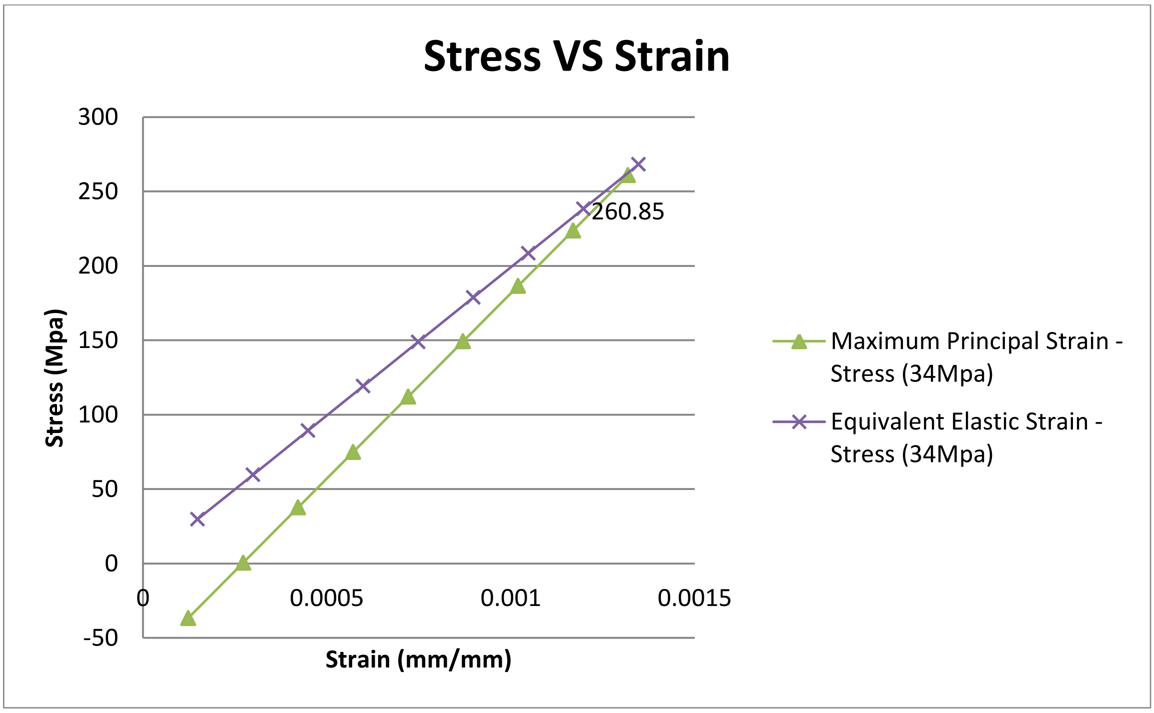


Figure 4.1: Stress vs. Strain Graph at Operating Pressure, 34Mpa

Referring to the table 4.3, the simulation results for the the maximum amount of principal stress acted by the designed wall is 261Mpa as shown in Figure 4.3. However, this value is still lower than the amount of theoritcal maximum allowable stress calculated, 279Mpa. On the other hand, the total deformation simulation of the system results is in linear axis deformation, thus, the dimater deformation is calculated. In order to obtain the diameter deformation value, the young's modulus or modulus of elasticity of the chamber has to be known. The material elasticity are calculated as follows;

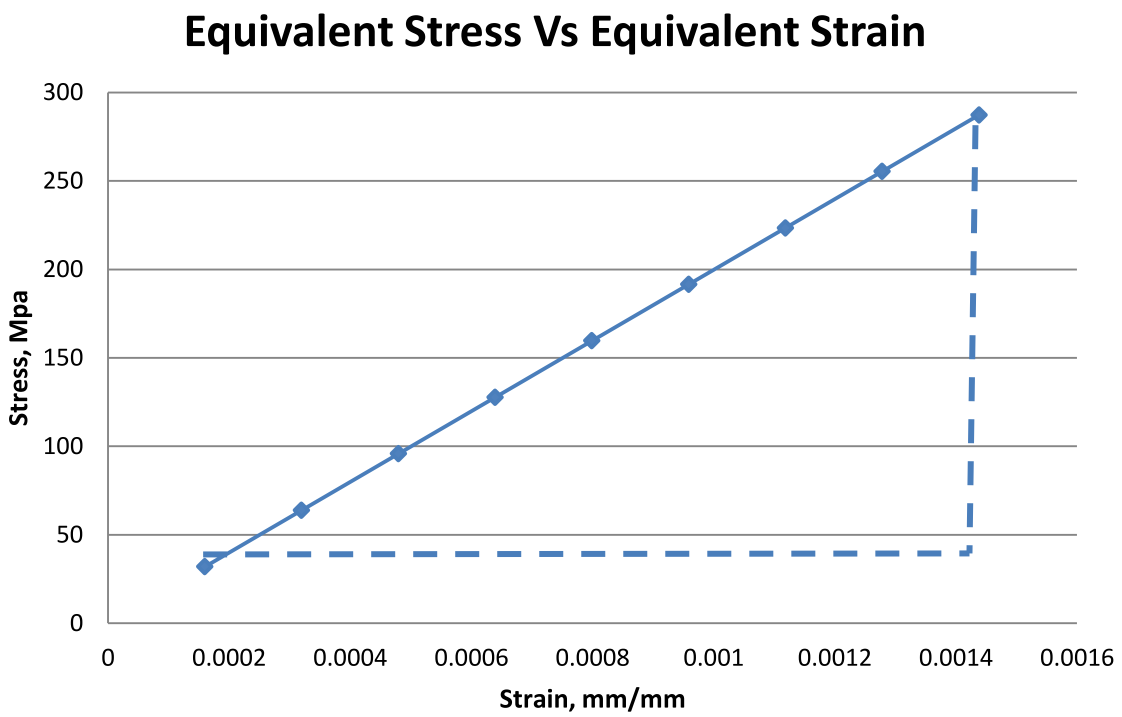


Figure 4.2 : Equivalent Stress Vs. Equivalent Strain Graph

To determine the young's Modulus, two points is selected based on the graph in Figure 4.2 and then calculated the gradient of the graph by the hooks law stress-strain relationship in the equation 2.4,

$$\frac{y_1 - y_2}{X_1 - X_2} = \frac{268.11 - 29.791}{0.0013472 - 0.0001487}$$

$$= 198847.73Mpa$$

Thus, the Young's' Modulus of the designed Hyperbaric Chamber is 2.0E5Mpa

Table 4.3: Simulation at Maximum Pressure before Fracture, 39.8Mpa

Maximum Principal Strain (mm/mm)	Maximum Equivalent Strain (mm/mm)	Maximum Principal Stress (Mpa)	Maximum Equivalent Stress (Mpa)
0.00013248	0.0001598	-39.252	31.919
0.0002934	0.00031961	0.059078	63.838
0.00045433	0.00047941	40.434	95.757
0.00061525	0.00063921	80.277	127.68
0.00077617	0.00079901	120.12	159.6
0.00093709	0.00095881	159.96	191.51
0.001098	0.0011186	199.81	223.43
0.0012589	0.0012784	239.65	255.35
0.0014199	0.0014382	279.49	287.27

Then, the limits of the design hyperbaric chamber is identified by tuning the amount of hyperbaric pressure to its limits. Since the theoritical allowable stress calculated earlier is 279Mpa, the designed chamber is test until the amount of Principal Stress higher than the amount of theoritical Maximum Allowable Stress.

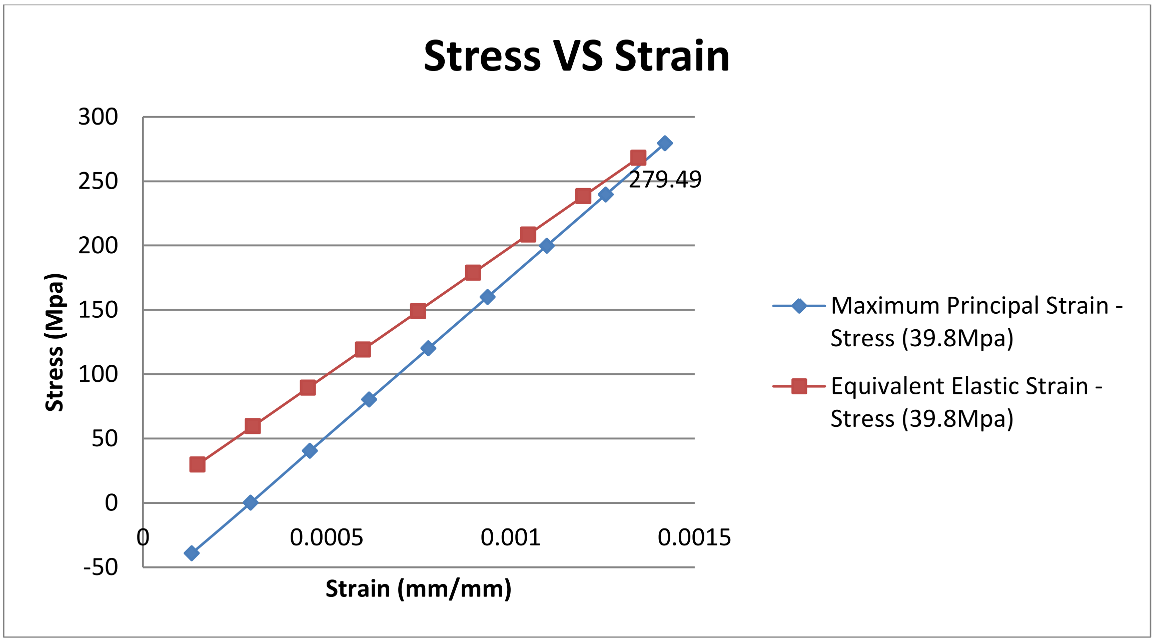
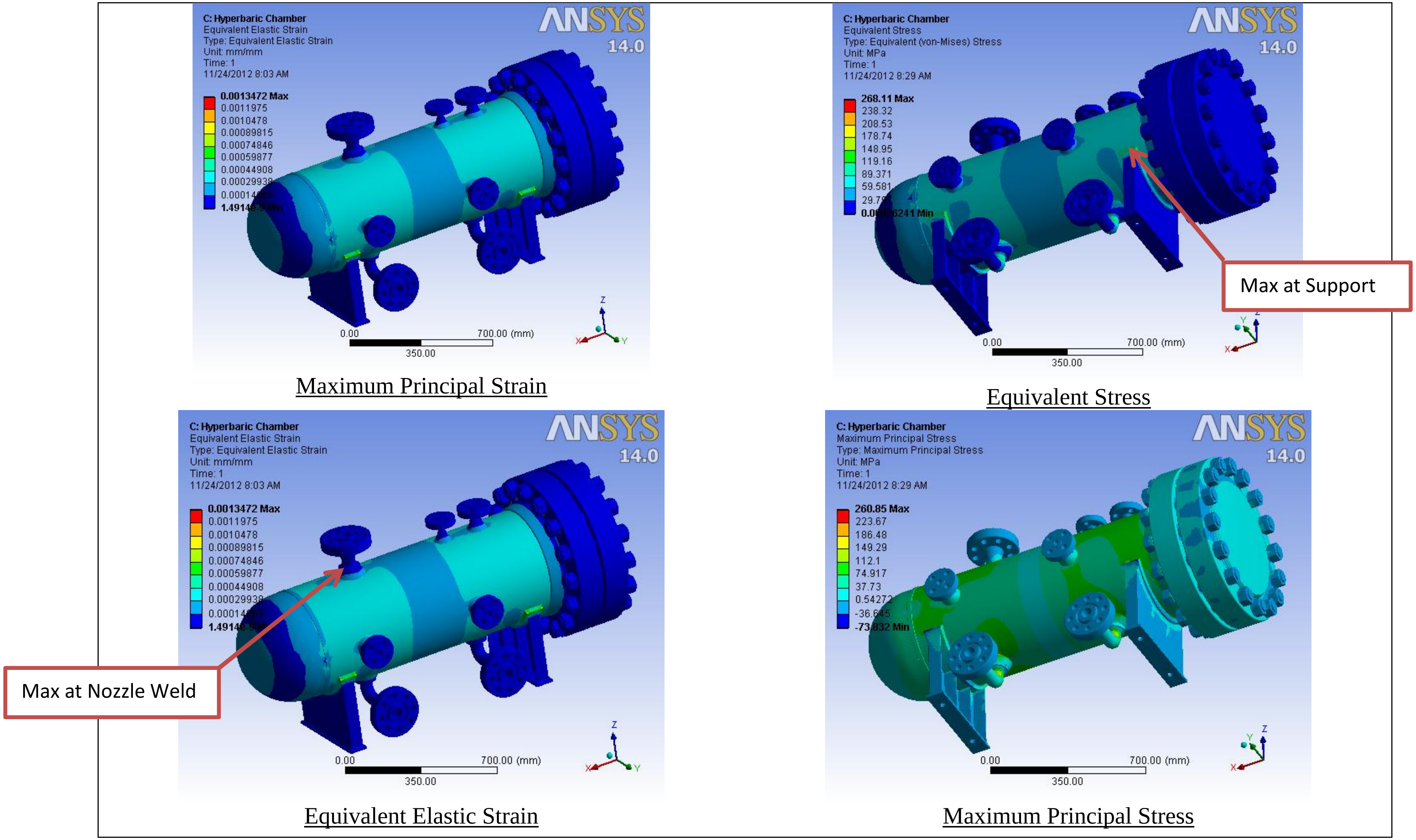


Figure 4.3: Stress vs. Strain Graph at Maximum Pressure

The simulation then be tabulated and graphed as shown in Figure 4.3 and Table 4.3. The pressure limit for the designed hyperbaric chamber is appoximately at 39.8Mpa where its maximum principal or allowable stress is approximately 279.5Mpa. Thus, in future design application, the chamber has to be ensure that the hyperbaric pressure test does not reached39.8Mpa.

Figure 4.4: The simulation of the Hyperbaric Chamber under ANSYS software



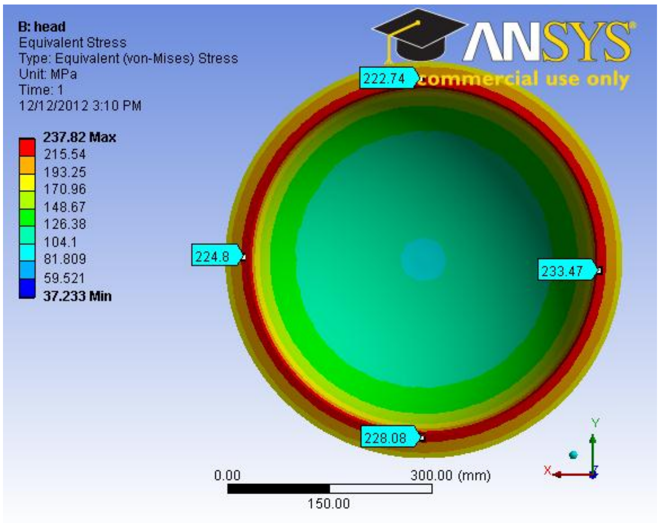
4.3 Diameter deformation of the hyperbaric chamber

The total deformation extracted from the simulation results is a linear deformation based on axis. Hence, in order to solve this problem a diameter deformation is calculated based on the hooks law as shown in the Equation 2.3, where in a stress-strain graph the stress acted in the system is directly proportional to the strain. The diameter deformation, δ can be calculated based on the strain equation,

$$\delta,\ D - D_o = e \times D_o \qquad \text{Eq.5.3}$$

Where, D_o is actual diameter, and D is the Chamber diameter after deformation

The Hyperbaric Chamber diameter deformation based on the equivalent stress taken at 5 different points which at the both ends of main cylindrical vessel and heads. Since the Young's Modulus calculated earlier would be 2e5Mpa, the diameter strain deformation can be calculated. After that, the value of the strain deformation is multiplied by the total deformation to know the exact value of the deformation at the points.



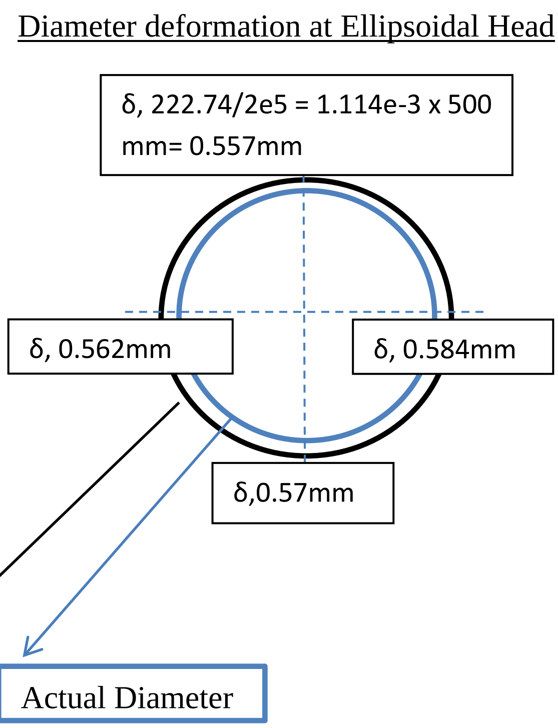
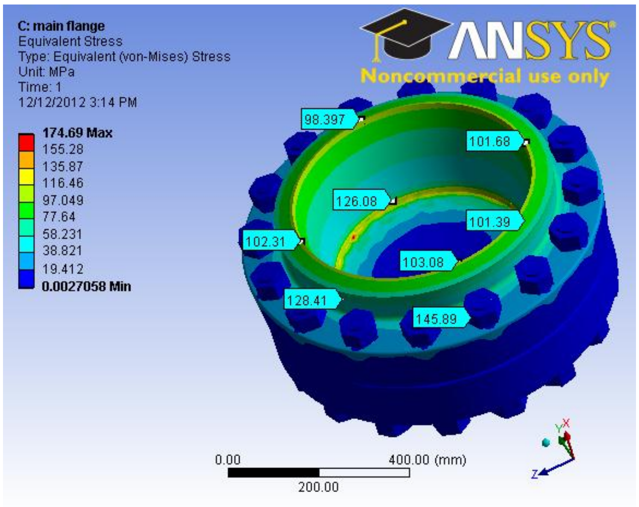


Figure 4.5: Ellipsoidal Head equivalent stress

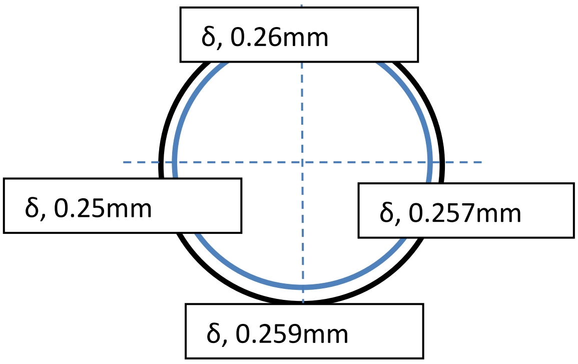
Figure 4.6: Flange Head equivalent stress



There are two define points

1- Diameter deformation at Weld-Neck Flange
2- Diameter deformation at Blind Flange

<u>Diameter Deformation at Weld-neck Flange</u>



<u>Diameter deformation at Blind Flange</u>

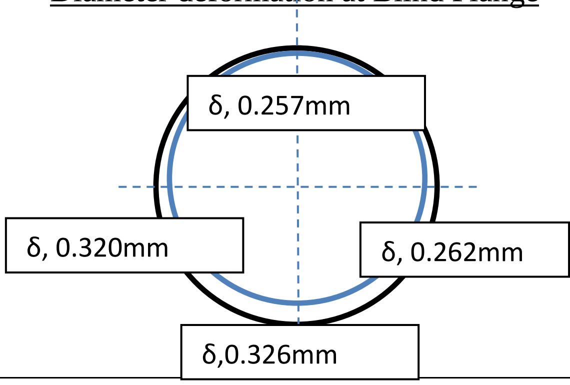
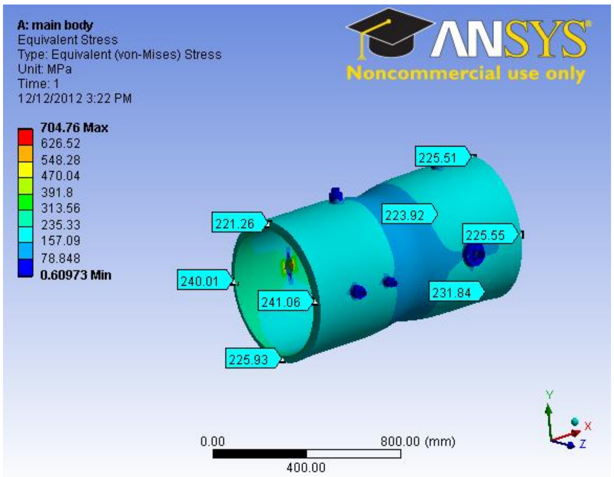


Figure 4.7 : Main Cylindrical vessel equivalent stress

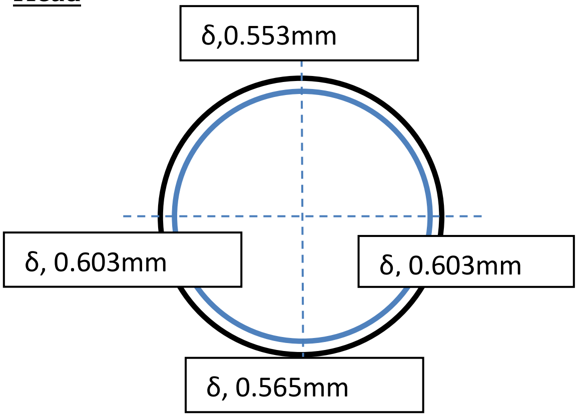


There are two define points

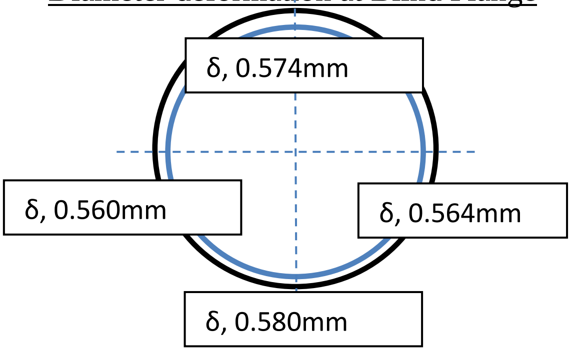
1- Weld joint between ellipsoidal head
2- Weld joint between flange head

Since the range value of diameter deformation at points is between 0.25mm to 0.603mm, thus it is not critical. By these results, it proves that the design is acceptable.

<u>Diameter Deformation between ellipsoidal Head</u>



<u>Diameter deformation at Blind Flange</u>



CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1. Conclusion

The project objective is to design a feasible hyperbaric chamber in accordance to the ASME pressure vessel guidelines and analyzed its design capability. The hyperbaric chamber was successfully design virtually as shown in the design blueprint in Appendix I. When analyzed, the hyperbaric test chamber does not failed under 0-5000psi or 0-34Mpa operating specified pressure. Besides that, 7 (seven) critical points or maximum stress point which have the highest possibility to fail due to its maximum stress acted are defined in the designed hyperbaric chambers. Lastly, the material allowable stress (279.5Mpa) at maximum allowable pressure limit exerted by the design is defined, 39.8Mpa as this point exceeded the theoretical allowable stress calculated. By this result, it is clearly shows that the hyperbaric chamber pressure design is acceptable as per required deep-water environment specified.

5.2. Recommendation

In order to achieve a better result, it is essential to conduct a proper experiment for the designed hyperbaric chamber. This is to compare the theoretical data calculated to the experimental data. Before future design fabrication, the integrity of the design stress defined are evaluate and analyze in order to cater the failure possibility due to over stress and high distortion at defined critical point.

The following are a few suggested recommendations on the hyperbaric chamber designed for further improvement;

a) A hinge in between the main vessel to the main flange is installed to ease of application during opening the main chamber doors.
b) To achieve the most feasible design without taking economic analysis into account, a lesser weight fiber based composite material is use.

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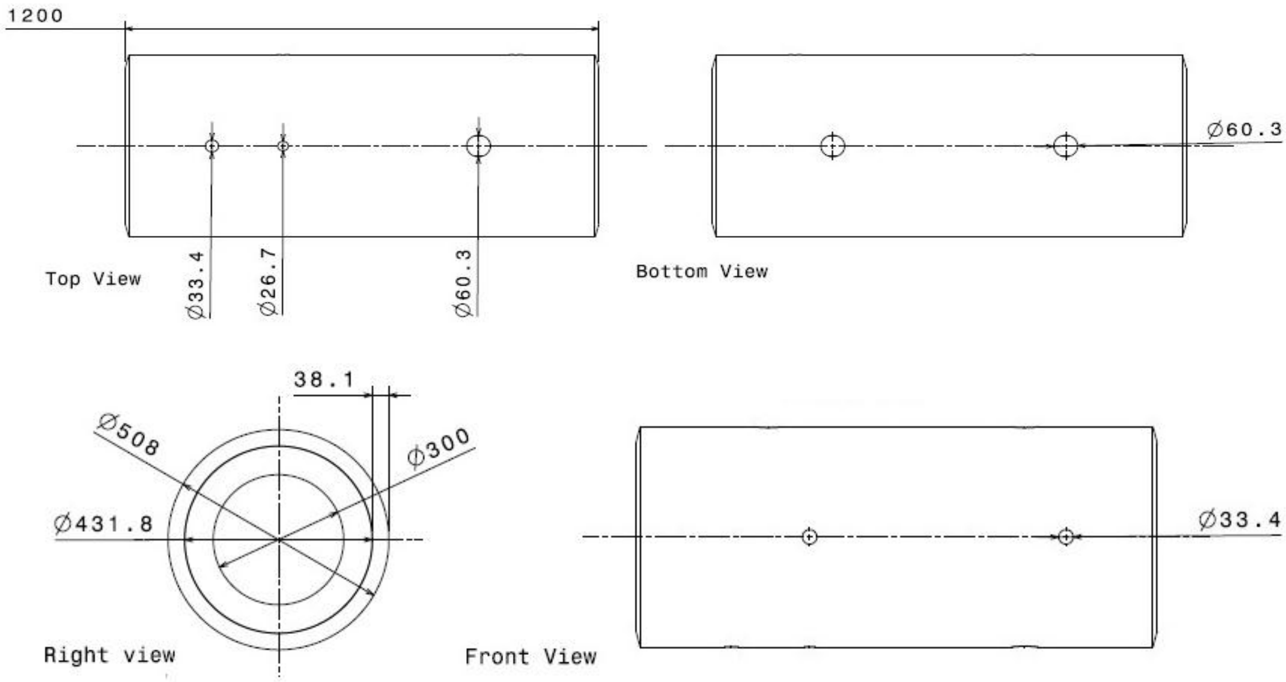
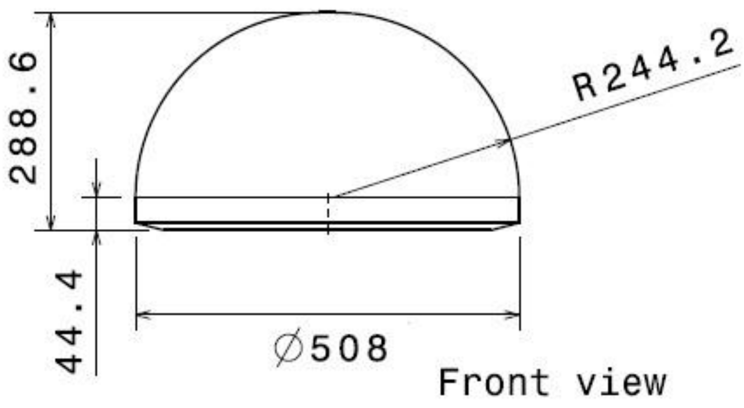
APPENDICES

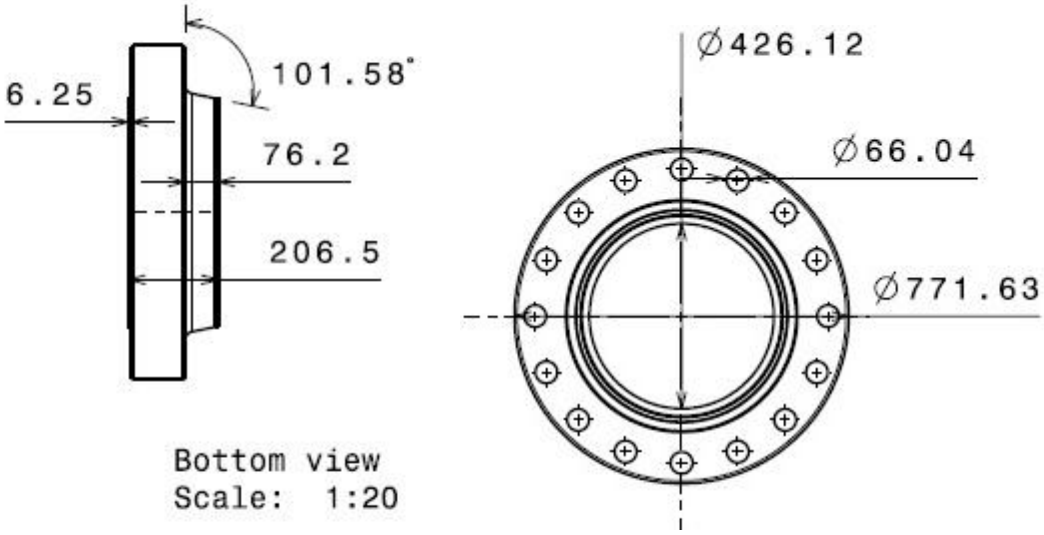
Appendix A.Detailed Dimension of the Hyperbaric Chamber Design



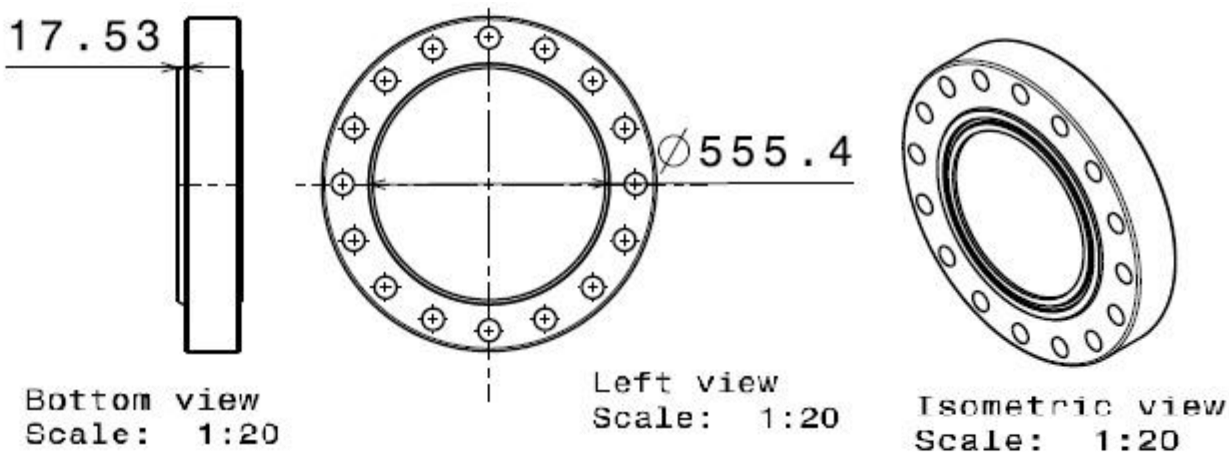
Appendix A.1 The sketch of the main vessel (not to scale)



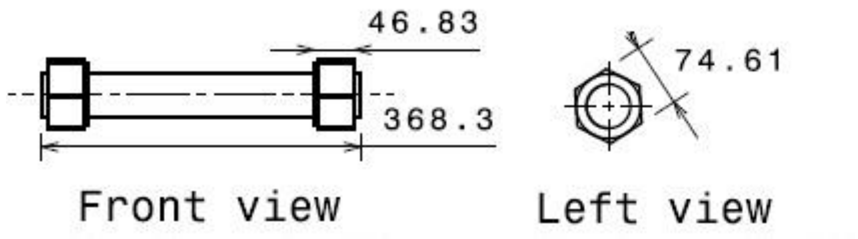
Appendix A.2 The Pipe Cap Sketch (not to scale)



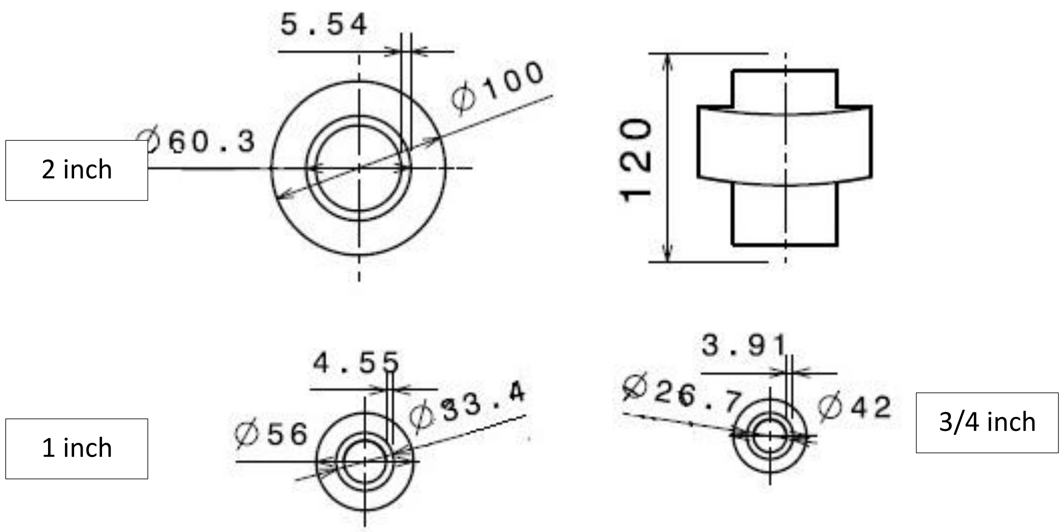
Appendix A.3 API 6A-6BX, Weld-neck Flange, 5000psi



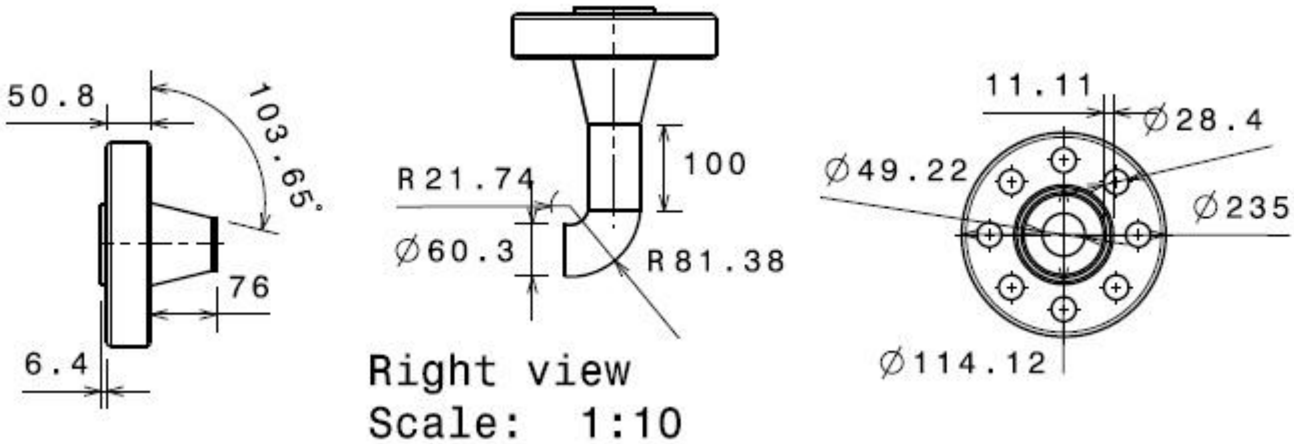
Appendix A.4 API 6A-6BX, Blind Flange, 5000psi



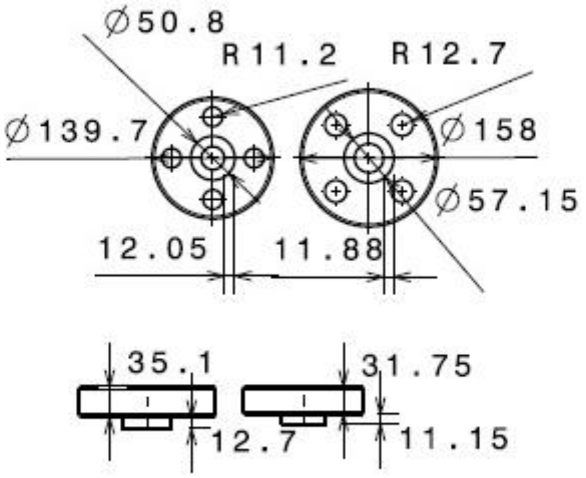
Appendix A.5 The sketch of the Bolts and studs in scale 1: 10



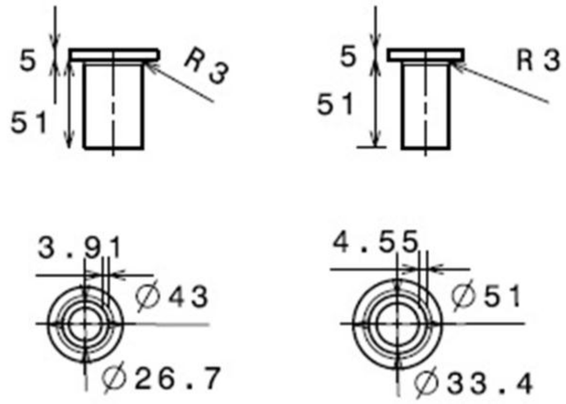
Appendix A.6 Nozzle compensation at a scale 1:5



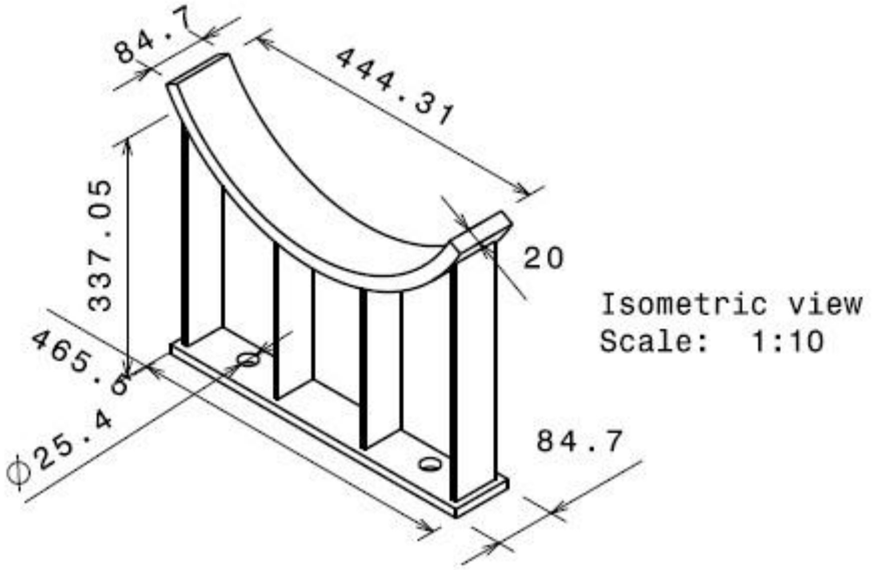
Appendix A.7 Sketch of the 2inch flange connected to the short Elbow



Appendix A.8 Sketch of weld-neck flanges with scale 1:10

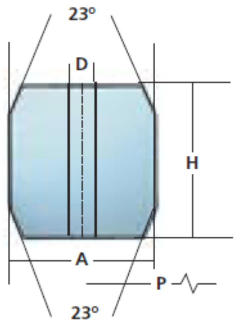


Appendix A.9 Sketch of the Stub-end with scale 1.5

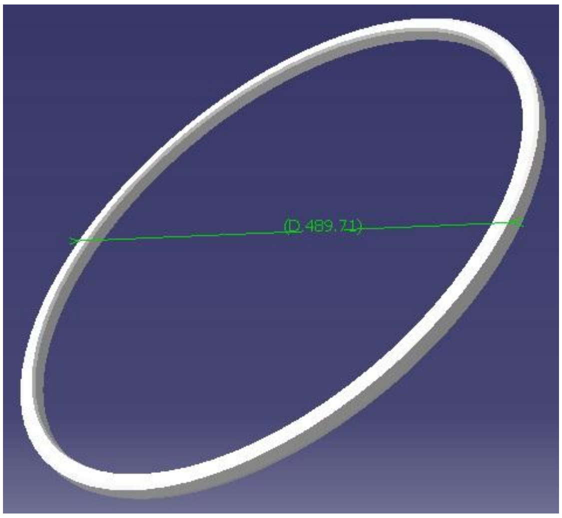


Appendix A.10 Sketch of Pressure Vessel Support, Saddle type with scale

Appendix B.CATIA Modeling of Hyperbaric Chamber



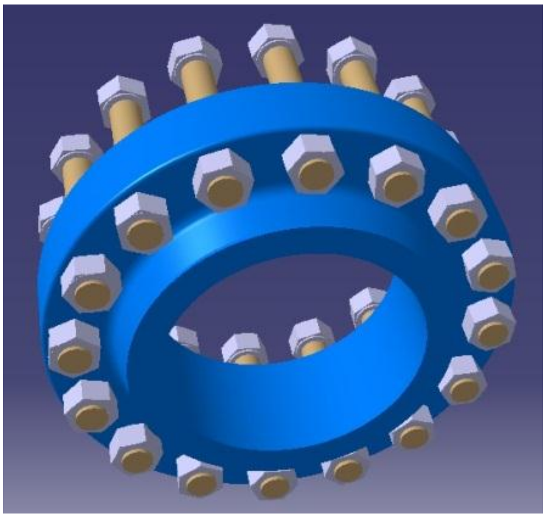
Tolerances (Inches)
A* (width of ring) +0.008, -0.000
D (hole size) + or - 0.02
H* (height of ring) + 0.008, - 0.000
OD (outside diameter of ring) + 0.000, - 0.006
23° (angle) + or - 0.25°



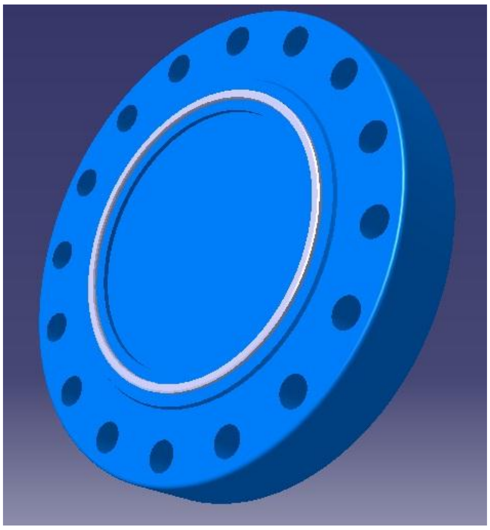
Where,

Outside Diameter of Ring **OD**	Height of Ring **H**	Width of Ring **A**	Hole Size **D**	Approx Weight KG's
Dimensions in Millimetres				
491.41	28.07	16.21	3.20	5.14
475.49	14.22	14.22	1.60	2.26

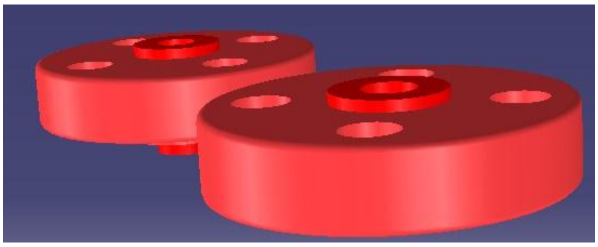
Appendix B.1. Ring Type Joint Dimension



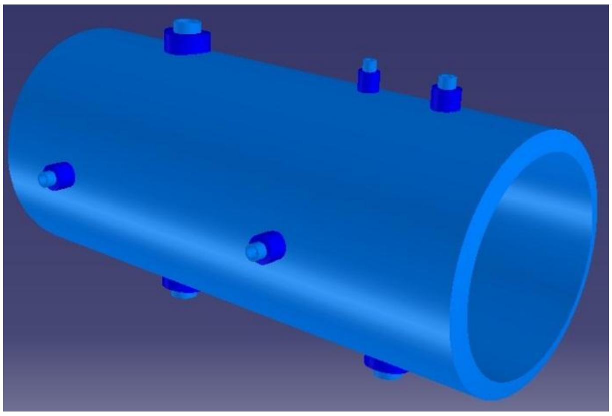
Appendix B.2. Stud and Bolts attached to Weld-neck Flange



Appendix B.3. Main Blind Flange and Ring Type Joint

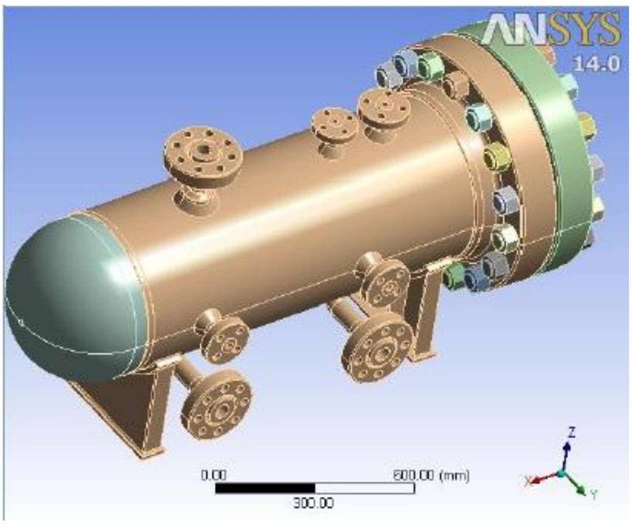


Appendix B.4. 1 inch and ¾ inch Lap Joint Flanges

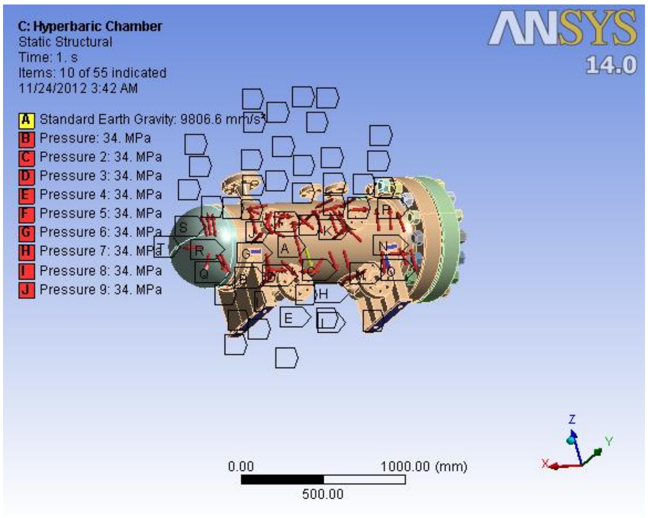


Appendix B.5. Main Cylindrical Vessel with Nozzle Compensation

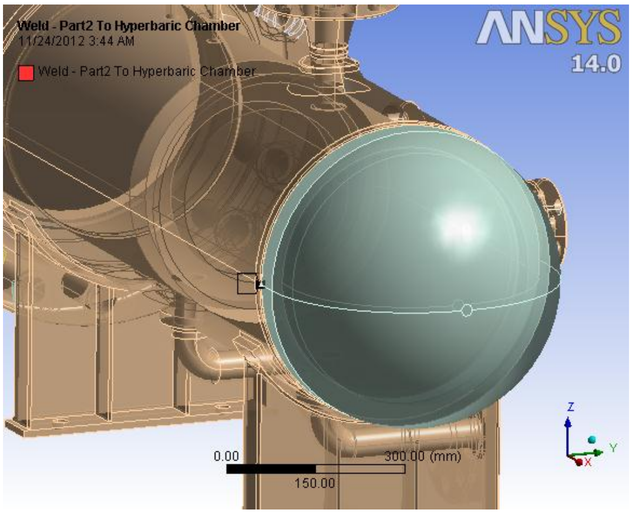
Appendix C. ANSYS Simulations of Hyperbaric Chamber



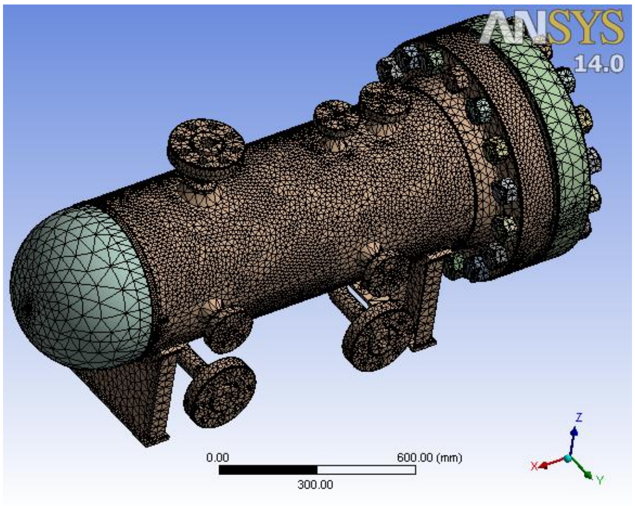
Appendix C.1 ANSYS 3D Modeling



Appendix C.2 ANSYS Applied Boundry Conditions

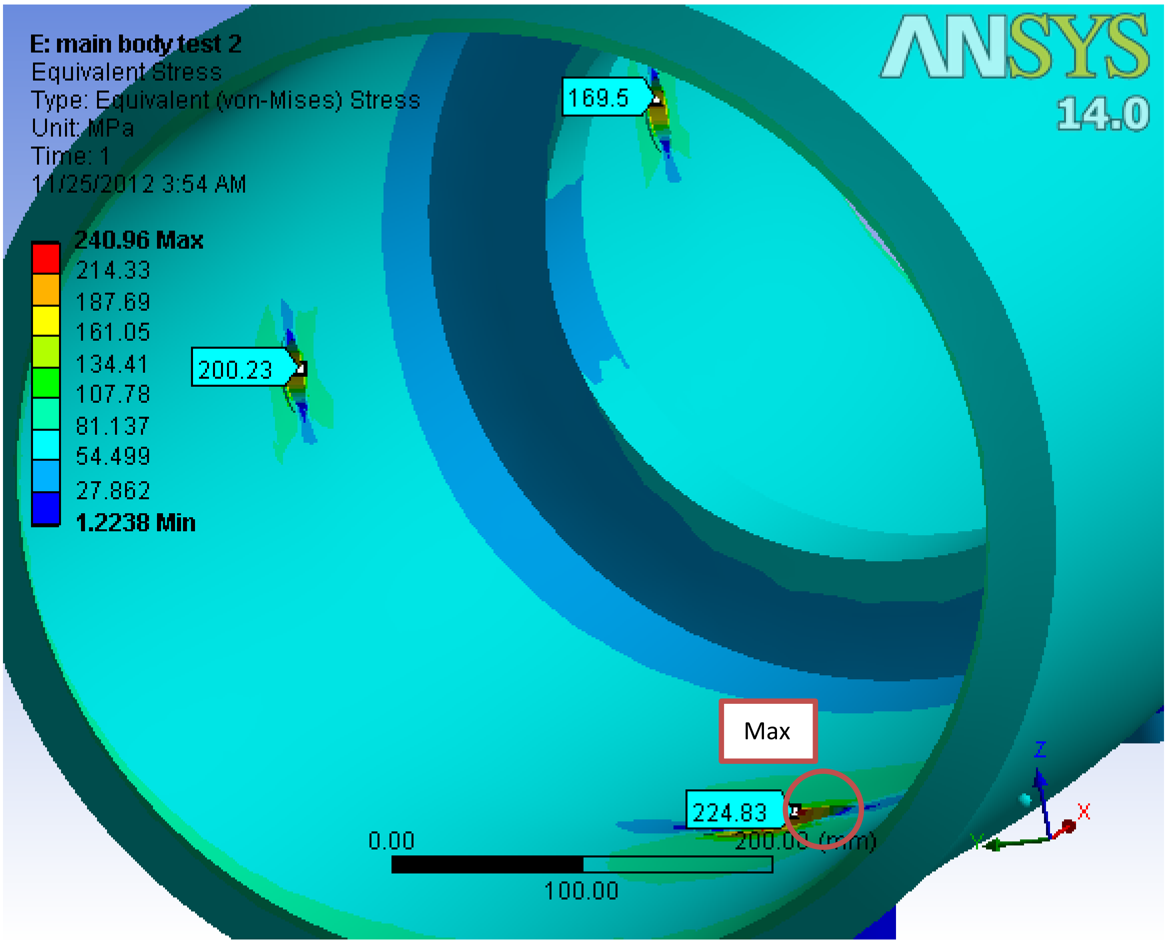


Appendix C.3 Applied Weld and Contact Point

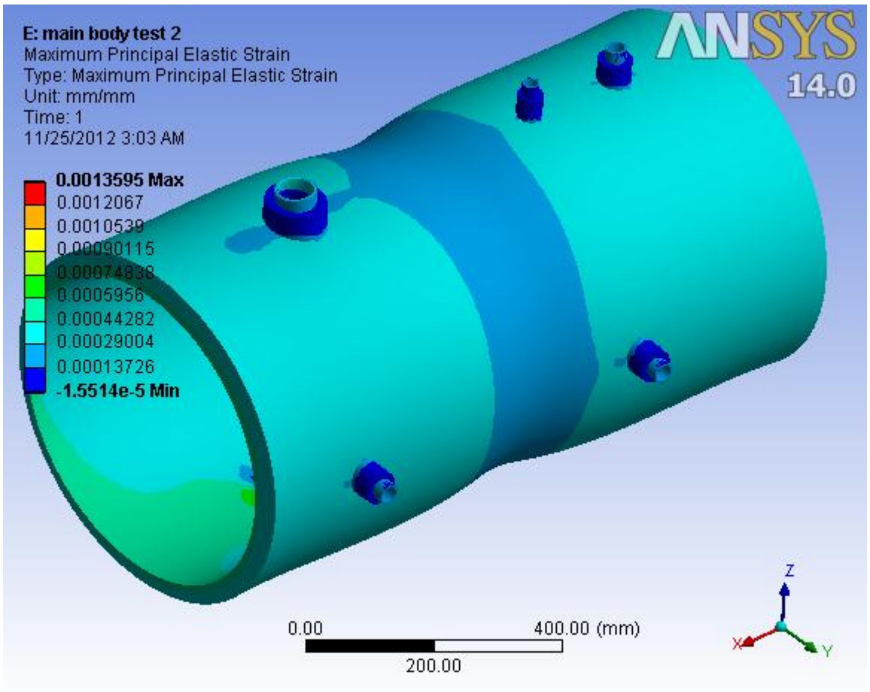


Appendix C.4 ANSYS Hyperbaric Chamber Meshi

Appendix D. ANSYS Results Simulation of the Main Vessel

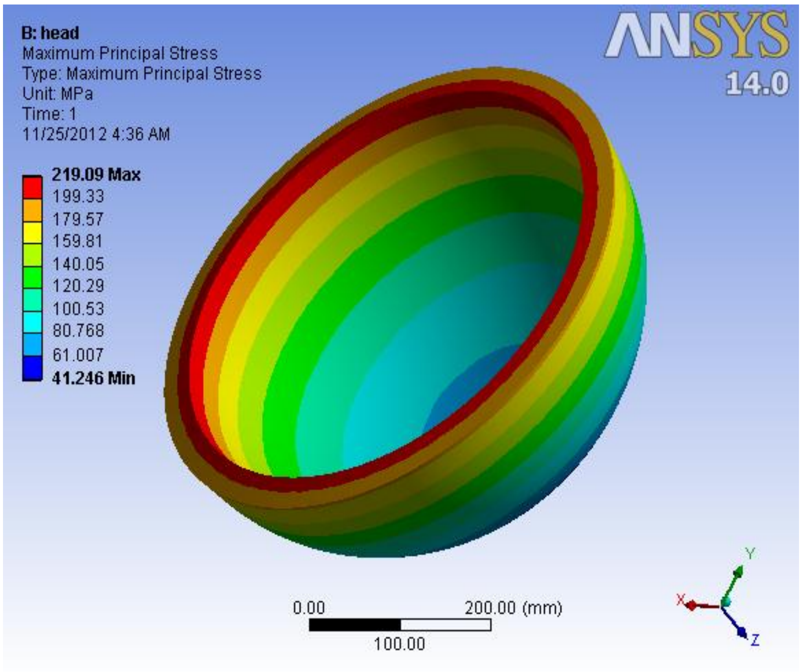


Appendix D.1 Equivalent Von-Mises Stress

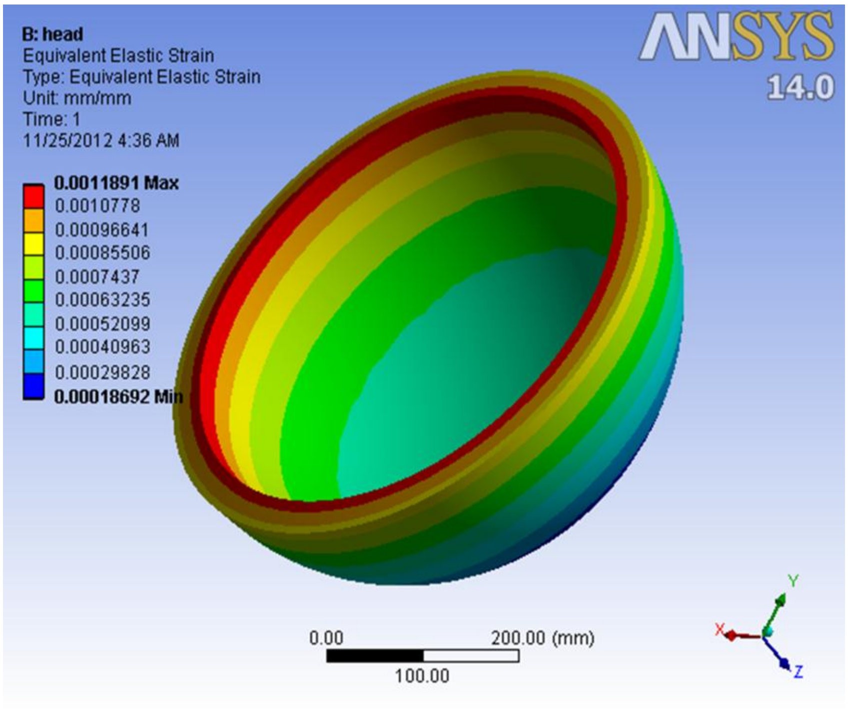


Appendix D.2 Maximum Principal Elastic Strain

Appendix E. ANSYS Results Simulations of Ellipsoidal Head

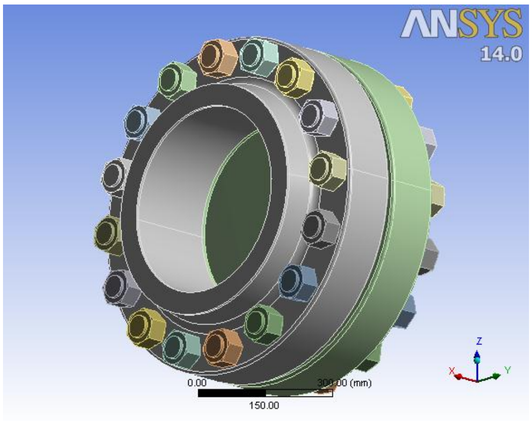


Appendix E.1 Maximum Principal Stress

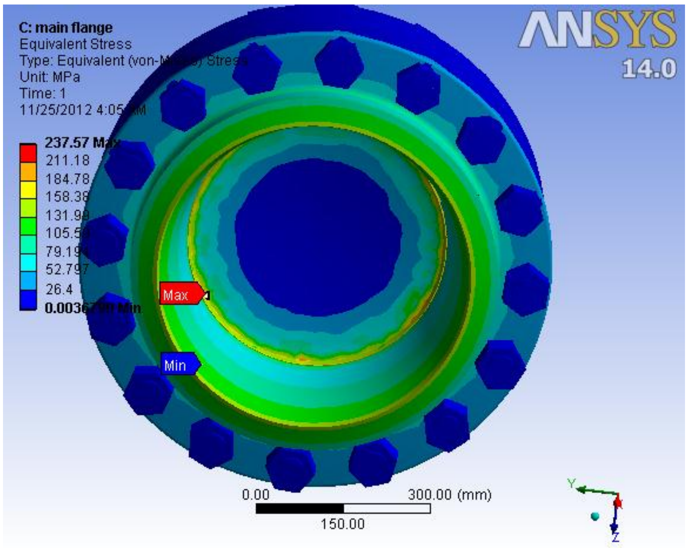


Appendix E.2 Equivalent Elastic Strain

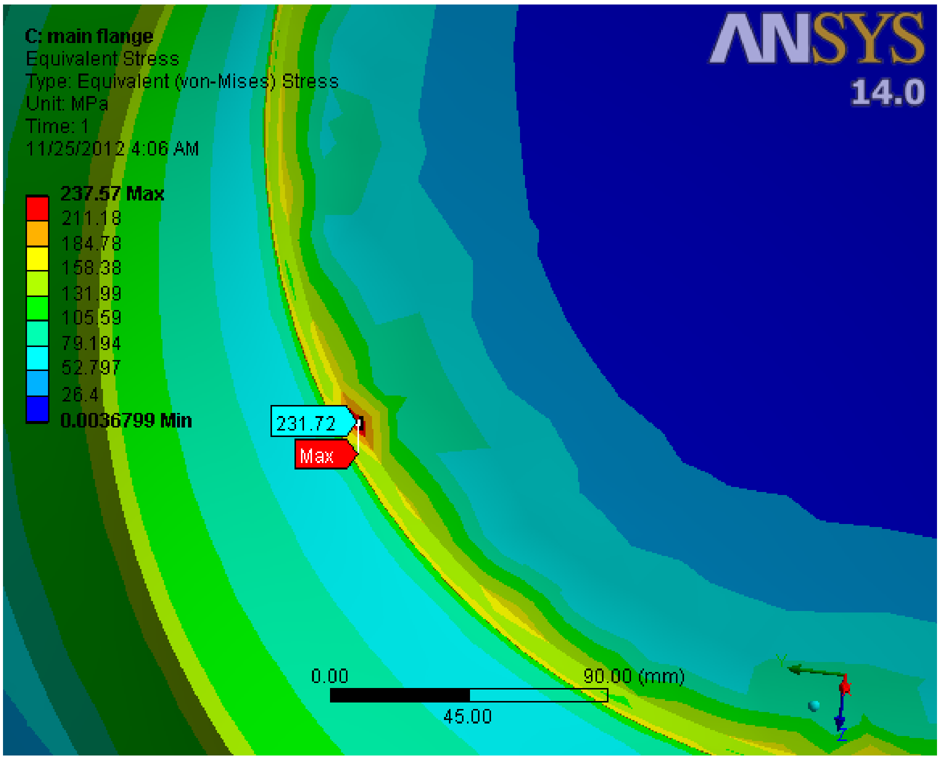
Appendix F. ANSYS Results Simulations of Flange Head Attached to Main Vessel



Appendix F.1 3D Modeling of ANSYS



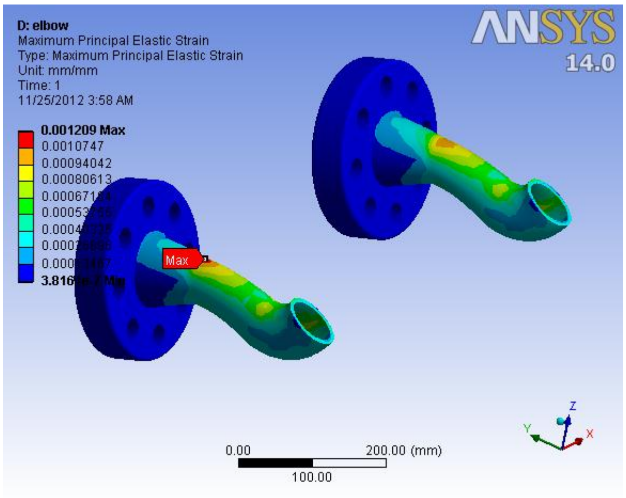
Appendix F.2 Equivalent Stress



Appendix F.3 Equivalent Stress at Maximum

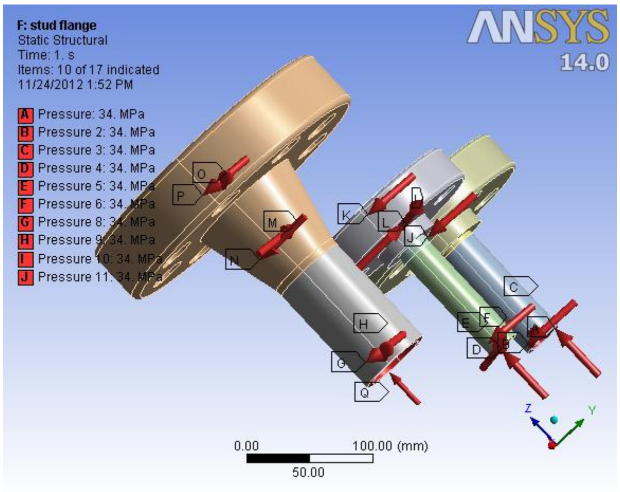
Appendix G. ANSYS Results Simulation of Elbow and Small Flanges

2inch Elbow attached to Weld-neck Flange

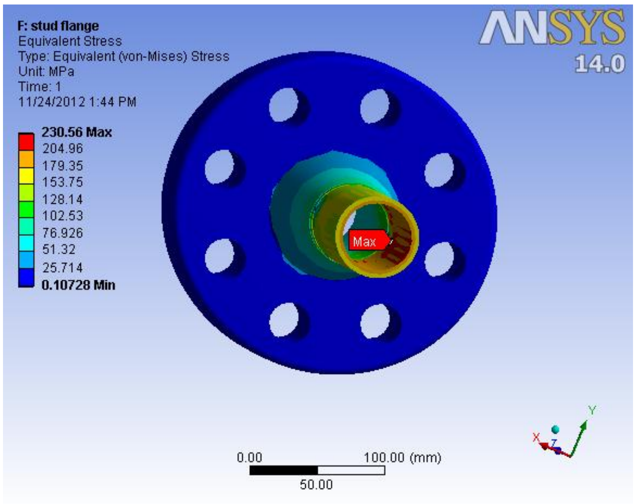


Appendix G.2 Maximum Principal Elastic Strain

Class 2500 Lap Joint Flange



Appendix G.3 ANSYS Apply Boundary Conditions



Appendix G.4 ANSYS Equivalent Stress

Appendix H. Project Gantt Chart

		FYP 1														FYP 2														
No.	WEEK	1	2	3	4	5	6	7	8	9	10	11	12	13	14	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Activities																														
1.0	Project Scope Validation																													
2.0	Project Introduction					*																								
3.0	Submission of Extended Proposal						*																							
4.0	Identify Design Criteria and Parameters																													
5.0	Idea Generation with the review on Literature																													
6.0	Proposal Defense									*																				
7.0	Submission of Interim Draft Report																													
8.0	Submission of Interim Report														*															
9.0	Conceptual Design													*																
10.0	Detailed Design																													
11.0	Modeling and Simulation																							*						
12.0	Analysis of Data																													
13.0	Submission of Progress Report																								*					
14.0	Conclusion and Recommendation																													
15.0	Pre-SEDEX Viva Presentation																													
16.0	Submission of Draft Report																											*		
17.0	Submission of Technical Paper and Dissertation																											*		
18.0	Oral Presentation																													
19.0	Submission od Project Dissertation																													*

(*) *key milestone*