Design Optimization and Stress Analysis of Pipes and Clamps of a Power Plant.

Submitted in partial fulfillment of the requirements for the award of degree of

BACHELOR OF TECHNOLOGY

IN

MECHANICAL ENGINEERING

Submitted by:

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Declaration

This is certify to that

- The thesis comprises my original work towards the degree of Bachelor of Technology in Mechanical Engineering at Nirma University and has not been submitted elsewhere for degree.
- Due acknowledgement has been made in the text to all other material used

Sign:

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I, Anirudh Topiwala (13bme003) give the undertaking that the Major Project entitled “Design Optimization and Stress Analysis of Pipes and Clamps of a Power Plant “submitted by us, towards the partial fulfillment of the requirements for the degree of Bachelor of Technology in Mechanical Engineering of Nirma University, Ahmedabad, is the original work carried out by us. We give assurance that no attempt of plagiarism has been made. We understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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This is to certify that, Mr. Anirudh Topiwala student of Mechanical engineering, 8^{th} Semester of, Institute of Technology, Nirma University has satisfactorily completed the project report titled Design Optimization and Stress Analysis of Pipes and Clamps of a Power Plant.

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The Project entitled **Design Optimization and Stress Analysis of Pipes and Clamps of a Power Plant** by Student **<u>ANIRUDH TOPIWALA (13BME003)</u>** is approved for the degree of Bachelor of Technology in Mechanical Engineering

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ABSTRACT

Coal based power plants are the most essential contributors to the total electricity generation in India. In a year, a total of 74 percentage of the electricity comes from the coal based power plant industry. Therefore there is a great need for reducing the overall cost of the power plant.

Also, with today's highly competitive market for coal based power plant, domestic as well as International players, there's a greater need in reducing the cost of the power plant with reduced construction to execution time. Because of this optimization of components have become a prime area in which active research is going on.

The maximum reduction of cost usually comes through the most basic but the most extensively used part of a power plant, which is the pipes and the clamps used in various packages.

The Demineralization and Pretreatment plant, requires large quantities of pipes to transport the demineralized water to the boiler and therefore, design and optimization of these pipes and the clamps used is the main objective of my report.

The optimization is carried out for the hold down clamps as they are the most efficient and the cheapest method of holding pipes when the pipes run on the very grounds. Also there ability to provide for axial displacement makes them the best type of clamp available in this case.

Key words: Design, Optimization, Ansys, FEA, Caesar II

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Nomenclature

tm	Thickness of pipe
t	Calculated thickness
c	sum of the mechanical allowances, ASME B1.20.1 [11]
D	outside diameter of pipe as listed in tables of standards or specifications or as measured
d	Inside diameter of pipe. For pressure design calculation, the inside diameter of the pipe is the maximum value allowable under the purchase specification.
E	Quality factor from Table A-1A or A-1B of ASME B31.1. [10] It depends on material and the method of manufacturing
P	Internal design gage pressure
S	Stress value for material from Table A-1 of ASME B31.1 [10]
W	Weld joint strength reduction factor in accordance with para. 302.3.5(e) of ASME 31.1 [10]
Y	Coefficient from Table 304.1.1 of ASME B31.1 [10].
t	Thickness of clamp
t1	Thickness of rib
d	Deformation of clamp

CHAPTER 1

Introduction

About two decades ago, in India, the design procedure for pipes and piping clamps for Power industry, Petrochemicals and Fertilizer Plants, in magnitude, depth and complexities were not fully evolved. Only in the recent past, we were exposed in detail to this field. Now we are self-sufficient in the field of piping clamps and design.

One of the major tasks in any process or power industry is the transportation of materials often in fluid from one place to another. The most commonly adopted method for the same is to force the fluid through the piping system. The piping system is the inter-connected piping subjected to the same set of design conditions. The piping system involves pipes but also fittings, valves and other specialties. These items are known as piping elements. Piping network is subjected to almost all the severest conditions of the plant such as high temperature, Pressure, flow and combination of these. To withstand these forces and to hold the pipes in place, robust and durable clamps are needed to be designed.

In the recent years, the trend is to develop better techniques so as to optimize the clamp design. Computer is being used extensively to obtain rapid solutions to the more complex problems of clamp design. More recently, it is being employed for production of piping detail drawings, piping isometrics, and bill of materials, cost estimation and control. Piping engineer has therefore a further responsibility in understanding and application of continually growing techniques of this nature.

A pipe support or pipe hanger is a designed element that transfer the load from a pipe to the supporting structures. The load includes the weight of the pipe proper, the content that the pipe carries, all the pipe fittings attached to pipe, and the pipe covering such as insulation. The four main functions of a pipe support are to anchor, guide, absorb shock, and support a specified load. Pipe supports used in high or low temperature applications may contain insulation materials. The overall design configuration of a pipe support assembly is dependent on the loading and operating conditions.

1.1 Objective and Scope of the Research

The objective of this thesis is to design a pipe with its hold down clamp and optimize the clamp to reduce the cost of production as much as possible.

The pipe in consideration is for connecting the condensate storage tank and demineralized water storage tank.

First the pipe thickness is calculated, which once finalized will undergo different loading conditions to calculate the stresses induced in the pipe.

After the design of the pipe is complete, the clamp is designed and various parameter optimization is carried out.

Finite element analysis method have been used as the key method to evaluate the stresses generated in the pipes and the clamp. The clamp had to be designed so as to withstand the forces under three different loading conditions.

The principle objective of the present research are:

- To calculate the thickness of a pipe using the standards available.
- To calculate the stresses induced in the pipes.
- To design and optimize the hold down clamp so as to withstand different loading conditions.

1.2 Problem Specification

The problem for the thesis is to design a pipe and a clamp for the connection between the condensate storage tank and demineralized water storage tank. The design should be able to withstand the various forces acting on the pipe and clamp as well as should be easily manufacturable at the least possible cost.

1.3 Methodology

The design methodology followed is for two components in specific, Pipes and clamps. For pipe design, the thickness of the pipe is initially calculated after which stress analysis is carried out. If the Stress analysis is not satisfactory the position of various supports are changed.

The clamp design methodology is also similar. In this, first various catalogues are studied for the same loading conditions to get an approximate ides of the various design dimensions. After studying these, the design parameters are optimized so as to reduce the overall material consumption. Also the ease of manufacturing is taken into account whilst setting the design parameters.

After setting the geometry of the clamp, stress analysis is carried using FEA in Ansys. The results are compared to ASME standards to check the stress limits and the design is finalized.

1.4. Contribution

I was able to actively participate in the day to day calculations that goes into designing a pipe and pipe support systems. I was also able to contribute to the complete pipe thickness calculation with stress analysis for the condensate storage tank connections package.

On a later stage I was able to provide optimized solutions for various cases of clamp design. I optimized the design parameters for the clamp. There were a total of three different loading cases with two different loading application criteria.

Finally, I was also able to carry out fillet radius and overall length optimization for the clamp, which in turn enabled us to finalize upon the design parameters which will lead to minimum cost and simplistic manufacturing.

1.5 Layout of the Thesis

The thesis mainly consists of two sections, which is the Pipe design and the clamp design. Initially, the literature is provided for the package for which the pipe and clamp are to be designed. The various brochures studied for estimating clamp parameters are also presented here.

In the first section or the Pipe design section, we first understand the problem, after which we start with the thickness calculations. After this, the stress analysis for the pipe is presented.

In the second section, we start with the need for clamp design. Which is followed by material specification and the various optimizations carried out for different loading conditions.

Finally, a brief summarization is presented to go through all the finalized designs for all the different loading conditions.

CHAPTER 2

LITERATURE REVIEW

2.1 Power Plant Systems

2.1.1 Introduction to Power Plant

A power plant may be defined as an assembly of machines or equipment that generates and delivers a flow of mechanical or electrical energy. The main equipment for the generation of electric power is the generator. When coupling it to a prime mover that runs the generator, electricity is generated. The power plant itself must be useful economically and environmental friendly to the society. [1]

The primary aim of a Power Plant Engineer is "To maximize conversion of some other form of energy to the electric power output and at the same time to minimize pollution impact."

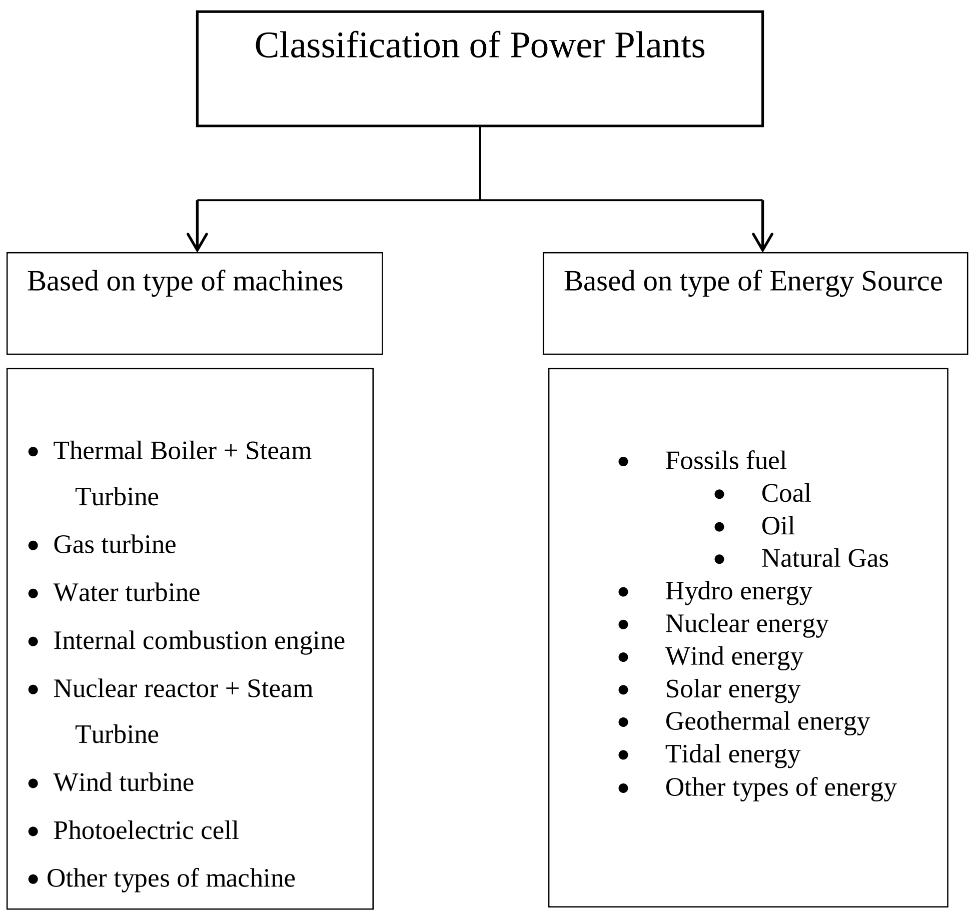


Fig 2.1: Classification of Power Plant

2.1.2 Rankine Cycle

The Rankine cycle is a vapour power cycle that forms the thermodynamic basis for most steam power plants. In the Rankine cycle steam flows to a turbine, where part of its energy is converted to mechanical energy. The reduced-energy steam flowing out of the turbine condenses to water in the condenser. [2]

A feed water pump returns the condensed liquid (condensate) to the boiler. The rejected heat from the steam entering the condenser is transferred to a separate cooling water loop.

The ideal Rankine cycle does not involve any internal irreversibility and consists of the following four processes:

- 1 to 2: Water from the condenser at low pressure is pumped into the boiler at high pressure. This process is reversible adiabatic.
- 2 to 3: Water is converted into steam at constant pressure by the addition of heat in the boiler.
- 3 to 4: Reversible adiabatic expansion of steam in the steam turbine.
- 4 to 1: Constant pressure heat rejection in the condenser.

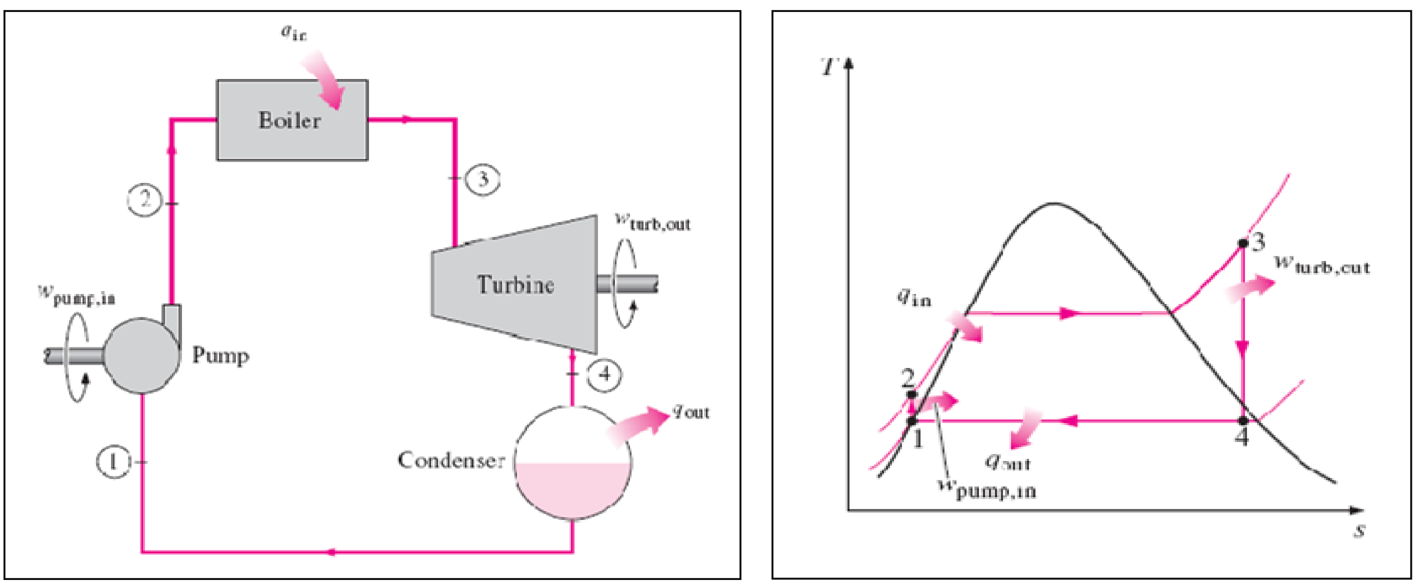


Fig 2.2: T-S & schematic operating diagram of Rankine cycle

2.1.3 Regenerative Rankine Cycle

In regenerative feed water heating cycle, part of the steam is extracted after partial expansion in the turbine and is used to heat up the feed water going to the boiler. In this process, superheat and latent heat of extracted steam is transferred to feed water to raise its temperature, i.e. sensible heat addition in the feed water is carried out before reaching the boiler. The drop formed due to condensation of extracted steam is recycled in to the feed water cycle at the appropriate point. [2]

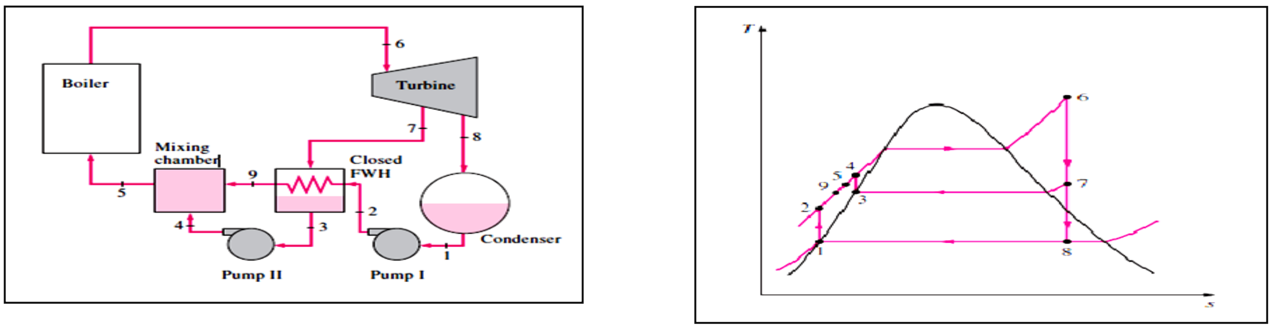


Fig 2.3: Regenerative Rankine Cycle

2.2 Demineralization and Pretreatment Plant

The objective of water treatment is to produce a boiler feed water so that there shall be:

(a) No scale formation causing resistance to passage of heat and burning of tube
(b) No corrosion
(c) No priming or foaming problems.

This will ensure that the steam generated shall be clean and the boiler plant will provide trouble free uninterrupted service.

As the types of boiler are not alike their working pressure and operating conditions vary and so do the types end methods of water treatment. Water treatment plants used in thermal power plants are designed to process the raw water to a water with very low in dissolved solids known as demineralized water. No doubt, this plant has to be engineered very carefully keeping in view the type of raw water to the thermal plant, its treatment costs and overall economics. [3]

Actually, the type of demineralization process chosen for a power station depends on three main factors:

(a) The quality of the raw water.
(b) The degree of deionization i.e. treated water quality
(c) Selectivity of resins.

Figure below shows a schematic diagram of water treatment process which is generally made up of two sections: Pretreatment section and Demineralization section.

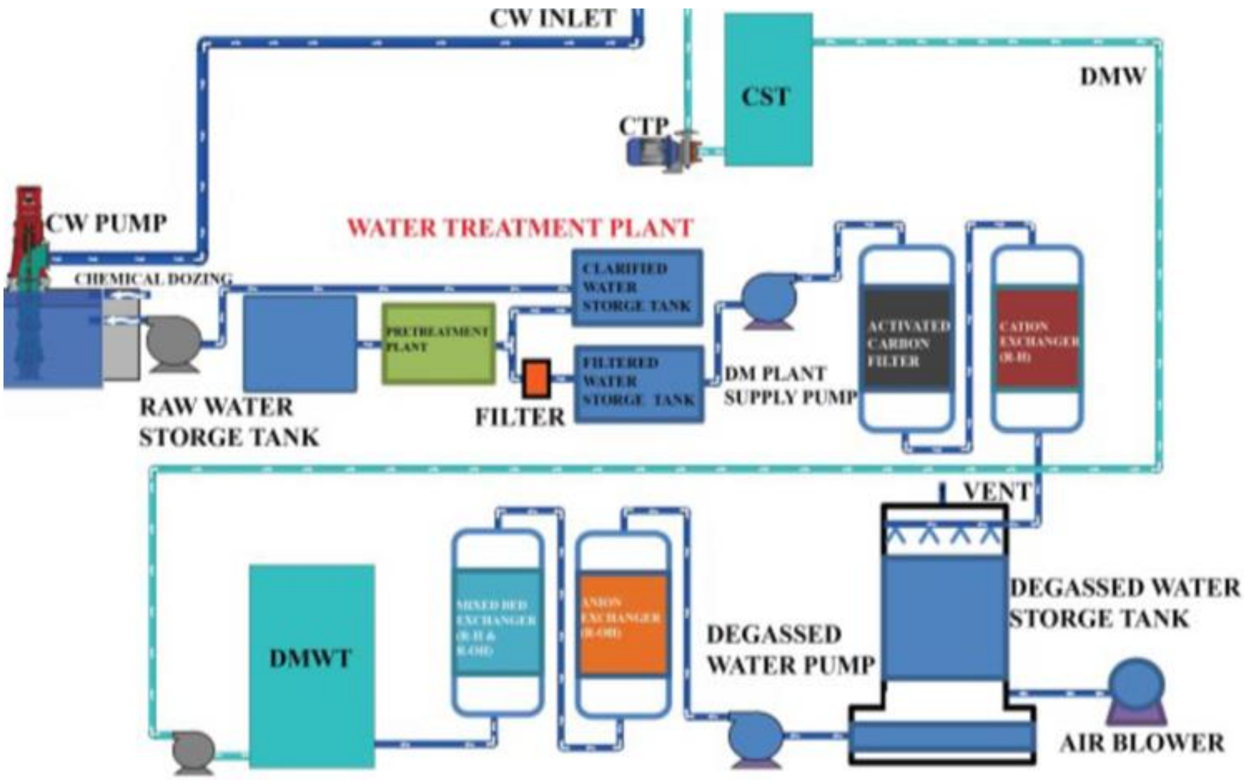


Fig 2.4: Flow diagram of DMPT plant

2.2.1 Pretreatment section

Pretreatment plant removes the suspended solids such as clay, silt, organic and inorganic matter, plants and other microscopic organism. The turbidity may be taken as of two types of suspended solids in water. Firstly the separable solids and secondly the non-separable solids (colloids). The coarse components, such as sand, silt etc., can he removed from the water by simple sedimentation. Finer particles however will not settle in any reasonable time and must be flocculated to produce the large particles which are settle able. Long term ability to remain suspended in water is basically a function of both size and specific gravity. The settling rate of the colloidal and finely divided (approximately 001 to 1 micron) suspended matter is so slow that removing them from water by plain sedimentation in tanks having ordinary dimensions is impossible. Settling velocity of finely divided and colloidal particles under gravity also is so small that ordinary, Sedimentation is not possible. It is necessary, therefore, to use procedure which "agglomerate the small particles into larger aggregates, which have practical settling velocities. [3]

The term " Coagulation' and 'flocculation" have been used indiscriminately to describe process of turbidity removal. 'Coagulation' means to bring together the suspended particles. The process describes the effect produced by the addition of a chemical Al2 (SP4)3 to a colloidal dispersion resulting in particle destabilization by a reduction of force tending to keep particles apart. Rapid mixing is important at this stage to obtain uniform dispersion of the chemical and to increase opportunity for particles to particle contact. This operation is done by flash mixer in the clarifloculator. Second stage of formation of settle able particles from destabilized colloidal sized particles is termed a "flocculation". Here coagulated particles grow in site by attaching to each other. In contrast to coagulation where the primary force is electrostatic or interionic, flocculation occurs by chemical bridging. Flocculation is obtained by gentle and prolonged mixing which converts the submicroscopic coagulated particle into discrete, visible & suspended particles. At this stage particles are large enough to settle rapidly under the Influence of gravity and may be removed.

If pretreatment of the water is not done efficiently then consequences are as follows:

(a) SiO2 may escape with water which will increase the anion loading.

(b) Organic matter may escape which may cause organic fouling in the anion exchanger beds. In the pro-treatment plant chlorine addition provision is normally made to combat organic contamination.

(c) Cation loading may unnecessary increase due to addition of Ca (OH) 2 in excess of calculated amount for raising the pH of the water for maximum floc formation and also Al2 (OH) 3 may precipitate out. If less than calculated amount of Ca (OH) 2 is added, proper pH flocculation will not be obtained and silica escape to demineralization section will occur, thereby increasing load on anion bed.

Typical Water Pre Treatment Layout

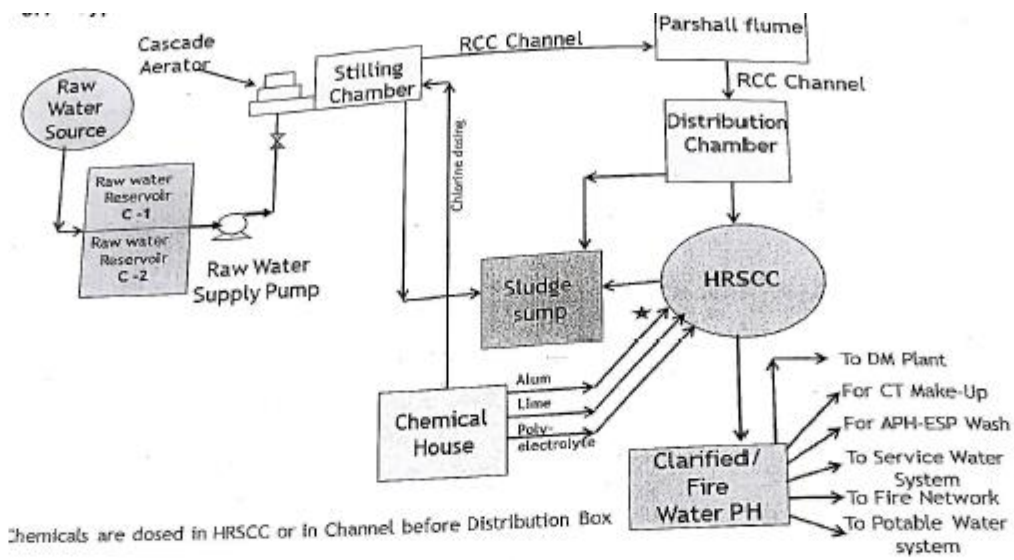


Fig 2.5: Layout of Water Pretreatment Plant

- Raw water is pumped through Cascade Aerator to Clarifier. Pump head is selected in such a way that water flows to the top of Aerator. Hydraulic gradient is provided from Aerator to Distribution box.
- In cascade aerator water gets separated into small particles, leading to removal of iron and dissolved gases from water.
- From Cascade aerator, water is collected at stilling chamber for breaking the turbulence in the water. Chlorine is dosed in stilling chamber.

- Overflow of the stilling chamber is diverted to distribution chamber via RCC channel. In this channel RCC parshall flume is created to facilitate flow measurement. - From Parshall flume water is collected in the distribution chamber. Distribution chamber breaks turbulence in the water.
- Clarifier removes sludge from raw water. Chemicals are dosed in the clarifier or ahead of clarifier.
- Clarifier mechanism ads as an agitator and stirs raw water for sludge separation.
- Over flow of the clarifier flows to clarified water storage tank through RCC channel.
- The chemicals which are dosed in or before clarifier are Alum, Lime, Poly electrolyte and Chlorine. These are stored and prepared in chemical house.
- Sludge generated in the clarifiers is discharged in the sludge sump. This sludge is pumped to the effluent treatment plant for further treatment.
- Clarified water stored in the tank is pumped to various applications from pump house with the help of horizontal centrifugal pumps.

Major components of PT Plant:

1) Cascade Aerator

Purpose: For Removal of Iron from raw water. Dissolved gases like CO2, Methane, Volatile matter, Chlorine, Ammonia, Hydrogen Sulphide, etc. also removed from raw water. [4]

Salient Features:

- Circular stepped cascade design
- RCC Structure
- Surface Flow Rate — 0.03 cum/Sq. in/hr

2) Stilling Chamber

Purpose: To break turbulence of water and ensure smooth and laminar flow after aerator. Normally chlorine is dosed at the stilling chamber.

Salient Features:

- RCC Chamber Flow Velocity 0.03-0.05 m/sec
- Retention time in stilling chamber : 60 Sec
- Drain of the stilling chamber is routed to sludge sump.

Fig 2.6: Cascade Aerator

3) Parshall Flume & Distribution Chamber

Purpose: Parshall flume is a convergent-divergent RCC structure to facilitate flow measurement. Distribution chamber is used to streamline the flow after parshall flume. [4]

Salient Features:

- Chemical dosing is done in channel between parshall flume and Distribution chamber.
- Chemicals are added in this area hence chemical resistant paint should be applied on RCC wall. Outlet of distribution chamber is connected to clarifier inlet via 1000-1100 NB pipe.
- Drain of distribution chamber is routed to sludge sump.

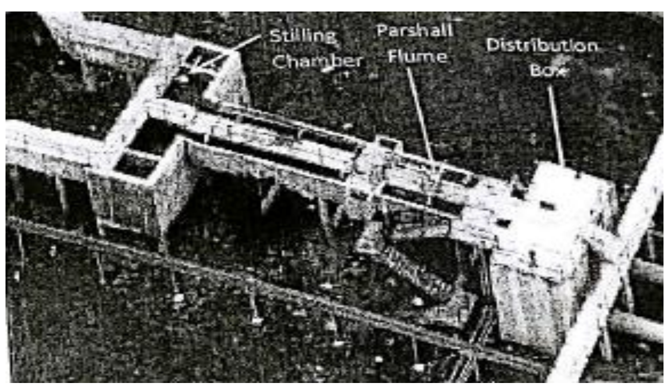


Fig 2.7: Stilling Chamber, Parshall Flume and Distribution Box

4) Clarifier

Purpose: To clarify raw water and generate clarified water.

Salient Features:

- Chemicals and whirl by scrapper remove sludge from the water and clarify it.
- Alum, Lime and Poly-electrolyte are dosed in clarifier or before clarifier
- Alum is used for coagulation of small sludge particles.
- Lime is used for chemical precipitation of the sludge
- Poly-electrolyte is used to optimize lime consumption.
- Rotating scrapper mechanism rotates at desire RPM to remove sludge from water.
- HRSCC has flocculation and clarification zone.
- In flocculation zone small sludge particles coagulate and prepare a dense sludge. Overflow of this zone enters in the clarification zone.
- More clarification of the water done in the clarification zone and overflow of this zone is routed to clarified water reservoir with the help of Outer launder and channel.
- Sludge collected at the bottom will be discharged in the sludge pit with the help of telescopic sludge disposal arrangement.
- Individual or combined sludge pit for each clarifier provided.
- Sludge disposal pumps are provided to discharge sludge to effluent treatment plan. [4]

Fig 2.8: Onsite Pictures of clarifier at Malva II Plant.

5) Clarifies Water Storage Tanks and Pump House

Purpose: To store water generated after clarification

Salient Features:

- Clarified water generated from clarifier collected at clarified water storage tank.
- Partially underground RCC tank
- Storage capacity: 6 hours of plant clarified water requirement.
- Normally capacity of clarifier water storage tank also takes care of fire water requirement for the Plant.
- Generally Clarified water pump house houses:
    - CT Make-up Pumps
    - APH/ESP area wash Water Pumps
    - DM Water Plant Feed Pumps
    - Service Water Pumps
    - Potable Water Plant Feed Pumps
    - Fire water Pumps.

6) Chemical House

Purpose: It is used for storage, preparation and dosing of chemicals in Pre-Treatment Plant.

Salient Features:

- It is a RCC building.
- It houses Dosing tanks, Dosing pumps of the chemicals like Alum, lime, Poly-electrolyte and chlorine.
- Chemical storage area also provided in the building. Normally chemical storage of one month requirement is provided

- Normally RCC tanks are provided for the lime & alum dosing.
- Electrical hoist provided for the chemical handling.

2.2.2 Demineralization

This filter water is now used for demineralizing purpose and is fed to Cation exchanger bed, but en route being first dechlorinated, which is either done by passing through activated carbon filter or injecting along the flow of water, an equivalent amount of sodium sulphite through some stroke pumps. The residual chlorine which is maintained in clarification plant to remove organic matter from raw water is now detrimental to cation resin and must be eliminated before its entry to this bed. [3]

Normally, the typical scheme of demineralization up to the mark against an average surface water, is three bed system with a provision of removing gaseous carbon dioxide from water before feeding to Anion Exchanger. Now, let us see, what happens actually in each bed when water is passed from one to another.

Resins, which are built on synthetic matrix of a styrene divinyl benzene copolymer, are manufactured in such a way that these have the ability to exchange one ion for another, hold it temporarily in chemical combination and give it to a strong electrolytic solution. Suitable treatment is a so given to them in such a way that a particular resin absorbs only a particular group of ions. Resins, when absorbing and releasing cationic portion of dissolved salts, is called cation exchanger resin and when removing anionic portion is called anion exchanger resin.

The present trend is of employing strongly acidic cation exchanger resin and strongly basic anion exchanger resin in a DM Plant of modern thermal power station. We may see that the chemically active group in a cationic resin is Sox-H (normally represented by RIO and in an anionic resin the active group is either tertiary amine or quaternary ammonium group (normally the resin is represented by ROH). The reaction of exchange may be further represented as below:

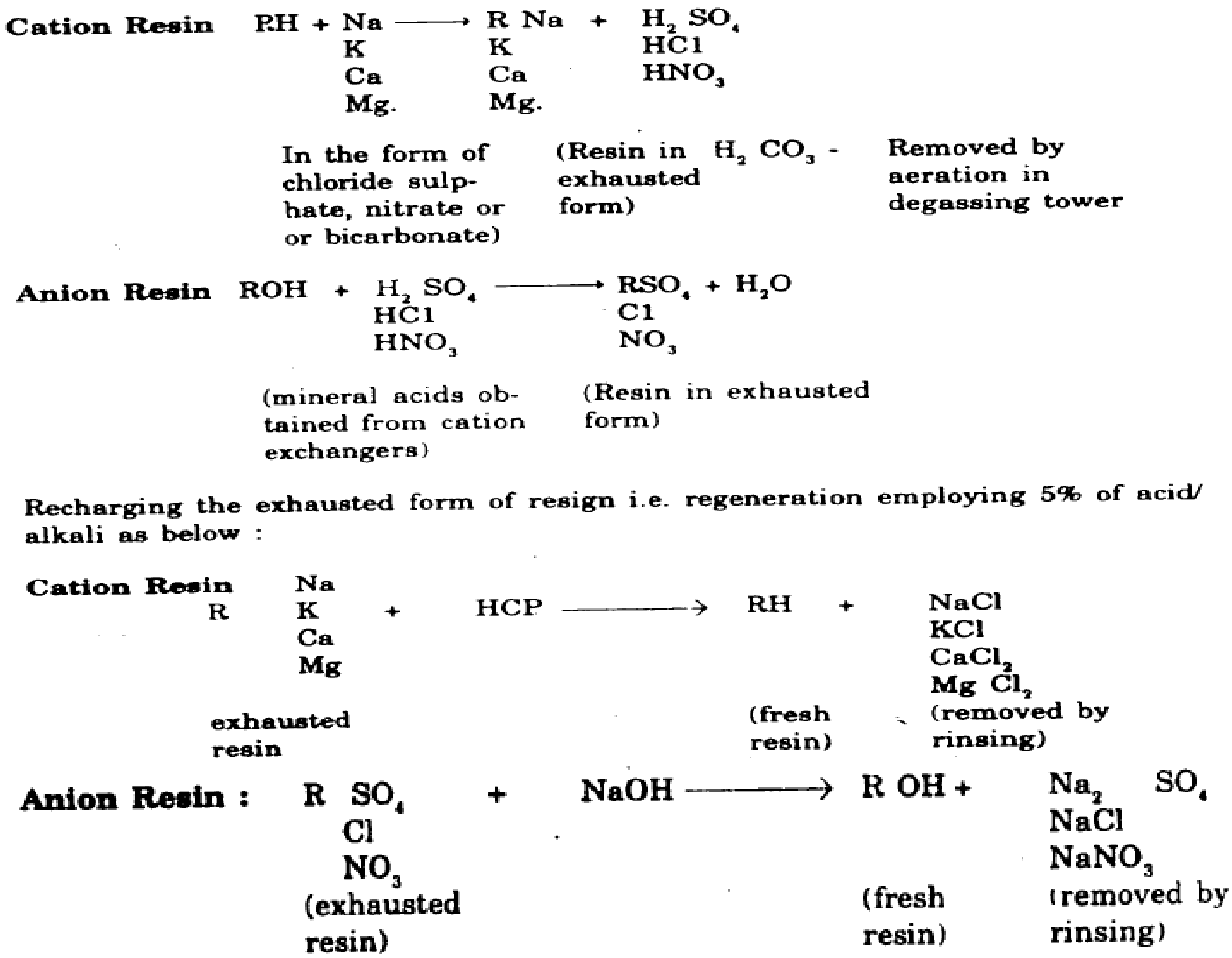


Fig 2.9: Reactions at cation and anion exchanges

As seen above the water from the ex-cation contains carbonic acid also sufficiently, which is very weak acid difficult to be removed by strongly basic anion resin and causing hindrance to remove silicate ions from the bed. It is therefore a usual practice to remove carbonic acid before it is led to anion exchanger bed. The ex-cation water is trickled in fine streams from top of a tall tower packed with reaching rings, and compressed air is passed from the bottom. Carbonic acid break into CO, and water mechanically (Henry's Law) with the carbon dioxide escaping into the atmosphere. The water is accumulated in suitable storage tank below the tower, called degassed water dump, from where the same is led to anion exchanger bed, using acid resistant pump.

The ex-anion water is fed to the mixed bed exchanger containing both cationic resin and anionic resin. This bed not only takes care of sodium slip from cation but also silica slip from anion exchanger very effectively. The final output from the mixed bed is an extra-ordinarily pure water having less than 0.2/Mhc conductivity, PH 7.0 and silica content less than 0.02 ppm. Any deviation from the above quality means that the resins in mixed bed are exhausted and need regeneration. Regeneration of the mixed bed first calls for suitable back washing and settling, so that the two types of resins are separated from each other. Lighter anion resin rises to the top and the heavier cation resin settles to the bottom. Both the resins are then regenerated separately with alkali and acid, rinsed to the desired value and air mixed, to mix the resin again thoroughly. It is then put to final rinsing till the desired quality is obtained. [4]

It may be mentioned here that there are two types of strongly basic anion exchanger. Type II resins are slightly less basic than type I, but has a higher regeneration efficiency than type I. Again as type II resins are unable to remove silica effectively. Type 1 resins also have to be used for the purpose. As such, the general condition so far prevailing in India, is to employ type II resin in anion exchanger's bed and type I resin in mixed bed for the anionic portion). It is also a general convention to regenerate the above two resins under through fare system i.e. the caustic soda entering into mixed bed for regeneration of type I anion resin, is utilized to regenerate type type II resin in anion exchanger bed The concept of utilizing the above resin and mode of regeneration is now a days being switched over from the economy to a more higher cost so as to have more stringent quality control of the final D.M. Water.

Internal Treatment

This final D.M effluent is then either led to hot well of the condenser directly as make up to boilers, or being stored in D.M. Water storage tanks first and then pumped for makeup purpose to boiler feed.

As the D.M. Water has a good affinity to absorb carbon dioxide and oxygen, and both are extremely harmful to metal surfaces for their destruction like corrosion, these have to be removed before it is fed to boiler. This is being done in deaerator. Still the residual oxygen which is remaining in the water is neutralized by a suitable doze of hydrazine, at the point after deaerator. To have further minimum Corrosion, the pH of feed water is to be maintained at around 9.0 for which purpose ammonia in suitable doze is added to this make up water at a point along with hydrazine stated above.

2.3 Pipe Supports

This section deals with the related work done in the field of pipe supports. The catalogue data for different pipe supports is also summarized.

Bergen Pipe Supports [5]: In this catalogue a detailed selection procedure is given for Ancillary Items or hold down piping clamps in our case.

Ancillaries are the hardware that complement spring supports and allow the connection of the pipe to the building structure. As simple as a pipe shoe or comprised of many items from a beam clamp through hanger rods, spreader beams and pipe clamps.

Selecting Ancillaries: The three main factors that will decide on which ancillary items you require are –

- The weight of the pipe being supported.
- The general arrangement of the support being designed.
- The temperature of both the pipe and the surrounding environment.

When thinking about the weight of the pipe or the load that the ancillaries will be expected to carry it is important to consider all possible loadings.

- Normal operating load, including the weight of heavy pipe clamps, riser clamps or spreader beams should be taken into account.
- It is important to consider any other factors that may cause increased loading during the whole operating life of the plant, examples are snow loading, wind loading, surge loading, temporary loads due to access and many other possibilities.

With regard to the actual support arrangement, there are many factors that will influence the choice of ancillary to be used.

- How is the support fixed? Is it hanging from or standing on steelwork or concrete?
- Is there a clear path to the pipe from the point of attachment to the structure?
- Is the pipe moving horizontally and vertically?
- In which direction are the forces being applied?

Finally, temperature has a significant influence on the ability of steel to withstand stress. High temperatures (above 350`C) cause steel to lose strength and we must begin to consider the phenomenon of creep. Low temperature, below 0` C, causes steel to become brittle and reduce its ability to withstand sudden increases in load.

Quite obviously the temperature of the fluid within the pipe will affect the pipe clamp or attachment to the pipe. The material of the pipe will be specified to suit its operating temperature and this may also dictate the material of the pipe clamp regardless of the actual design temperature.

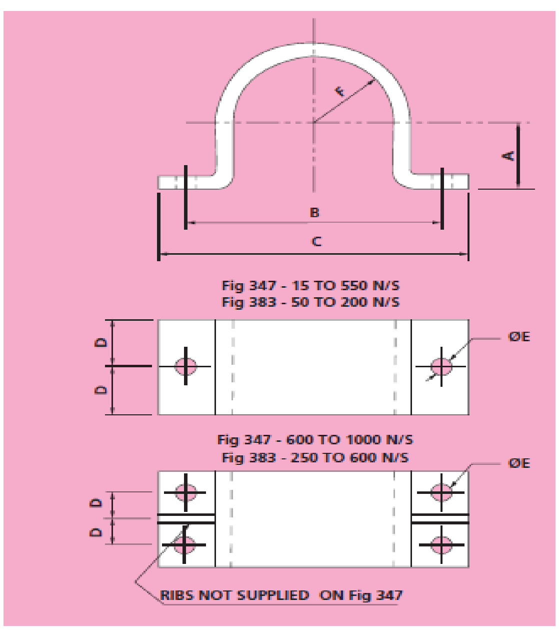
Considering the above factors and the structural limitations present on site, **Hold Down Clamp** was finalized to be used in all the three loading conditions.

Bergen Pipe Supports, Product Catalogue-Metric [6]:

Once the type of ancillary was fixed, data for design parameters was collected. In our case the internal diameter of the pipe is 508mm, keeping a clearance of 5mm as per the standard thumb rule the I.D of clamp is fixed at 513 mm. For the given pipe size, the clamp dimensions are:

Table 2.1: Design Parameters for hold down clamp from Bergen Pipe Supports Catalogue.

PART No.	PIPE SIZE	PIPE O/D	A	B	C	D	ØE	F	STEEL SIZE
F347-500	500	508.0	258	640	710	30.0	28	256	60×15
F347-550	550	559.0	278	690	760	30.0	28	282	60×15



Material: Carbon Steel

As the required pipe size is not specified, design parameters close to the ones presented above were taken.

Fig2.10: Design Parameters for hold down clamp from Bergen Pipe Supports Catalogue.

Cooper Industries, pipe supports [7]:

Here also the clamp parameters are based upon the diameter of the pipe size. Given that our O.D is 508mm, the following data was taken from the data sheet.

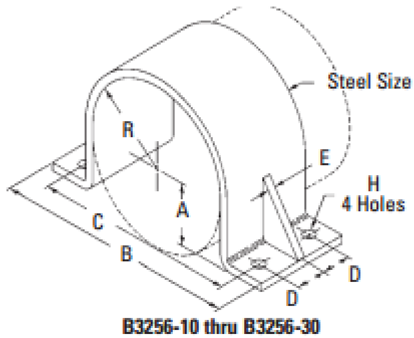


Fig2.11: Design Parameters for hold down clamp from Cooper Industries, pipe supports.

Table 2.2: Design Parameters for hold down clamp from Cooper Industries, pipe supports.

Part No.	Pipe Size		A		B		C		D		E	
	in.	(mm)	in.	(mm)	in.	(mm)	in.	(mm)	in.	(mm)	in.	(mm)
B3256-20	20"	(500)	9 3/4"	(247.6)	28"	(711.2)	24 1/2"	(622.3)	3 1/2"	(88.9)	1/2"	(12.7)
B3256-24	24"	(600)	11 3/4"	(298.4)	32"	(812.8)	28 1/2"	(723.9)	3 1/2"	(88.9)	1/2"	(12.7)

As the required pipe size is not specified, design parameters close to the ones presented above were taken.

Carpenter & Paterson Ltd [8]:

The clamp parameters here are given with respect to the O.D of pipe as well as the maximum loading conditions. Taking similar nomenclature for clamp parameters as above, the data is as follows:

Table 2.3: Design Parameters for hold down clamp from Carpenter & Paterson Ltd.

Pipe Size(mm)	A(mm)	B(mm)	C(mm)	D(mm)	E(mm)	Max Radial Load	Max Lateral Load
508	256	760	670	90	14	27KN	9.5KN

More weightage is given to these set of parameters as they are exactly for pipe size of 508mm and maximum loading conditions are also specified.

ANSI/ MSS SP-58: Pipe Hangers & Supports: Materials, Design, Manufacture, Selection, Application and Installation. [9]

The allowable stresses for the pipe supports is taken from this standard.

The pipe material selected is ASTM A312 GRADE TP304. Now considering galvanic corrosion between stainless steel, the clamp material is also selected as stainless steel.

The plate material available for stainless steel support fabrication is ASTM A240 TP304.

As the clamp is made from bending the plate, the clamp material is **ASTM A240 TP304.**

The **allowable stress** according to the above code for this material is **13.3 ksi or 91.7MPa.**

All the other material properties taken from the standard are as follows:

Coefficient of Thermal Expansion: 1.545E-05 C^-1

Young's Modulus: 1.938E+05

Tensile Yield Strength: 205 MPa

Compressive Yield Strength: 515 MPa

CHAPTER 3

Design and Stress Analysis of Pipe

3.1 Input parameters for pipe design

The pipe design needs to be carried out for the pipe connecting the condensate storage tank and the demineralized water storage tank. The internal gauge pressure is at 34.46 kg/cm2g and the design temperature is 27`c. The corrosion allowance is 1.2 mm. [4]

3.2. Design Procedure

Initially thickness is calculated based on internal gauge pressure and then it is verified by different methods for different conditions.

Once the thickness is calculated stress analysis is further carried on.

The ambient conditions play a major role in determining the pipe thickness. Some of them are:

1. Cooling Effects on Pressure: because of excessive cooling vacuum can be created, thus pipe should be able to withstand the excessive pressure.
2. Fluid Expansion Effects
3. Atmospheric Icing (when design minimum temp is below 0`c)
4. Low Ambient Temperature

Various methods used for verification of the pipe thickness are as follows.

1. External pressure verification
2. Underground thickness calculation
3. Thread check
4. Bend check
5. Hydro test calculation
6. Indian boiler regulation (IBR)

Internal Pressure:

By thumb rule we will take Design Pressure= 1.3* Internal Pressure.

P= 1.3*38.46= 50 kg/cm2g

3.3 Thickness Calculation

All the formulas and calculations are based upon the ASME code 31.1. [10]

The required thickness of straight sections of pipe shall be determined in accordance with following equation.

tm = t + c

$$\mathbf{t = \frac{PD}{2(SEW+PY)}}$$

STEP 1:

• For the above temperature find the allowable stress value from table A-1 in ASME B31.1 [10]

STEP 2:

• Now from Table 302.3.5 in ASME B31.1 based on temp and material select the weld joint strength reduction factor W.

• Here for given parameters W = 1

STEP 3:

• Now from table A1-A or A1-B in ASME B31.1 select the weld quality factor E on the basis of material specification.

• Here for given material specification E = 1

STEP 4:

• Now from Table 304.1.1 in ASME B31.1 select the Y coefficient based on material and temperature.

• Here for given case Y = 0.4

STEP 5:

• Now put all the selected values given in equation: $t = \frac{PD}{2(SEW+PY)}$

STEP 6:

• Now add the corrosion allowance and milling tolerance in the calculated to get tm

C.A. = 1.2 mm (As per given in PMS), and milling tolerance is 12.5%

• We get the thickness of pipe at 38`c as seen below:

Some values which are constant for all diameters are:

E=1 (quality factor) Y=0.4 (coefficient) W= 1 (Weld Joint factor)

C1= 1.2(corrosion allowance)   C2=0 (thread allowance)   C3= mill tolerance= 12.5%

Overall c= c1+c2+c3

STEP 7:

• Now we will select the standard thickness with respect to the total thickness found out.

• Usually the procured thickness is very high with respect to the standard thickness because of various reasons. Some of these reasons are discussed later in the report

Table 3.1: Thickness calculation of Pipe

Diameter	O.D	Design Pressure	Allowable Stress	tm	c1	c3	c	T	Available thicknes	AT-c3	MAX ALLOWED PRESSURE BY REVERSING THE FORMULA
NPS(in)	mm	kg/mm2g	kg/mm2	tm= PD/2(SEW+PY)			c= c1+c3	t+c			
0.5	21.3	0.5	8.01499	0.648205293	1.2	0.081	1.281025662	1.93	2.11	2.03	2.899879015
6	168	0.5	8.01499	5.12173478	1.2	0.6402	1.840216847	6.96	7.11	6.47	1.115397457
24	610	0.5	8.01499	18.56362576	1.2	2.3205	3.52045322	22.1	22.83	20.5	0.971686141
42	1067	0.5	8.01499	32.471129	1.2	4.0589	5.258891125	37.7	Custom Made	33.7	0.910450295
56	1422	0.5	8.01499	43.27455055	1.2	5.4093	6.609318818	49.9	Custom Made	44.5	0.902141974
64	1626	0.5	8.01499	49.48271392	1.2	6.1853	7.38533924	56.9	Custom Made	50.7	0.899009979
All calculations are made for material ASTM A106 B											
The design temp		427`c									
c= c1+2+c3			E=1								
c1 (corrosion allowance)= 1.2			w=1								
c2(thread allowance)=0			y=0.4								
c3(mill tolerance) = 12.5%											

• We can see that the difference between procured thickness and the total thickness increases with increase in diameter. This is because with increase in diameter the stress caused due to various effects increases significantly.

3.4 Various Cross check for finalizing the thickness of pipe.

Some factors which affects the thickness are:

o External pressure verification

o Bend check

o Hydro test calculation

o Underground thickness calculation

o Thread check

o Indian boiler regulation (IBR)

3.4.1 Bend Check

The minimum required thickness tm of a bend, after bending, in its finished form, shall be determined in accordance with following equations.

$$t = \frac{PD}{2((SEW)/I+PY)}$$

Where at the intrados (inside bend radius)

$$I = \frac{4\left(\frac{R1}{D}\right) - 1}{4\left(\frac{R1}{D}\right) - 2}$$

And at the extrados (outside bend radius)

$$I = \frac{4\left(\frac{R1}{D}\right) + 1}{4\left(\frac{R1}{D}\right) + 2}$$

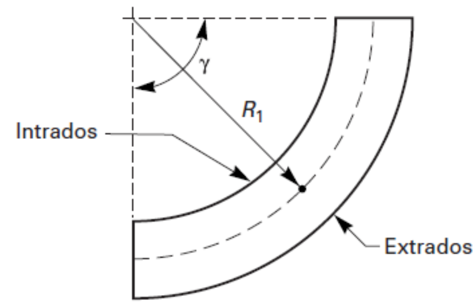


Fig 3.1: Intrados and Extrados

At the sidewall on the bend centerline radius, I = 1.0, and where R1 = bend radius of welding elbow or pipe bend.

Thickness variations from the intrados to the extrados and along the length of the bend shall be gradual. When pipes are bent on site to achieve curve, it becomes mandatory to check whether the pipe will be able to sustain the pressure or not because of variation in thickness at intrados and extrados of the pipe.

Hence thickness at intrados and extrados is calculated as shown below for different diameters of the pipes and for the different curve radius as per the equation shown above.

For all the conditions procured thickness after removing allowance at the intrados and extrados is greater than the actual design thickness. Hence pipes will be able to handle the design internal pressure. Here, pipe of thickness 6’’ (168.3mm O.D) is checked.

Table 3.2: Bend Check Verification

SR.NO	O.D (mm)	Bend Radius	Bend Radius (mm)	Intrados (mm)	Extrados (mm)	tm intrados	tm extrados
1	168.3	1D	168.3	1.5	0.833	7.590208	5.283822
2	168.3	3D	504.9	1.1	0.928571	5.620225	5.364179
3	168.3	6D	1009.8	1.045455	0.961538	5.348624	5.229358

- If we want to compare it with standard thickness, we will also have to add the various tolerances to the respective intrados and extrados thickness.
- The tolerance ‘C’ can be obtained from previous data.
- Intrados thickness will always be more than extrados thickness.
- From the above results we can see that even though the thickness is decreasing at extrados it is greater than the minimum required thickness and therefore the design is safe.

3.4.2 Leak test or Hydro test

Prior to initial operation, each piping system shall be leak tested. Each weld and each piping component, except bolting and individual gaskets to be used during final system assembly and pressure-relieving devices to be used during operation, shall be hydrostatically or pneumatically leak tested.

The hydrostatic test pressure at every point in a metallic piping system shall be not less than 1.25 times the design pressure as per the ASME B31.1 [10].

P= $\frac{2SEWt}{\mathrm{D-2tY}}$

Table 3.3: Leak Test

SR.NO	DIA NPS(in)	OD mm	T-C3 mm	Max Allowed Pressure (Kg/mm2g)	Max Allowed Pressure (Kg/cm2g)		
1	0.5	21.3	2.028974338	2.899879015	289.9879015		
2	6	168.3	6.469783153	1.115397457	111.5397457		
3	24	610	20.50954678	0.971686141	97.16861408		
4	42	1067	33.671129	0.910450295	91.04502948		
5	56	1422	44.47455055	0.902141974	90.21419745		
6	64	1626	50.68271392	0.899009979	89.90099789		
Material is A106B				Hydro Test Pressure= 1.5*P = 1.5*50 = 75 kg/cm2g = Pt = Test Pressure			

As Pt or test pressure is below allowable pressure the pipe is safe.

3.4.3 External Pressure Check

This check is for pipes when it is to be used below ground level. This is done to ensure that the pipe will withstand the increased external force acting on it. The allowable pressure is calculated based upon various experimental tabled laid down in ASME B 31.1. [10]

As external pressure check is not required for the current pipe configuration, another problem case parameters are used to demonstrate the check.

Temperature = 38°C Pressure = 17.7bar Material = A106(C)

C1 (corrosion allowance) = 1.5mm C2 (thread allowance) = 0mm C3 (mill allowance) = 0.5mm

Value of A and B taken from Table G, ASME Section 2

Table 3.4: External pressure check

Sr. No.	Dia meter (mm)	Procured Thickness without allowance (mm)	L/Do	Do/t	A	B	Allowable Pressure (bar)	Design Pressure (bar)
1.	60.3	1.73	50	34.85549	0.00661	17248.50	45.49229577	17.7
2.	114.3	3.08	50	37.11039	0.00401	17002.99	42.11992579	17.7
3.	168.3	5.11	50	32.93542	0.00882	17314.80	48.32949112	17.7

As the Allowable pressure is greater than the design pressure the pipe is safe under external loading.

3.4.4 Thread Check

When threads are required on the pipe, pipe should be designed accordingly. The height of the threads should be added to the designed thickness and then pipe specifications should be given for the procurement. Here we have followed the reverse process.

We can obtain the standard pitch for the given diameter of the pipe from ANSI B1.20.1.

Relation between height of the thread and pitch can be given as

$h=0.866025p$

If thickness after removing allowances and thread height from the procured thickness is higher than the designed thickness, then threads are allowed to be produced on that pipe, otherwise threads should be avoided.

Again, as thread check is not required for the current pipe configuration, another problem case parameters are used to demonstrate the check.

Sample calculations for thread check are shown below.

Extra thickness left = Procured thickness – h – Original thickness

Note: all the above thickness are without any allowances

- Here for 1.5’’and 6” dia. pipe thickness after removing allowances and thread height from procured pipe thickness is higher than the designed thickness. Hence threads are allowed to be produced on the pipe.
- But for pipe diameters 2’’in pipe thickness after removing allowances and thread height from procured pipe thickness is smaller than the designed thickness. Hence threads are not allowed to be produced on these pipes.

Table 3.5: Thread Check

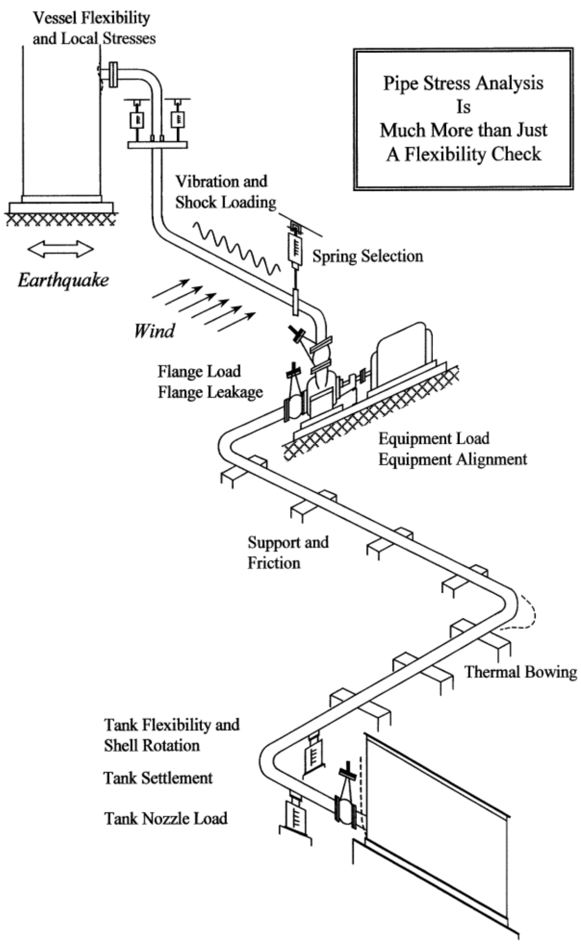
O.D (mm)	Thread per inch	Pitch	H=0.866025 P	Procured Thickness without allowance (mm)	Tm (mm)	Extra thickness
60.3	11.5	2.2086	1.9127	1.73	0.469482255	-0.652267
114.3	8	3.175	2.7496	3.08	1.310345996	1.05002
168.3	8	3.175	2.7496	5.11	0.376052951	1.29116

3.5 Stress Analysis

Pipe stress Engineers calculate the stress in a piping system subject to normal operating loads such as pressure, weight, and thermal expansion, and occasional loads such as wind, earthquake, and water hammer. All piping systems are connected to equipment such as vessels, tanks, pumps, turbines, and compressors; the piping stress analysis also involves evaluation of the effect of the piping forces and moments to the connecting equipment.

As the piping stress is controlled by the arrangement of the supports and restraints, the scope of piping stress includes also pipe supports. The whole scope of this work is generally referred to as piping mechanical.

3.5.1 Reasons for carrying out stress analysis



1. In order to keep stresses in the pipe and fittings within code allowable levels.
2. In order to keep nozzle loadings on attached equipment within allowable of manufacturers or recognized standards (NEMA SM23, API 610, API 617, etc.)
3. In order to keep vessel stresses at piping connections within ASME Section VIII allowable levels.
4. In order to calculate design loads for sizing supports and restraints.
5. In order to determine piping displacements for interference checks.
6. In order to solve dynamic problems in piping, such as those due to mechanical vibration, acoustic vibration, fluid hammer, pulsation, transient flow, and relief valve discharge.
7. In order to calculate and check Flange Leakages.
8. In order to help optimize piping design.

Fig 3.2: Piping System

3.5.2 Using Caesar II

CAESAR II is a FEA pipe stress analysis software program developed, marketed and sold by COADE Engineering Software. This software package is an engineering tool used in the mechanical design and analysis of piping systems. The CAESAR II user creates a model of the piping system using simple beam elements and defines the loading conditions imposed on the system. With this input, CAESAR II produces results in the form of displacements, loads, and stresses throughout the system. Additionally, CAESAR II compares these results to limits specified by recognized codes and standards.

.

Let’s now go through a step by step procedure of calculating the stresses:

1. First, we create the model of the piping system from the Piping isometric. Here as the actual isometric cannot be shown because of confidentiality reasons, an example problem has been used.
2. The inputs are then entered. The required inputs are:
   - Material
   - Design temperature and pressure
   - Working temperature and pressure.
   - Thickness of pipe
   - Density of fluid.
   - Insulation
   - Allowances (corrosion, mill, thread, etc.)
   - Nozzle Displacement if any

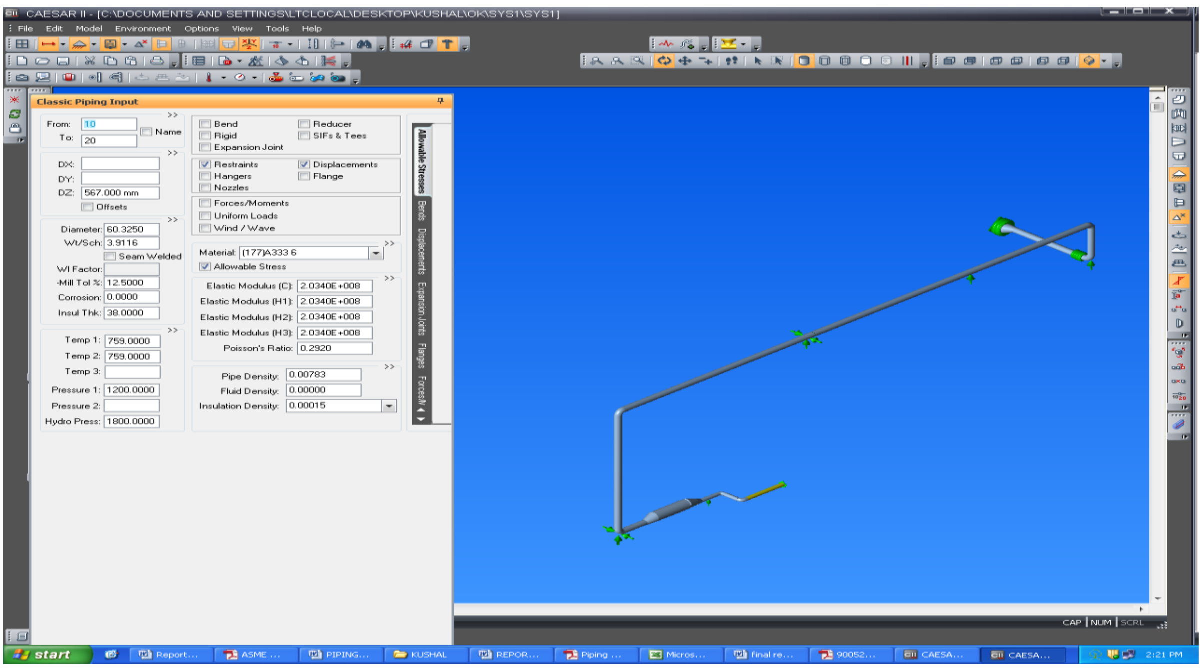


Fig 3.3: Model made from isometric in Caesar II

3. Once the model is ready, we check for errors and proceed with the analysis by generating the report.
4. Various reports can be generated based upon the required conditions.

5. Usually we check for three loading conditions
   - Sustained
   - Operating
   - Hydro, expansion and occasional

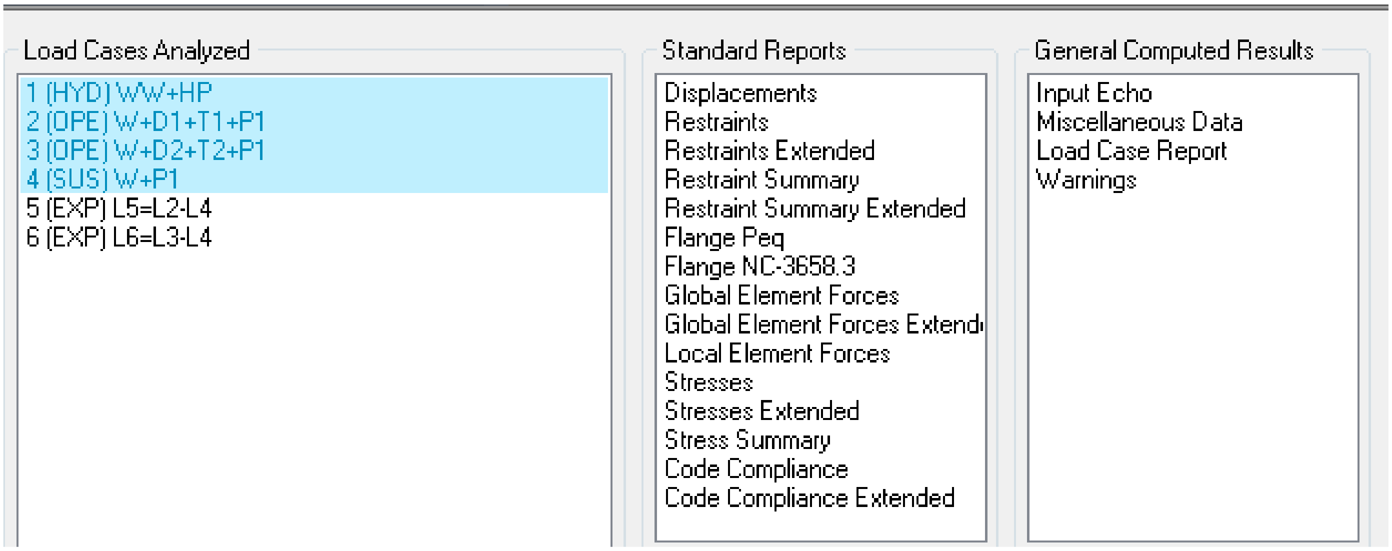


Fig 3.4: Load cases to be given to the model.

6. We first check for sustained loading conditions. In this we check for

- Displacement in the pipes
- Stresses generated due to sustained loads, mainly sagging.
- A minimum sagging of 5mm is usually tolerable.
- We can see that the allowable stress is greater than code stress the by a ratio of 6.1(code stress ratio), therefore the pipe is safe.

```
Highest Stresses: (   KPa    ) LOADCASE 4 (SUS) W+P1
CodeStress Ratio (%):        6.1  @Node    240
Code Stress:              8465.2  Allowable:   137895.1
Axial Stress:             6216.2  @Node     90
Bending Stress:           4496.3  @Node    240
Torsion Stress:            187.4  @Node     20
Hoop Stress:             12996.4  @Node     80
3D Max Intensity:        14822.9  @Node    360
```

Fig 16: Output for sustained loading condition.

7. Then we generate the report for operating condition.

• The stresses at support and nozzles are checked in this condition.

• The stresses generated are compared with allowable stresses given in the support and nozzle load standards.

```
 NO CODE STRESS CHECK PROCESSED: LOADCASE 2 (OPE) W+D1+T1+P1

Highest Stresses: (    KPa     ) LOADCASE 2 (OPE) W+D1+T1+P1
OPE Stress Ratio (%):         0.0  @Node    260
OPE  Stress:             293078.7  Allowable:        0.0
Axial Stress:              6268.8  @Node     80
Bending Stress:          290245.6  @Node    260
Torsion Stress:           63101.1  @Node    310
Hoop Stress:              12996.4  @Node     80
3D Max Intensity:        297098.3  @Node    260

 NO CODE STRESS CHECK PROCESSED: LOADCASE 3 (OPE) W+D2+T2+P1

Highest Stresses: (    KPa     ) LOADCASE 3 (OPE) W+D2+T2+P1
OPE Stress Ratio (%):         0.0  @Node    258
OPE  Stress:              78002.4  Allowable:        0.0
Axial Stress:              6237.2  @Node    170
Bending Stress:           73959.7  @Node    258
Torsion Stress:           14751.6  @Node    310
Hoop Stress:              12996.4  @Node     80
3D Max Intensity:         78403.6  @Node    210
```

Fig 3.5: Output for operating condition. There is an option to input multiple input temperature to compare multiple results simultaneously.

8. Finally we check for Hydro, Expansion and occasional loading

Again the stresses are checked so that they are below the allowable stresses. For Hydro load we will only consider the hydrostatic pressure and force due to weight of pipe.

We can see that the code stress is less than the allowable stress by a factor of 4.9(code stress ratio), thus the pipe is safe under this condition.

9. If in any of the above condition the stresses exceed the allowable stresses, then parameters are adjusted to make the system safe. Usually the supports are adjusted to bring down the stresses under the limit.

Stress Summary

```
CAESAR II Ver.5.20.3,  (Build 100715)   Date: JUN 21, 2016    Time: 15:18
Job: C:\DOCUMENTS AND SETTINGS\LTCLOCAL\DESKTOP\KUSHAL\OK...\SYS1
Licensed To: L&T CHIYODA LIMITED   --  ID #34882
STRESS SUMMARY REPORT: Highest Stresses Mini Statement
Various Load Cases

LOAD CASE DEFINITION KEY

Piping Code: B31.3      = B31.3 -2006, May 31, 2007


 CODE STRESS CHECK PASSED      : LOADCASE 1 (HYD) WW+HP

Highest Stresses: (   KPa    ) LOADCASE 1 (HYD) WW+HP
CodeStress Ratio (%):        4.9  @Node    240
Code Stress:             11806.3  Allowable:   241316.5
Axial Stress:             9322.8  @Node     90
Bending Stress:           5849.2  @Node    240
Torsion Stress:            222.5  @Node     20
Hoop Stress:             19494.6  @Node     80
3D Max Intensity:        22234.4  @Node    360
```

Fig 3.6: Output for hydro test

3.6 Brief Summarization of Pipe Design

- Initially the pipe thickness was calculated for the given input conditions.
- The procured thickness is then calculate by performing various checks as needed.
- Finally closest available thickness is taken as the procured thickness.
- The procured thickness for different diameter sizes are than send to the client for approval.
- The piping system which is developed by the layout department is modelled in Caesar II.
- Stresses are then checked for various loading conditions.
- If the allowable stresses are higher than the current stresses the system is safe.
- If not, the type of pipe supports used or the location of the various supports are changed to meet the allowable stresses.

CHAPTER 4

Design and Stress Analysis of Clamp

4.1 Input parameters for clamp design

The strap down pipe clamp is basically used to hold the pipe and transmit the forces of the pipe to the ground.

It is mostly used when the pipes are laid down on ground level. Here, the clamps will be used to hold down the pipe connecting the condensate storage tank and the DM water tank. This is the same pipe for which the design analysis was carried for above.

The geometry design and modeling has been done in Autodesk fusion and the stress analysis is carried out in Ansys Workbench.

The reaction forces on the clamps have been calculated in Caesar II by considering various static and dynamic loading conditions.

The forces are:

CASE 1: Fx= 1343 N Fy= 26213 N Fz= 31495 N

CASE 2: Fx= 39596 N Fy= 15543 N Fz= 18675

CASE 3: Fx= 1169 N Fy= 22761 N Fz= 19700 N

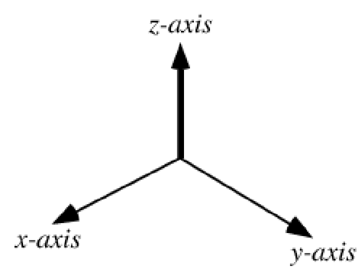


Fig 4.1 Axis Representation

4.2 Clamp materials

The clamp material is largely dependent on the material of the pipe. As stated above the clamp material selected is ASTM A240 TP304. The allowable stresses are 13.3 ksi. [9]

With the given materials and the assembly, the density and volume of the clamp assembly is as follows:

Volume	6.0744e+006 mm^3
Mass	48.777 kg

All the analysis is done at a design temperature of 35`C.

	A	B	C	D	E
1	Contents of Engineering Data			Source	Description
2	Material				
3	A240				
4	SS AISI 202				
5	~~Stainless Steel~~			G	
6	~~Structural Steel~~			G	Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5 -110.1
*	Click here to add a new material				

Properties of Outline Row 3: A240

	A	B	C	D	E
1	Property	Value	Unit		
2	Density	8.03	g cm^-3		
3	Isotropic Secant Coefficient of Thermal Expansion				
4	Coefficient of Thermal Expansion	1.545E-05	C^-1		
5	Zero-Thermal-Strain Reference Temperature	22	C		
6	Isotropic Elasticity				
7	Derive from	Young's Modulu...			
8	Young's Modulus	1.938E+05	MPa		
9	Poisson's Ratio	0.3			
10	Bulk Modulus	1.615E+11	Pa		
11	Shear Modulus	7.4538E+10	Pa		
12	Tensile Yield Strength	205	MPa		
13	Compressive Yield Strength	515	MPa		

Fig4.2: Material definition of ASTM A240 TP304 in ANSYS.

- The material for nuts and bolts is taken as Stainless Steel AISI 202, as per the standard conventions.

Properties of Outline Row 4: SS AISI 202

	A	B	C	D	E
1	Property	Value	Unit		
2	Density	7.855	g cm^-3		
3	Isotropic Secant Coefficient of Thermal Expansion				
4	Coefficient of Thermal Expansion	1.2402E-05	C^-1		
5	Zero-Thermal-Strain Reference Temperature	22	C		
6	Isotropic Elasticity				
7	Derive from	Young's Modulu...			
8	Young's Modulus	2.0477E+05	MPa		
9	Poisson's Ratio	0.29			
10	Bulk Modulus	1.6252E+11	Pa		
11	Shear Modulus	7.9369E+10	Pa		
12	Tensile Yield Strength	412.3	MPa		
13	Compressive Yield Strength	667.41	MPa		

Fig4.3: Material definition for Stainless Steel AISI 202 in Ansys.

4.3 Need for Clamp Design

The design parameters here are taken in reference to the various standard catalogues [2] [3] [4] available for the hold down type clamp. After comparing the parameters across different sources and taking into account the maximum loads given in Carpenter & Paterson Ltd [4], we make a standard model of our clamp. The problem with this model though is that the loading condition required are far greater than the maximum loads given in the standards. Therefore a new clamp is designed using the existing data so as to withstand the different loading condition.

To get an idea of the increase in strength required, let us now compare the forces of loading conditions to the maximum allowable forces as mentioned in the standard.

Table 4.1: Force comparison between actual Reaction forces and Standard allowable forces.

Force Direction	Fx(Axial) KN	Fy (Lateral) KN	Fz (Radial) KN
Allowable Forces	0	9.5	27
Case 1	1.343	26.213	31.495
Case 2	29.596	15.543	18.675
Case 3	1.169	22.761	19.7

From the above data we can easily conclude that a need for designing a customized highly robust clamp is needed.

4.4 Clamp Design for condition 1 loading.

In condition 1 the Fy load is on line element as opposed to area loading in condition 2. Also in the latter stage pressure loading is given to get a more uniform force distribution.

The general parameters taken are as follows:

Table4.2: Design Parameters for customized hold down clamp

Pipe Size (mm)	A (mm)	B (mm)	C(mm)	D (mm)	E (mm)	R (mm)
508	256.5	749.3	656.15	130	Variable	256.5

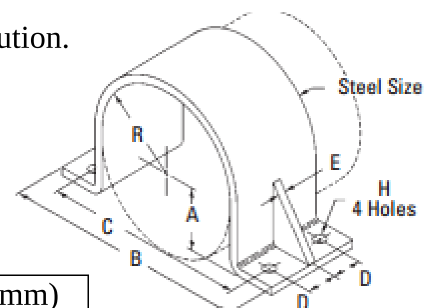


Fig 4.4: Clamp Model

1) Thickness of clamp = 14mm.

To start the designing, the initial clamp thickness is taken as 14mm.

- Ribs are designed on to the clamp so that the clamp can be designed with the least thickness dimension.
- With t=14mm, OD = 541 mm

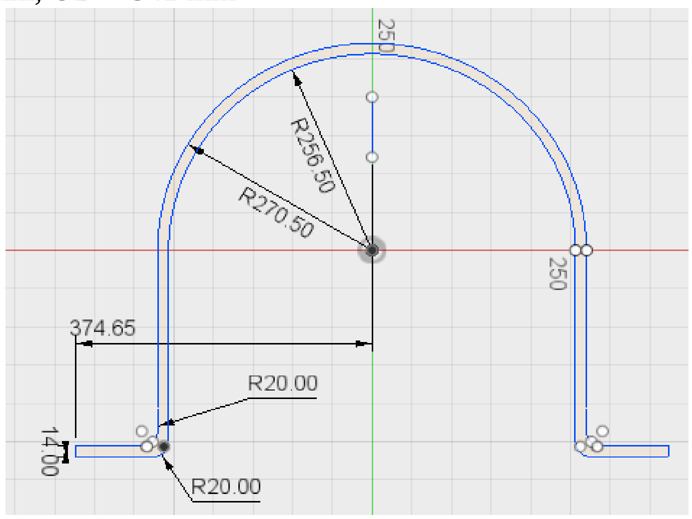
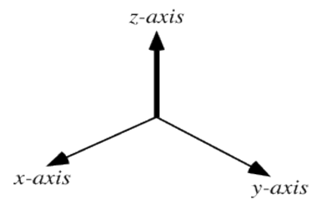


Fig 4.5: Dimensions of clamp when t=14mm



Case 1: Fx= 1343 N Fy= 26213 N Fz= 31495 N

- The forces are given on point, line or surface of the model depending upon the requirement.
- The Fz force is acted by the pipe on the lower surface of the top side of the clamp or the curvature of the clamp.
- The Fy is acted by the pipe of the edge of the clamp where the curvature ends.
- The Fx or axial force is also acting on the curvature of the clamp.
- Notice that here the force values are given for loading as opposed to pressure loading in condition 2.

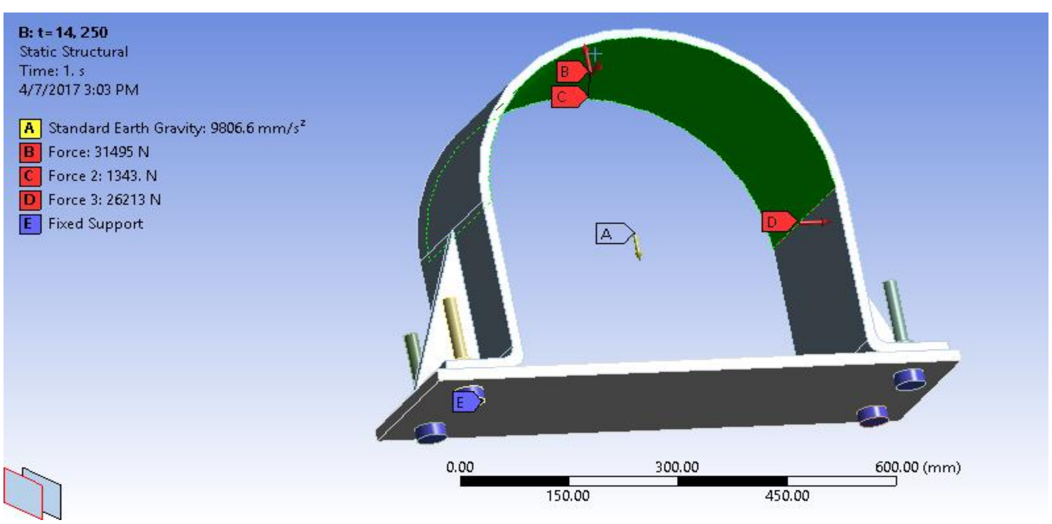


Fig 4.6: Forces on the Clamp.

- The meshing parameters are as follows:

Details of "Mesh"

Defaults	
Physics Preference	Mechanical
Relevance	100
Sizing	
Use Advanced Size Function	On: Proximity and Curva...
Relevance Center	Fine
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Fine
Curvature Normal Angle	Default (12.0 °)
Proximity Accuracy	0.5
Num Cells Across Gap	Default (5)
Min Size	Default (9.4894e-002 mm)
Proximity Min Size	Default (9.4894e-002 mm)
Max Face Size	Default (9.48940 mm)
Max Size	Default (18.9790 mm)
Growth Rate	Default (1.50)

Fig4.7: Meshing Parameters

On evaluating the results the maximum stress are developing on the supporting ribs. Especially the right rib which is resisting almost all of the Fy force. The max stress is of 20 ksi.

As the maximum stress is higher than the allowable stresses of 13.3 ksi, the thickness of the clamp is increased. As t=14 mm is failing for the first case, no analysis is carried out for the second and the third case and we move to t=16mm.

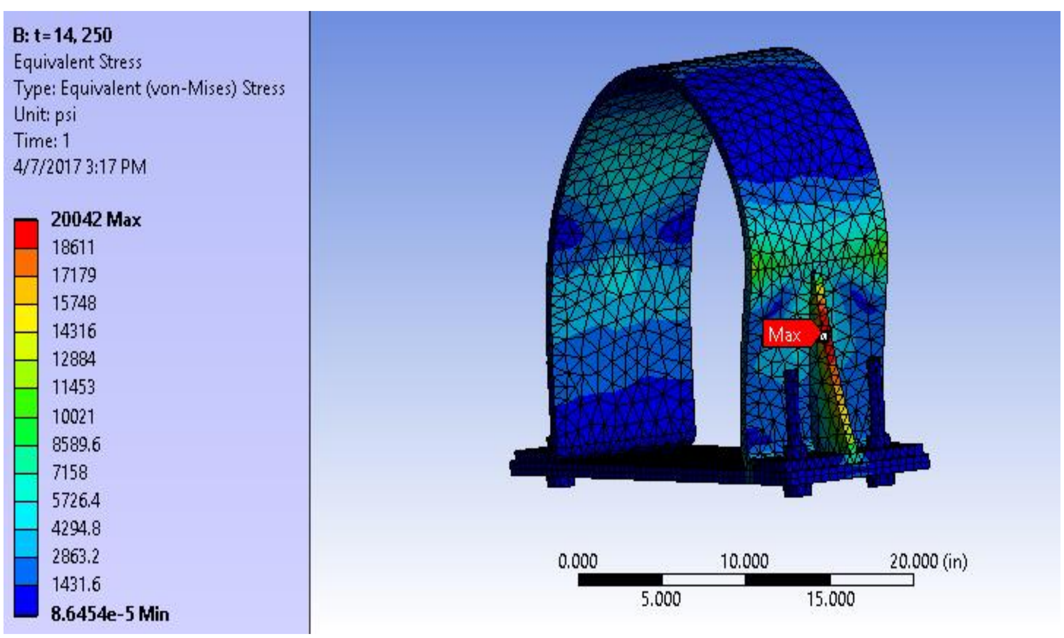


Fig 4.8: Stress Analysis for clamp at t=14mm

2) Thickness of clamp = 16mm.

- Here thickness =16mm for the plate.
- With considering all the previous data, the maximum stresses are again developing on the rib and they are 16.125 ksi.
- As this is also failing the criteria for allowable stresses at 13.3, we will again increase the thickness of the plate.
- Now thickness taken would be t=18mm.

3) **Thickness of clamp = 18mm.**

The thickness of clamp = 18mm. As we will see later on this thickness meets the allowable stress criteria and therefore we will study the geometry and stress data in detail.

a) Geometry:

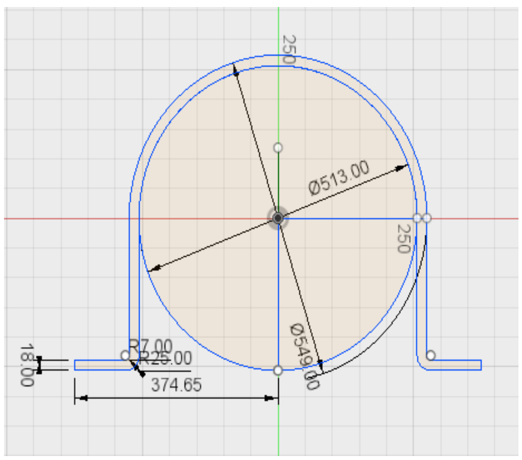
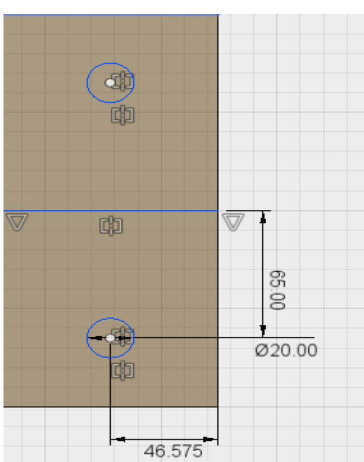


Fig 4.9: Front view of the clamp. The OD obtained is 549mm . The Fillet radius is taken as R =25 mm and r=7mm.



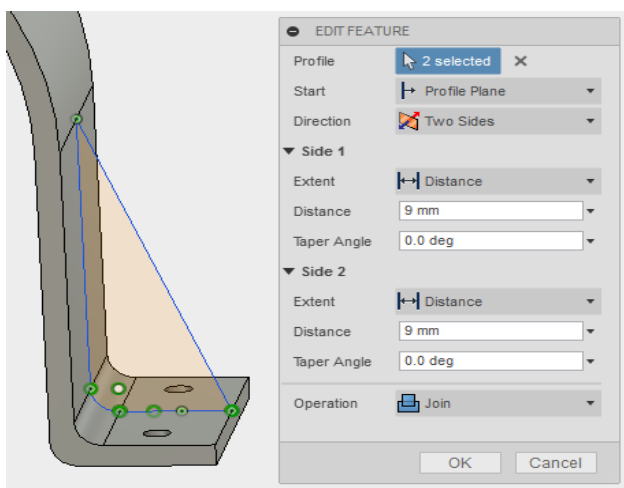


Fig4.10: Top view of the clamp. Fig 4.11: Front view of the clamp depicting the thickness of rib.

Bolt and Nut Design:

- Corresponding to radius of 20mm. [9]

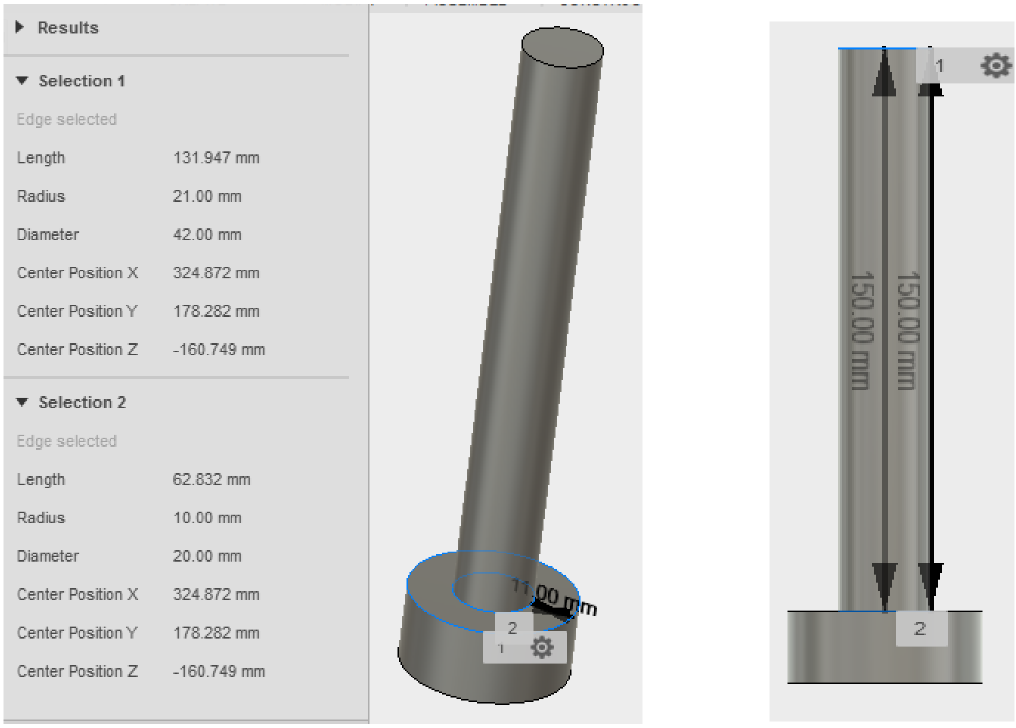


Fig4.12: Dimensions of the Bolt

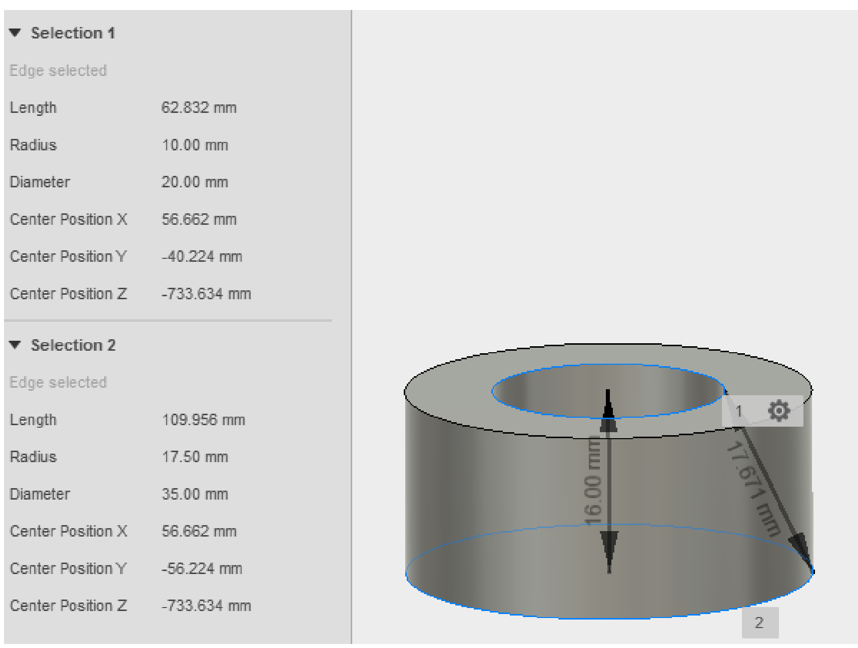


Fig4.13: Dimensions of the Nut

Bottom Plate Design

- The bottom plate is only used to the rest the pipe.
- The dimensions are therefore in correspondence to that of the clamp.
- A thickness of 16mm is given, so that there is some clearance between the pipe and the ground.
- The reaction forces found on the clamp have also considered the sagging effect between two clamps because of the increase in elevation.

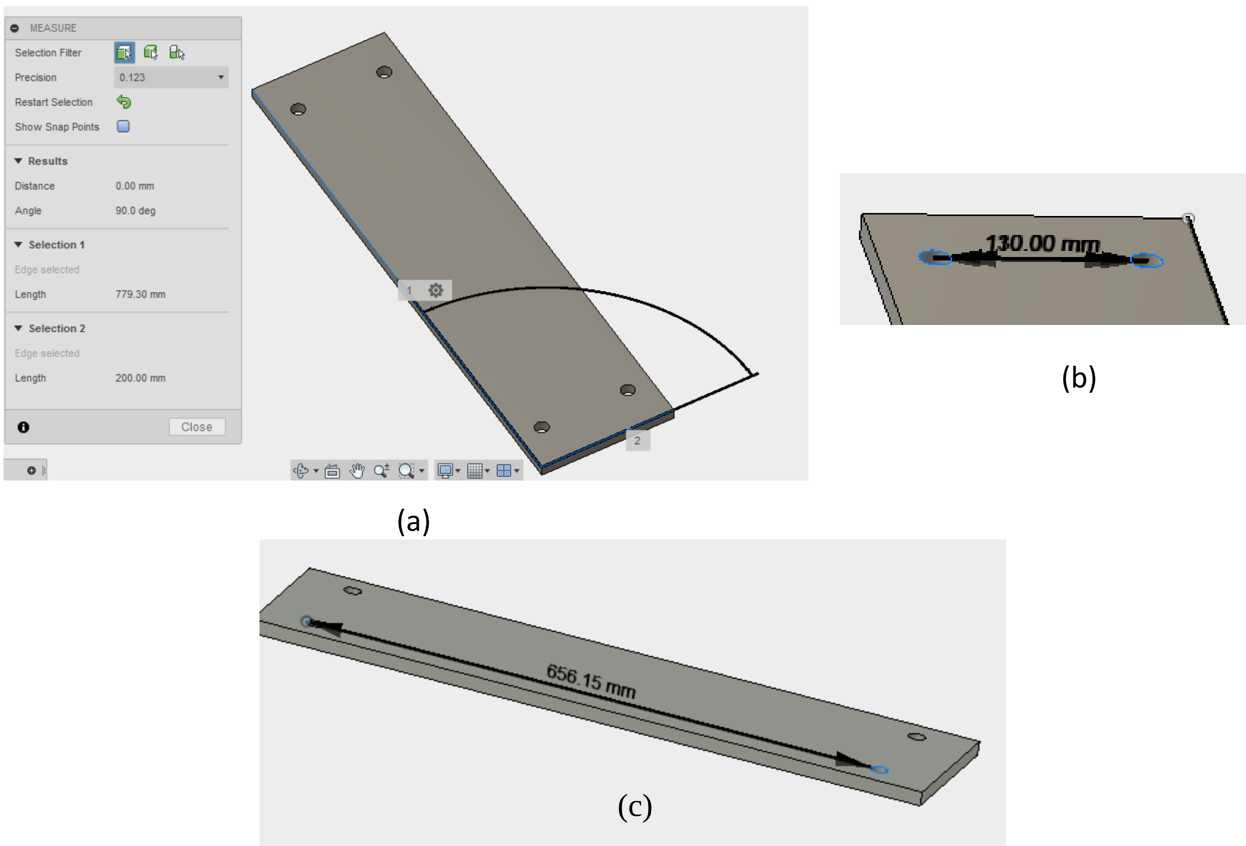
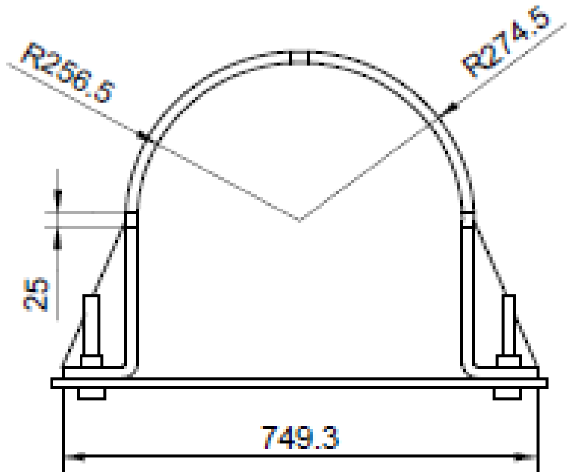


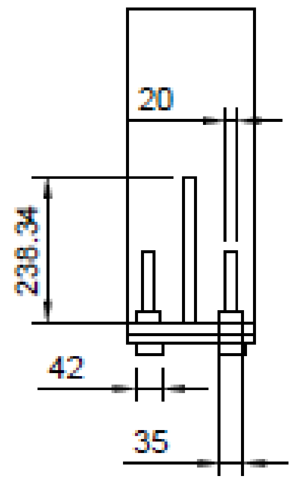
Fig 4.14: (a) Length and Breadth if the plate. (b) The distnce between two holes and the dia of the holes. (c) The longitudinal distance between the holes.

- The width or the depth of the clamp was initially kept at w=250 mm, but due to the required increased in thickness , the width of the clamp was reduced to w=200 mm to compensate for the increase in material required.
- A model drawing of the whole clamp assemnbly is shown below

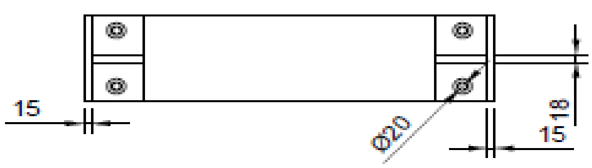
Overall Clamp Assembly

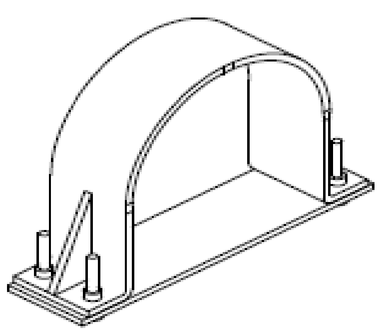


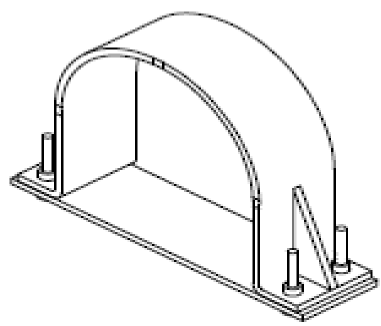
(a)



(b)







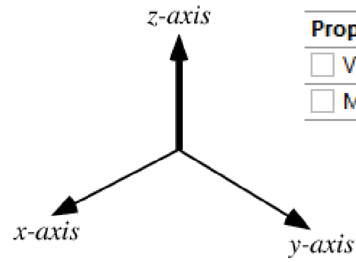
(d) (e)

Fig4.15: (a) Front View (b) Right hand side view (c) Top view (d) Isometric view (e) Isometric view

(b) Stress Analysis

CASE 1:

- Fx= 1343 N
- Fy= 26213 N
- Fz= 31495 N



Properties	
Volume	8.9448e+006 mm³
Mass	71.768 kg

- R= 25, r=7, l= 749.3, W=200

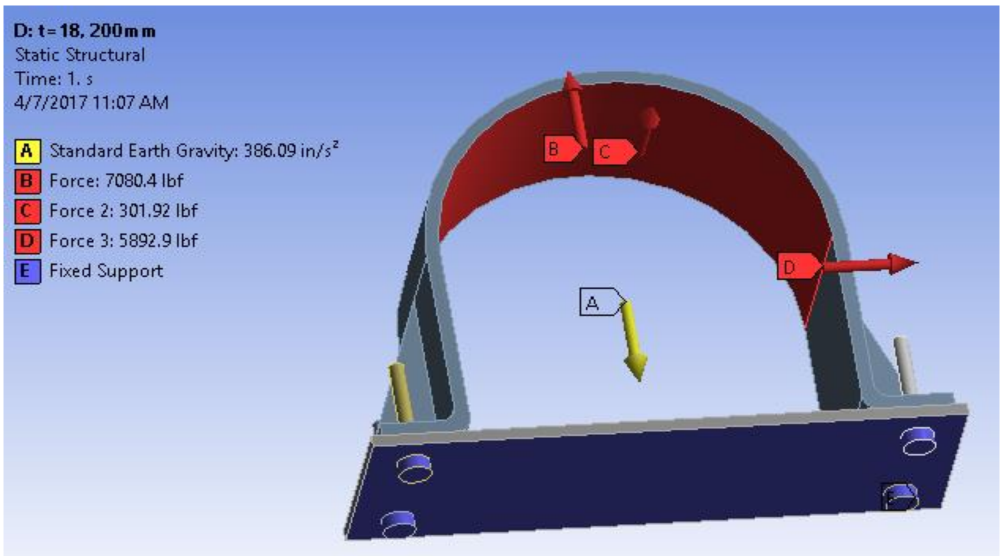


Fig 4.16: Force representation for case 1.

(a) Deformation:

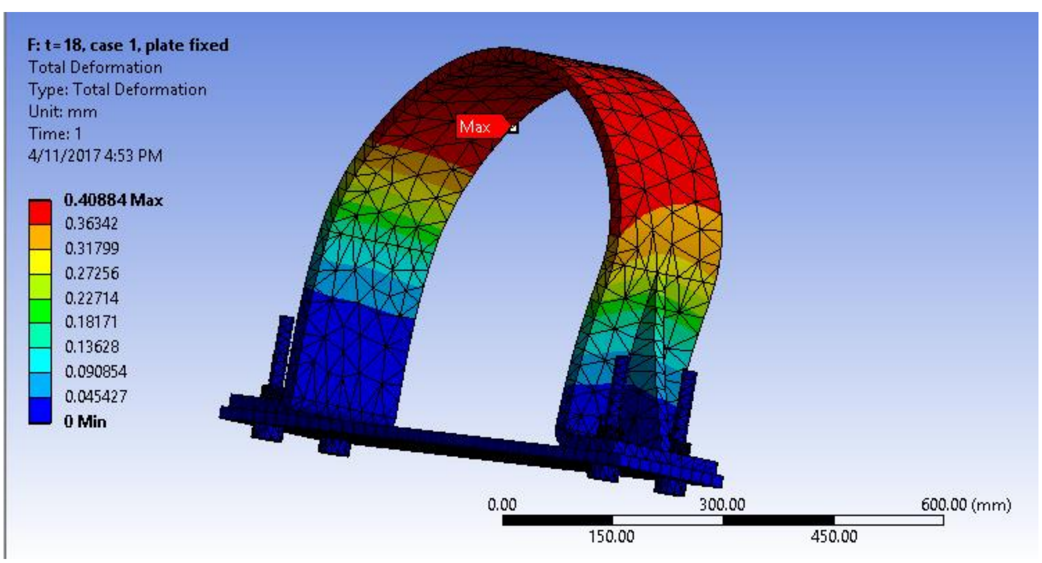


Fig 4.17: Maximum deformation for case 1

(b) Equivalent Stresses

- The maximum stresses are induced in the ribs.
- Here the value of maximum stresses is 13.033 ksi, but as this is below the allowable limit of 13.3 ksi, the clamp is safe.

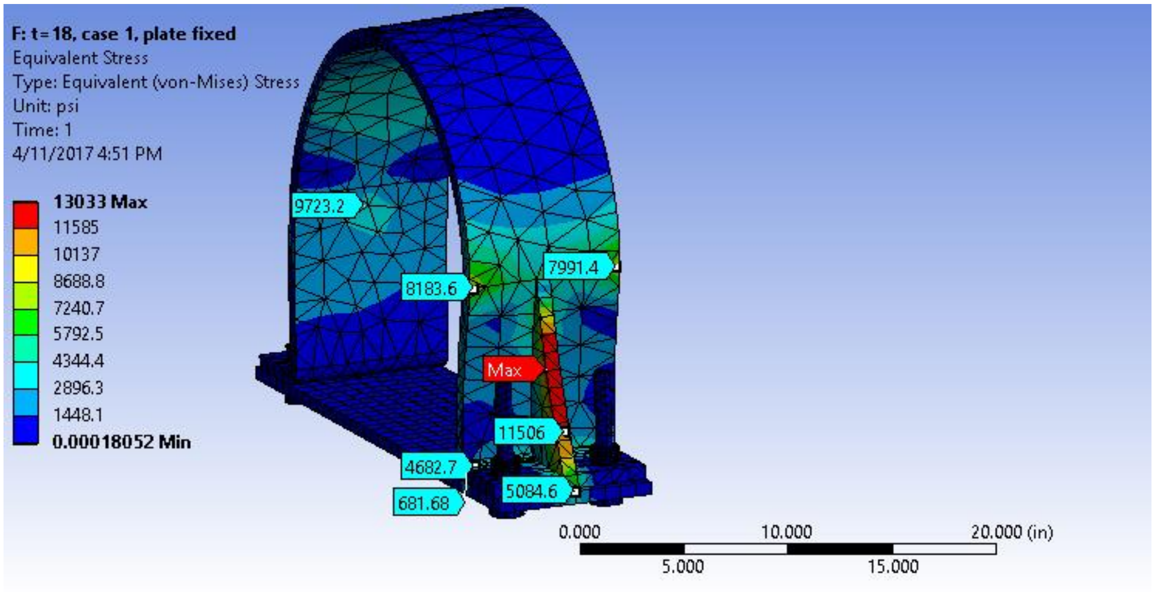
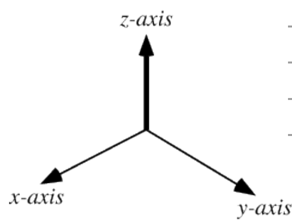


Fig 4.18: Equivalent Stresses for case 1

CASE 2:

- Fx= 39596 N
- Fy= 15543 N
- Fz= 18675 N



Properties	
Volume	8.9448e+006 mm³
Mass	71.768 kg

- In this case a specific requirement of axial stops were needed, as the axial force is very high. To perform analysis a rectangular face separation was created where the axial forces would act.

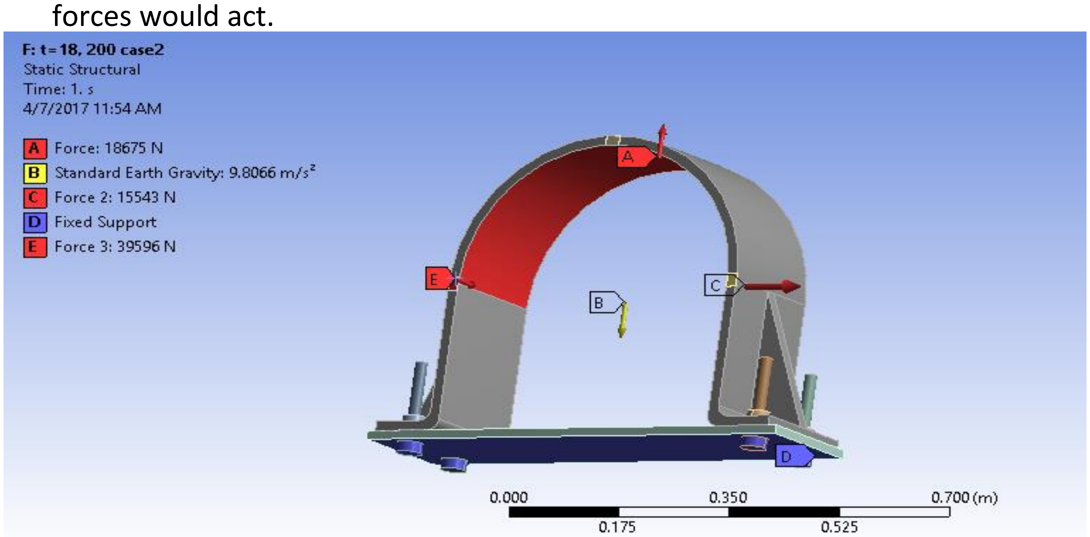


Fig 4.19: Force representation for case 2

(a) Deformation

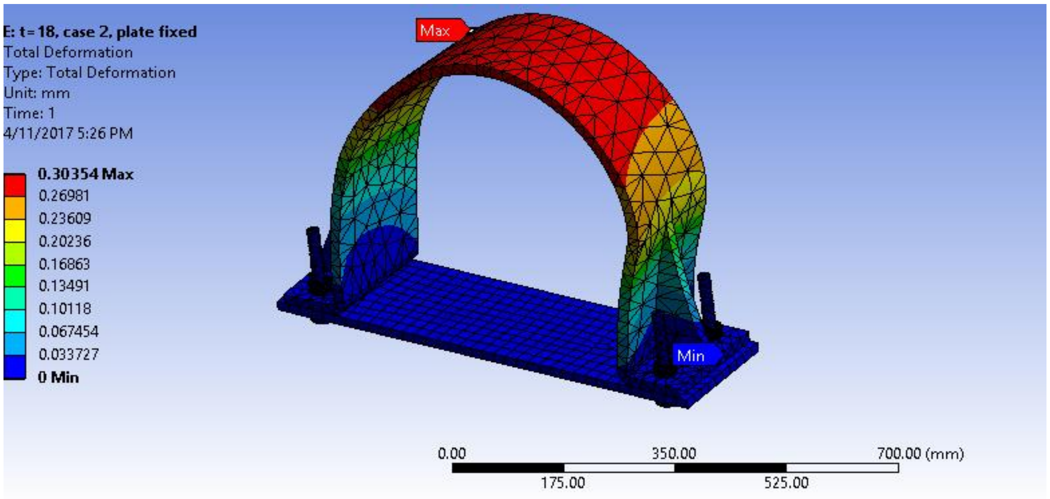


Fig 4.20: Maximum deformation for case 2

(b) Equivalent Stresses

- The maximum stresses are induced at the bend with a value of 13.311 ksi. As this is 0.011 ksi above the limit of 13.3 ksi, it is allowable.
- The stresses at the ribs for the right side and left side of the clamp as shown below are 7.5 and 5.7 ksi respectively.
- The stresses at the fillets are: 13.103, 8.918, 12.712, and 12.436 < 13.3 ksi. Hence design is safe.

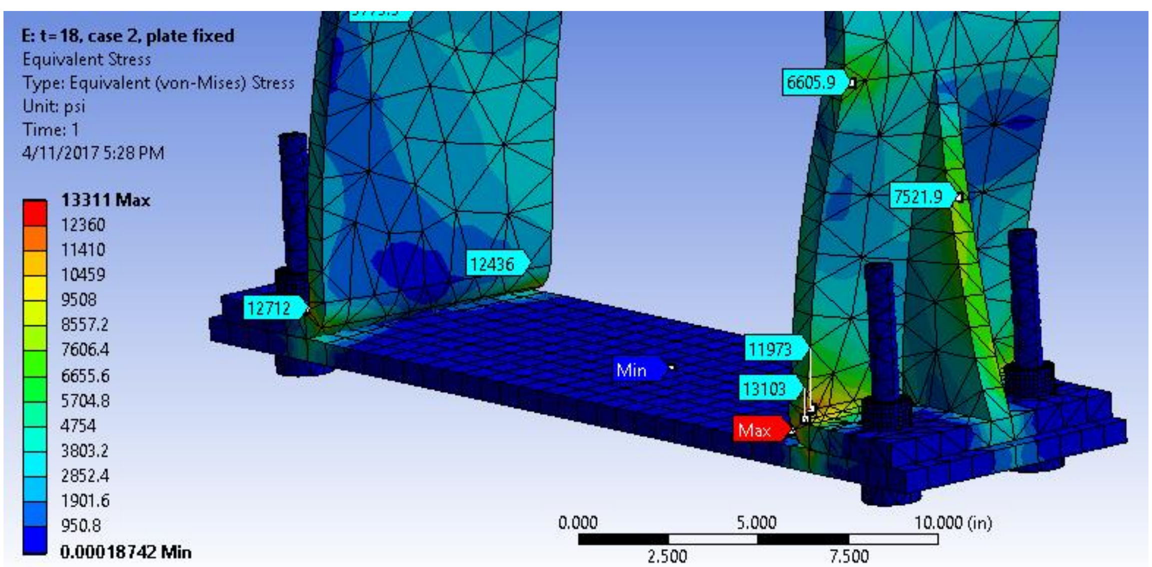
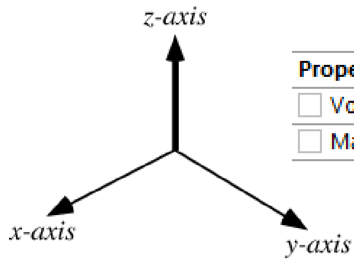


Fig 4.21: Equivalent Stresses for case 2.

CASE 3:

- Fx= 1169 N
- Fy= 22761 N
- Fz= 19700 N



Properties	
Volume	8.9448e+006 mm^3
Mass	71.768 kg

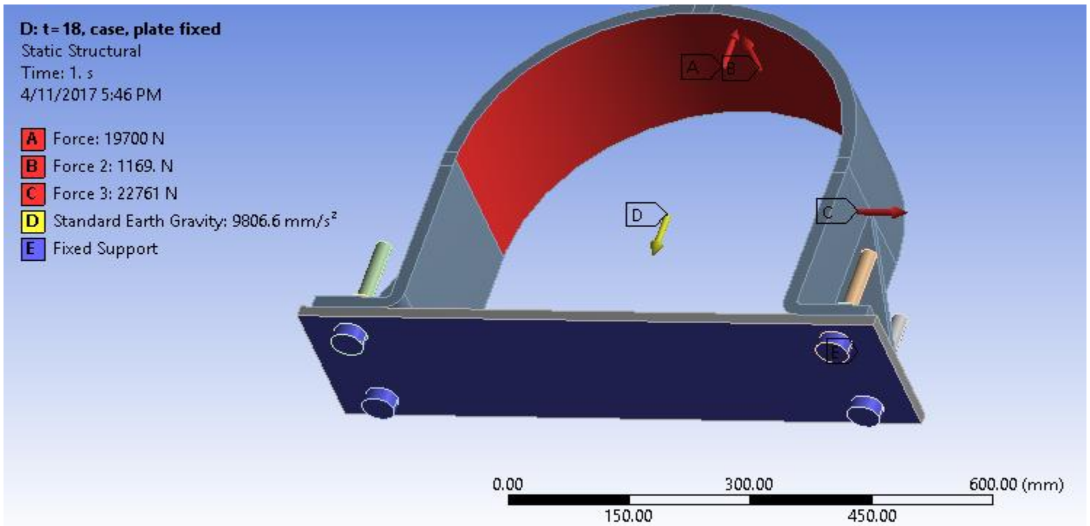


Fig 4.22: Force representation for case 2

(a) Deformation

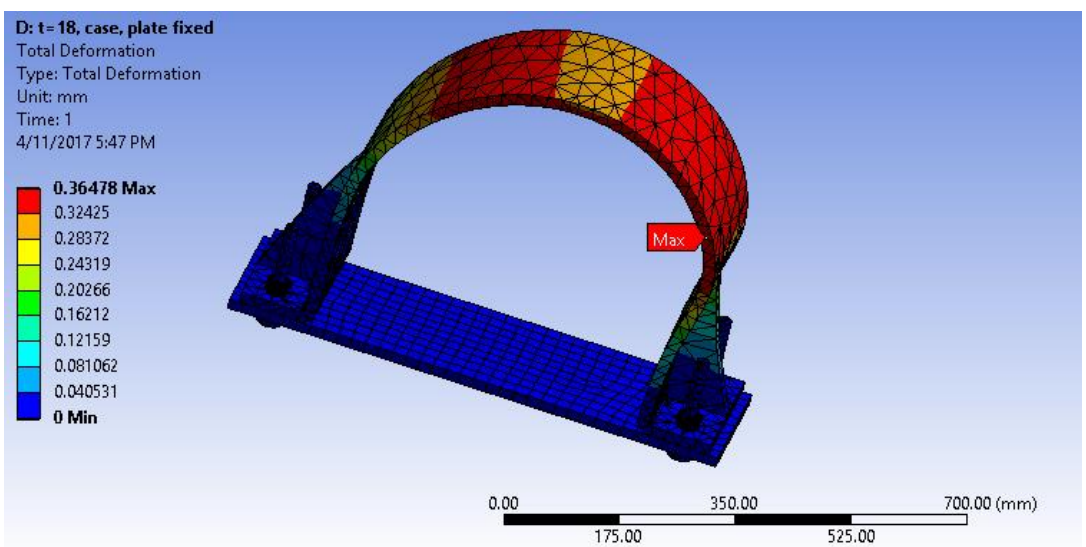


Fig 4.23: Maximum deformation for case 3

(b) Equivalent Stresses

- The max stress is on the rib, 12.383 < 13.3 ksi and hence safe.

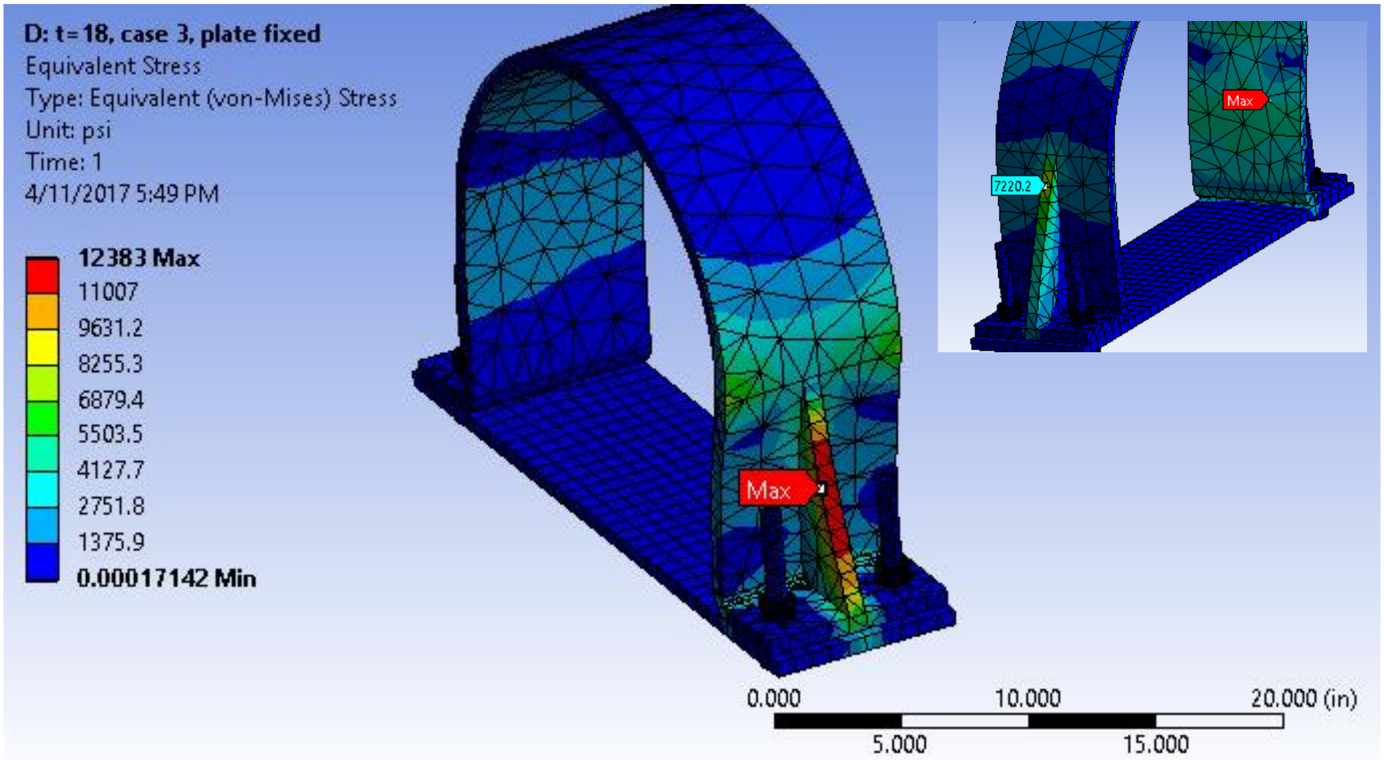


Fig 4.24: Equivalent Stresses for case 3

Summary of Loading Conditions for thickness of plate = 18mm

Table 4.3: Summary of Loading Conditions for thickness of plate = 18mm

Loading Condition	Fx (KN)	Fy (KN)	Fz (KN)	Deformation	Stresses Induced
1	1.343	26.213	31.495	0.4088mm	13.033 ksi
2	29.596	15.543	18.675	0.30354 mm	13.311 ksi
3	1.169	22.761	19.7	0.36478mm	12.383 ksi

We can conclude that thickness of 18 mm can withstand all the loading conditions and hence it is safe to use.

4.5 Length Optimization of clamp

- Length optimization is carried out to cut down on the material being used whilst keeping the induced stresses under the allowable limit.
- Here I have reduced the length of the clamp from the current l= 749.3mm to the standard lengths as available in various catalogues [2] [3] [4].
- The standard length available are: l= 690, 710,749.3 and 760mm.
- Length less than 690mm is not taken as then the surface area for the bolt head will decrease very much.
- Length more than 760mm is not taken as then it will interfere with the geometry of construction.
- The stress is first calculated for loading condition 1 and if the clamp passes, loading condition 2 and loading condition 3 are considered.

1) Length L= 690mm

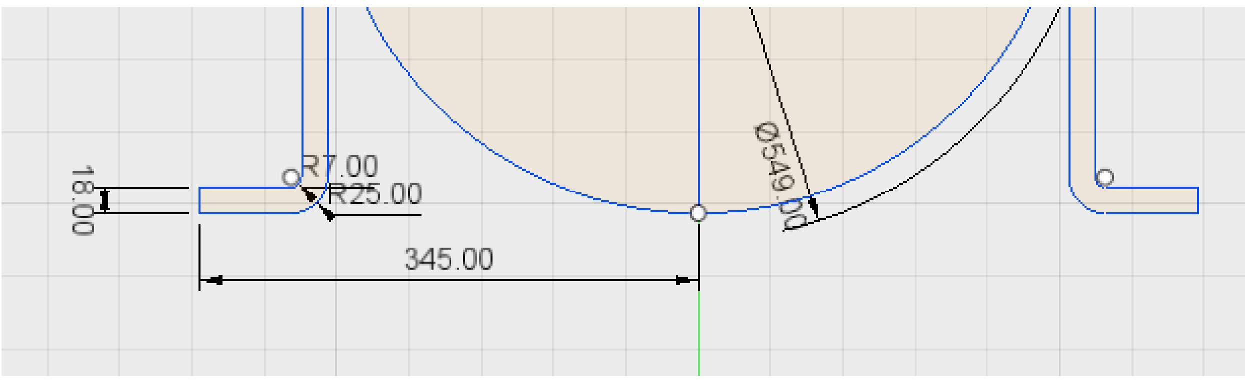


Fig 4.25: Sketch of Clamp to indicate the change in length to L= 690mm

(a) Equivalent Stresses:

- The maximum stress generated is again in the bolt and has decreased to 26.444 ksi from 31.523 ksi.
- The stress on the ribs and the lower part of the clamp had risen significantly from 11.3 ksi to 16.69ki, thus exceeding the allowable stresses of 13.3 ksi and making the clamp unsafe.
- The deformation is 0.9154 mm.

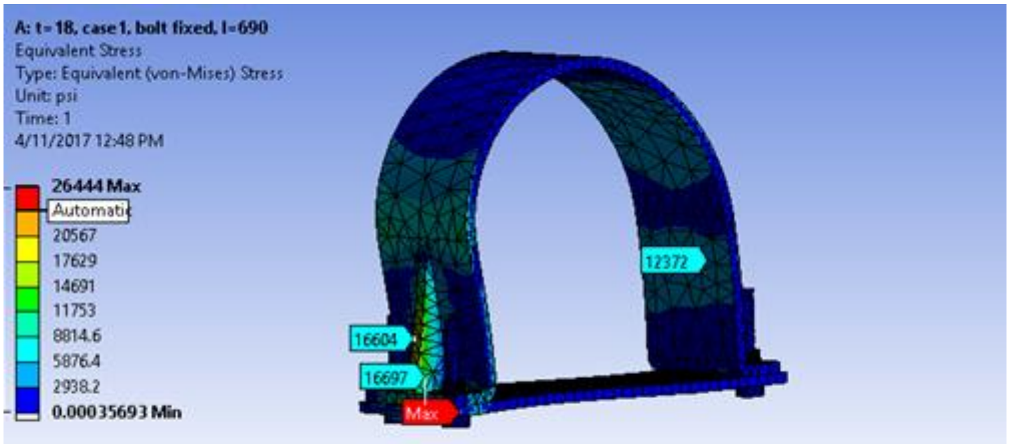


Fig 4.26: Equivalent Stresses when L=690mm

2) Length L= 710mm

- The maximum stress generated is again in the bolt and has decreased to 28.351 ksi from 31.523 ksi.
- The stress on the ribs and the lower part of the clamp had risen significantly from 11.3 ksi to 14.235ki, thus exceeding the allowable stresses of 13.3 ksi and making the clamp unsafe.
- The deformation is 0.65709 mm

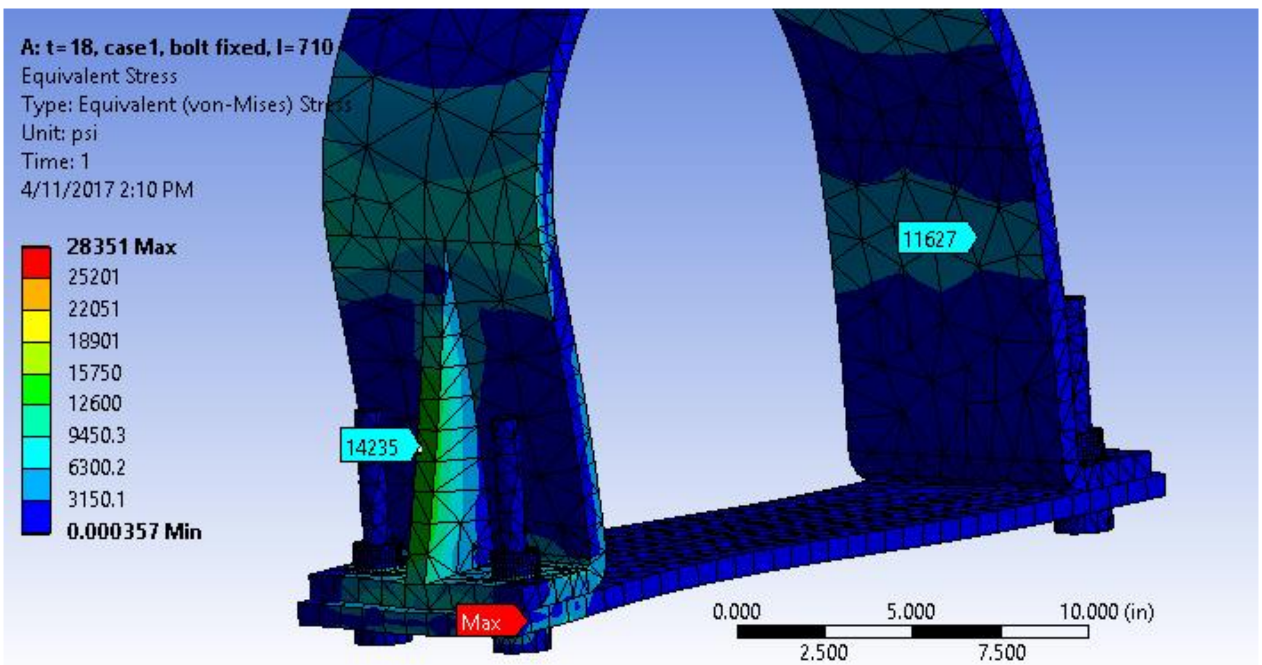


Fig 4.27: Equivalent Stresses when L=710mm

3) Length L=760mm

- The maximum stress generated is again in the bolt and has decreased to 26.527 ksi from 31.523 ksi.
- The stress on the ribs and the lower part of the clamp had also decreased to 10.550 ksi compared 14.235ki of the original length, thus higher length is preferred over l= 749.3 mm, for case 1.

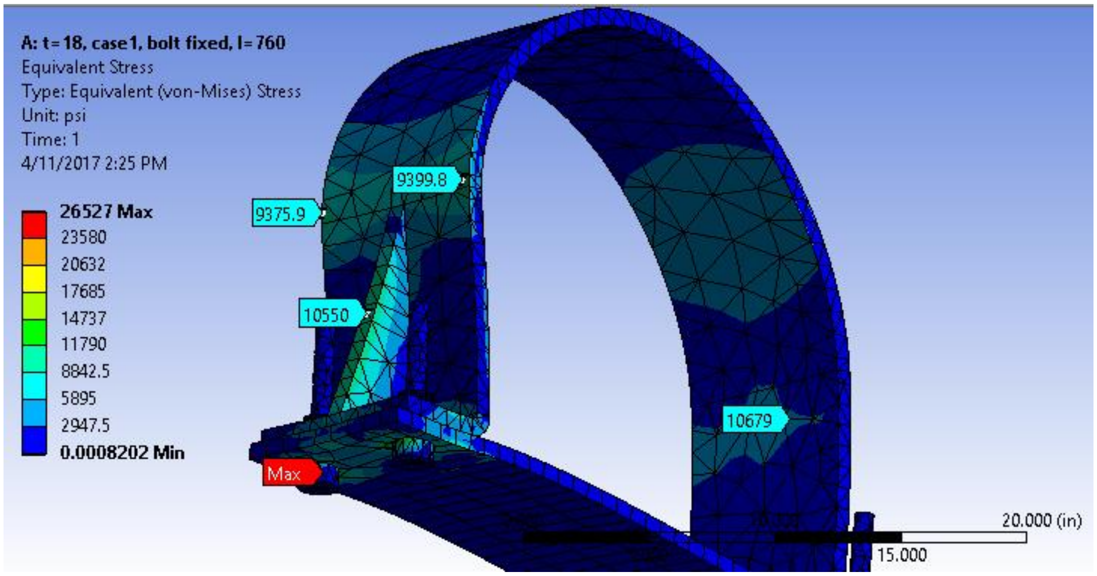


Fig 4.28: Equivalent Stresses when L=760mm, Case 1

- Now as l= 760 has passed the allowable stress limit for case 1, we will analyses it for case 2.

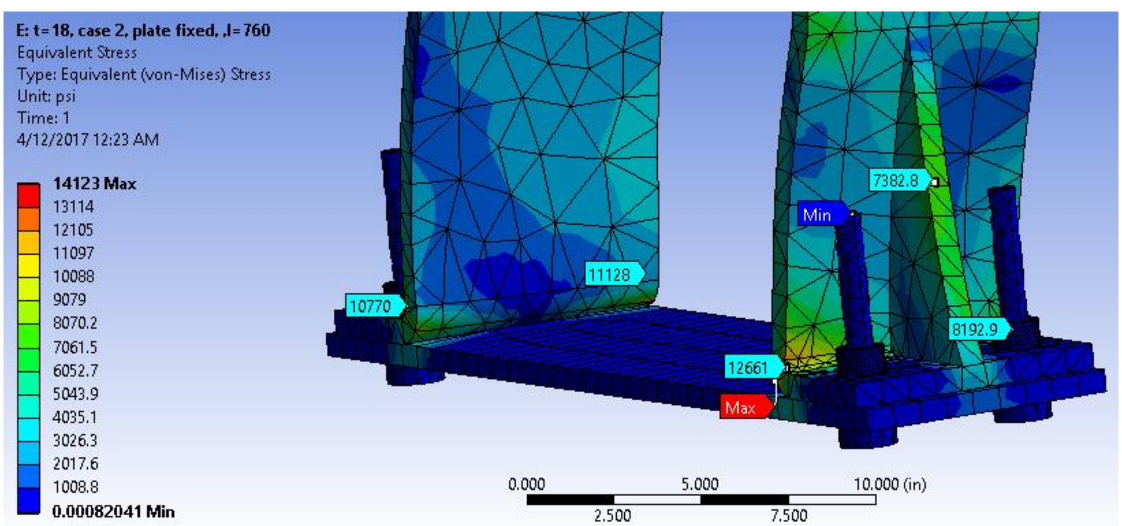


Fig 4.29: Equivalent Stresses when L=760mm, Case 2

- Here we see that for case 2, the higher length clamp is failing at 14.123 ksi exceeding the allowable limit of 13.3 ksi.
- For reference, the figure below is for original l=749.3mm under loading condition 2.

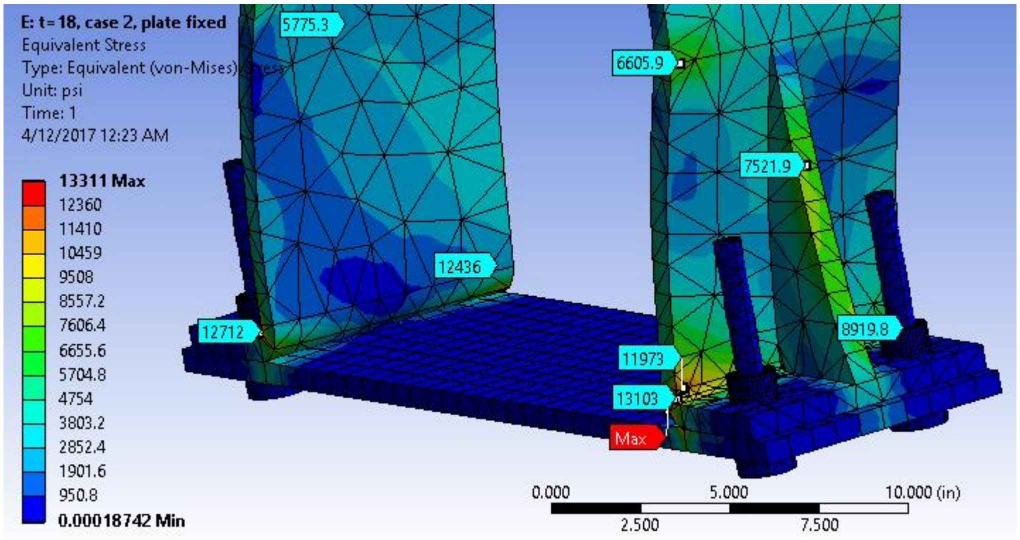


Fig 4.30: Equivalent Stresses when L=749.3 mm, Case 2

- From the above results we can conclude that with increase in length the stresses decreases for case while they increase for case 2.
- As L= 760 mm failed to be safe under case 2, analysis for case 3 was not done.
- Below is a summary of the above analysis:
  - The deformation is in (mm)
  - The stresses presented are maximum stresses which are on the ribs for case 1 and in the fillet under tension for case 2. (Refer Fig22).
  - Stresses are in ksi.
  - The allowable stress is 13.3 ksi. [5]

Table 4.4: Summary of Length Optimization

Length L= (mm)	Case 1		Case 2		Case 3	
	Deformation	Stress	Deformation	Stress	Deformation	Stress
749.3	0.40884	13.033	0.30354	13.311	0.36478	12.383
690	0.9154	16.69				
710	0.81805	14.235				
760	0.65709	10.550	0.5961	14.123		

- After carrying out length optimization, we can conclusively say that the original arrangement with L= 749.3mm would be safer than the other standard lengths available.

4.6 Fillet Radius Optimization

- Now once we have decided on the length of 749.3 mm, we will carry out fillet radius optimization.
- Initially for all the analysis a fillet radius of R=25 and r=7 were taken. These were taken because a fillet radius of less than r=7 is difficult to fabricate.
- Therefore we will try and carry out analysis for a higher fillet radius of R=30 and r=12.
- Further increase in fillet radius is not possible as, then the length available for the nut to rest on will decrease drastically.
- Here I have presented analysis for the optimized length of 749.3mm and 760 mm for case 1 and case 2, so that we can derive a relation between the fillet radius and the stresses induced. Case 3 is not considered as it is similar and lesser in magnitude to case 1.
- All the data is in tabular form to easily understand it.
    - The stresses presented are maximum stresses which are on the ribs for case 1 and in the fillet under tension for case 2. (Refer Fig22).

Table 4.5: Fillet Radius Optimization.

Length L= (mm)	Stress (case 1) (ksi)		Stress (case 2) (ksi)	
	R= 30mm	R=25mm	R= 30mm	R=25mm
760	12.307	12.391	14.271	14.123
749.3	12.881	13.033	14.621	13.311

- From all the above cases, we can conclude that:
    - In case 1, stress decreases with increase in fillet radius.
    - In case 2, stress increase with increase in fillet radius.
- Therefore we will use the original fillet radius of R=25mm and r=7m

Conclusion:

- **From the above analysis, we can conclude that a thickness of 18 mm, l = 749.3 mm, R= 25mm and r= 7mm clamp size can withstand all the forces under various cases and therefore the clamp will be safe.**

4.7 Case 2: Optimization

During this analysis, we came up with another solution to withstand the forces for case 2. The analysis is presented here.

- The plate supporting the pipe is shortened here. That is only the region where the pipe will be in physical contact with the plate, will be taken. This would drastically reduce the material requirement for bottom plate.
- Now as there is no bottom plate under the clamp, analysis is carried out for clamp without any bottom plate.
- This optimization was required as we needed to bring the stresses below the allowable limit, which right now are just above the limit of 13.3 ksi.
- **Geometry**:

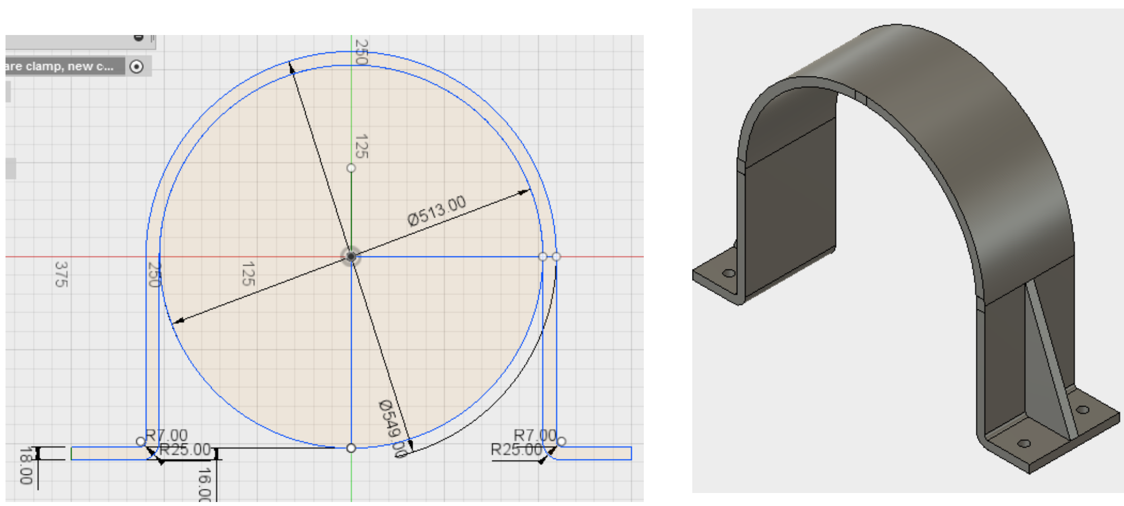
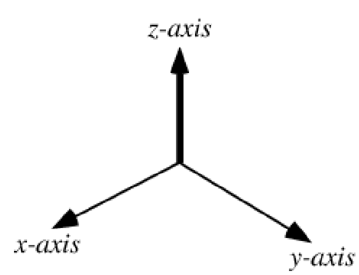


Fig 4.31: Sketch for clamp without bottom plate.

- Case 2:
    - Fx= 39596 N
    - Fy= 15543 N
    - Fz= 18675 N



- We can see that the stresses induced are higher than the allowable stress and therefore clamp without bottom plate will fail in case 2.

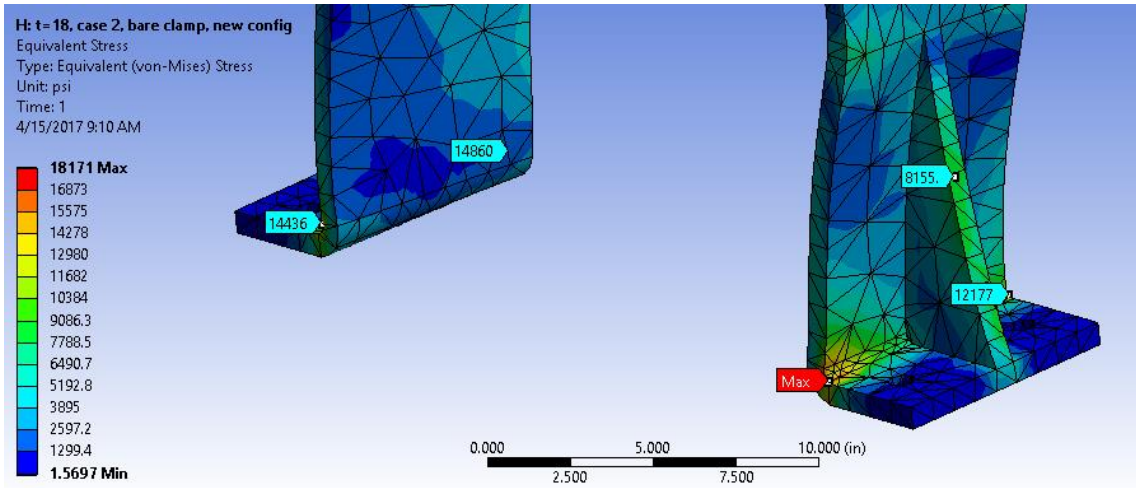


Fig 4.32: Stress for case 2 without bottom plate

Now as the point of failure is the fillet section of the clamps, two ribs are used to reduce the induced stresses.

- **Geometry:**

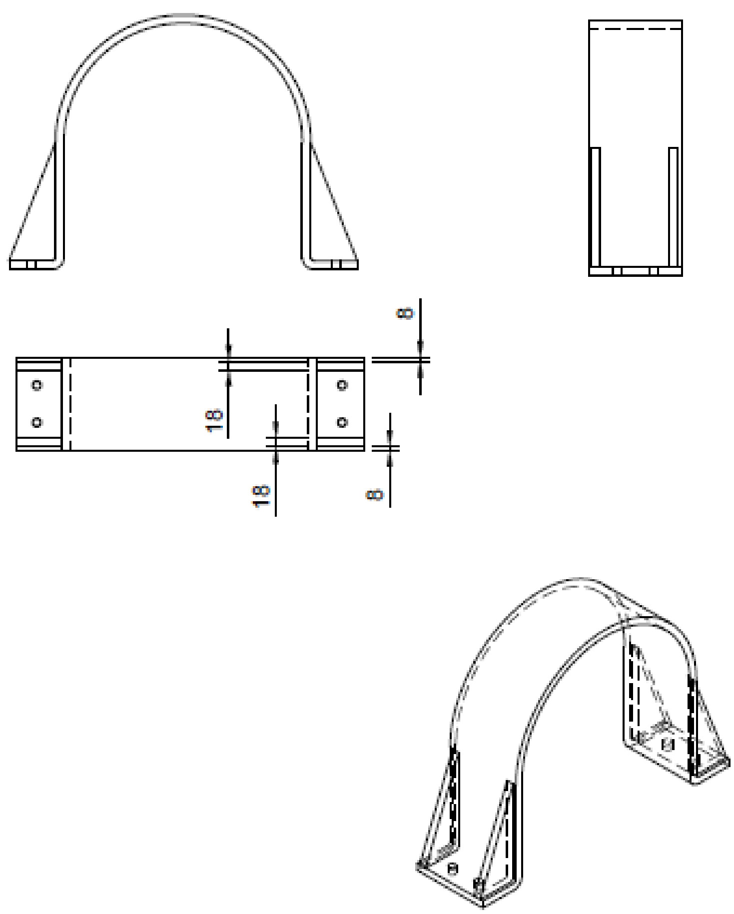


Fig 4.33: Sketch of clamp with two ribs

Equivalent Stresses induced in Case2:

- **As the stress is still above 13.3 ksi, we will increase the width of the clamp to 250 mm from 200mm.**

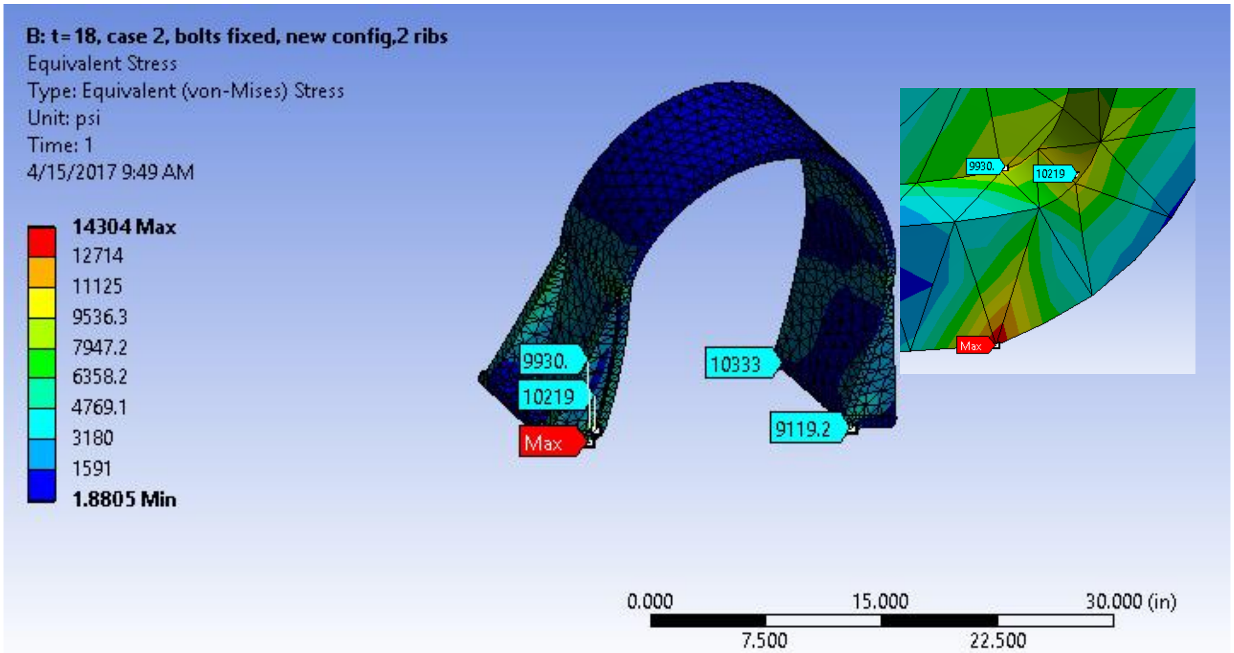


Fig 4.34: Stresses induced for clamp with two ribs, case2

Equivalent Stresses induced in Case2 with width = 250mm

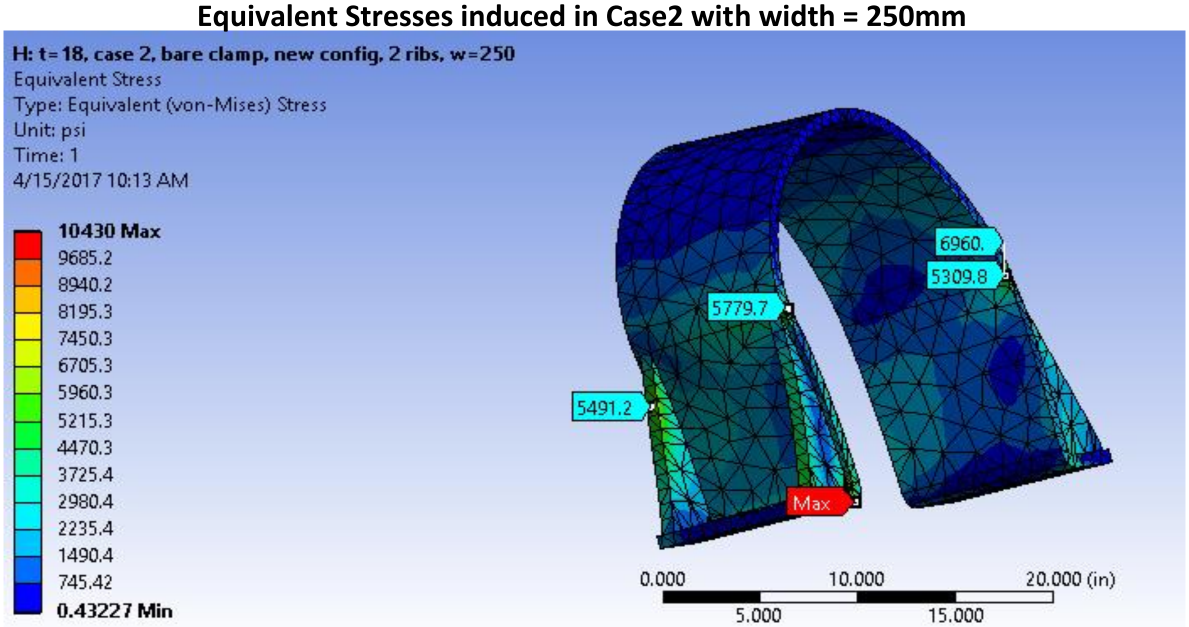


Fig 4.35: Stresses induced for clamp with two ribs, case2 and width = 250m

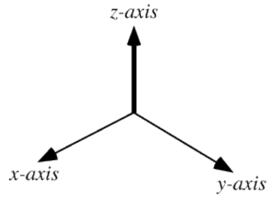
- As the stresses are below the allowable limit of 13.3 ksi. The clamp is safe.

Conclusion:

- **For case 2 another solution is developed where, the length of the bottom plate is reduced and two ribs are provided for the support. Also the width of the clamp is increased to 250mm here.**

4.8 Case 3: Optimization

- As the number of clamps required are very high in this case, separate optimization is needed to check whether any other option is available or not.
- Case 3:
    - Fx= 1169 N
    - Fy= 22761 N
    - Fz= 19700 N



- The data is presented in tabular format as follows:

Table 4.6: Case 3 Optimization

Thickness of clamp	Thickness of rib (mm)	Stress (ksi)
t= 14 mm	14	20.101
t =16 mm	16	16.034
t= 16mm	18	13.784
t=18 mm	18	**12.383**

- **From the above analysis, we can conclude that only t=18 mm and rib thickness of 18mm will be able to withstand the forces of case 3.**

4.9 Clamp Design for condition 2 loading.

On a later stage the loading conditions was changed to pressure loading for more uniform force distribution. Also the lateral load was changed to area loading from line loading.

The area is calculated based upon the dimensions presented in the geometry section.

Below are the diagram to reflect the change in loading conditions for all the three cases.

Table 4.7: Pressure Loading Conditions

	Case 1			Case 2			Case 3		
	Fx	Fy	Fz	Fx	Fy	Fz	Fx	Fy	Fz
Force N	26213	31495	1343	15543	18675	39596	22761	19700	1169
Area cm2	2090.3	2015.3	2015.3	2090.3	2015.3	7.5	1662.2	1612.6	1612.6

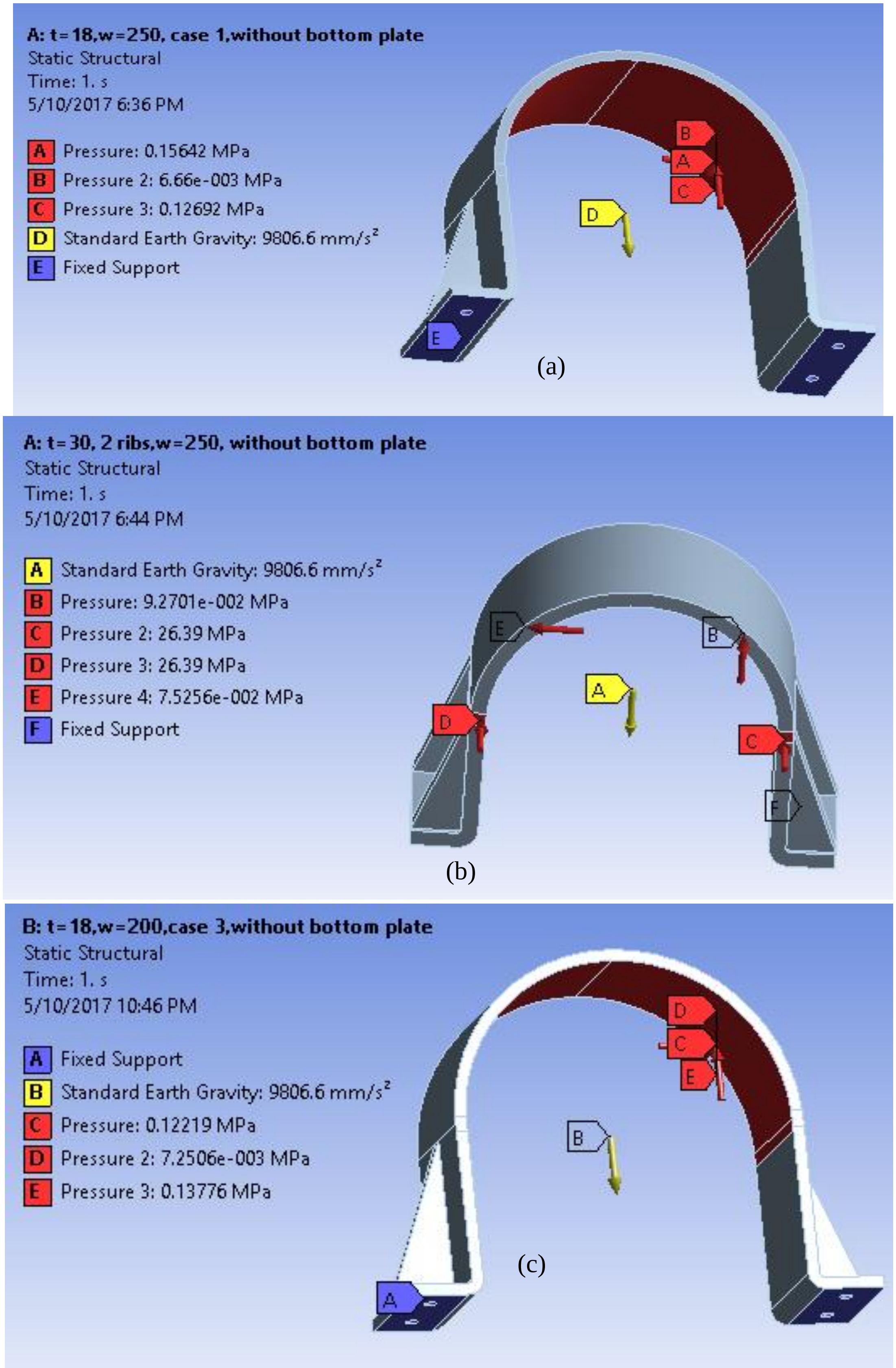


Fig 4.36: Pressure loading for all the three cases is depicted above.

Table 4.7: Stresses for all the three cases with loading condition 2,

t= thickness of plate, t1= thickness of ribs, w= width of the clamp.

thickness	Case 1	Case 2	Case 3
t=16, t1=16,w=200	-	-	15.496
t=16, t1=18,w=200	-	-	15.439
t=16, t1=18, w=250	16.377		**13.030**
t=18, t1=18,w=200	14.426	-	**11.374**
t=18, t1=18,w=250	**12.763**		
t=18, t1=20,w=200	13.904	-	
t=20, t1=20,w=200	**11.446**	45.128	
t=20, 2 ribs, t1=20 ,w=200		25.865	
t=25. 2 ribs, t1=25,w=200		18.867	
t=30. 2 ribs, t1=30,w=200		14.504	
t=30. 2 ribs, t1=30,w=250		**13.256**	
t=25. 3 ribs, t1= 25,w=200		17.591	
t=30. 3 ribs, t1=30,w=200		**13.184**	

In the above table the fields marked red are above the allowable limit of 13.3 ksi whereas the fields marked green are safe and under the limit.

In all the above cases two design solutions are developed. The client can then choose any of the design configuration according to his needs.

Case 1: 1) 18 mm thick plate with rib of 18 mm and width of clamp equal to 250 mm.

2) 20 mm thick plate with rib of 20 mm and width of clamp equal to 200 mm.

Case 2: 1) 30 mm thick plate with 2 ribs of 30mm and width of clamp equal to 250mm.

2) 30 mm thick plate with 3 ribs of 30mm and width of clamp equal to 200mm.

Case 3: 1) 16 mm thick plate with rib of 18 mm and width of clamp equal to 250mm.

2) 18 mm thick plate with rib of 18 mm and width of clamp equal to 200 mm.

These results are approved by the design team at LT-SL and will likely to be incorporated in the technical specs.

In the diagrams below, the stress distribution for the final configuration is depicted.

Case 1: 18 mm thick plate with rib of 18 mm and width of clamp equal to 250 mm.

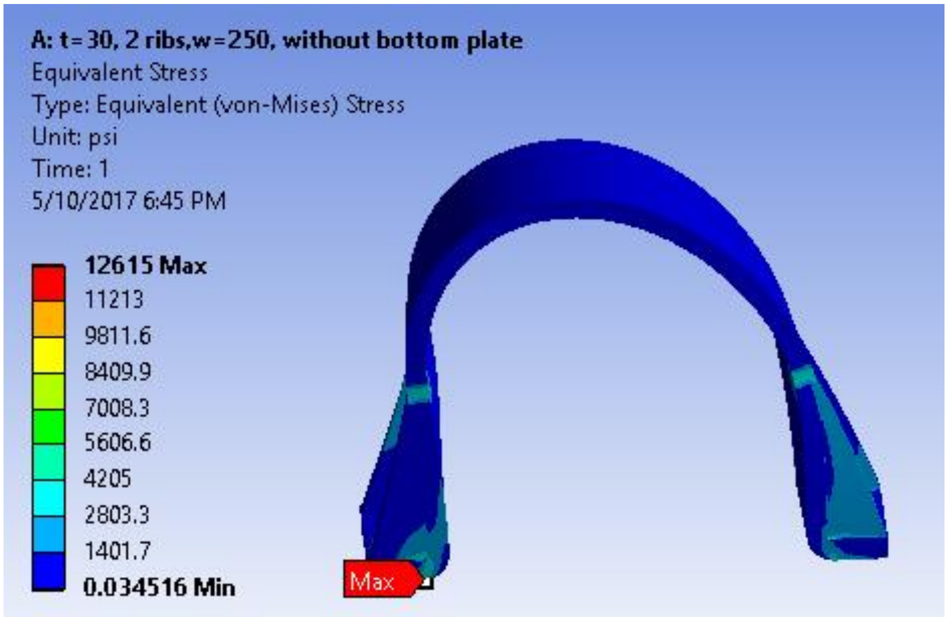


Fig 4.37: Stress Distribution for case1, t= 18, t1= 18, w= 250mm with 1 rib.

The maximum deformation for the above case is 0.57958

Case 2: 30 mm thick plate with 2 ribs of 30mm and width of clamp equal to 250mm.

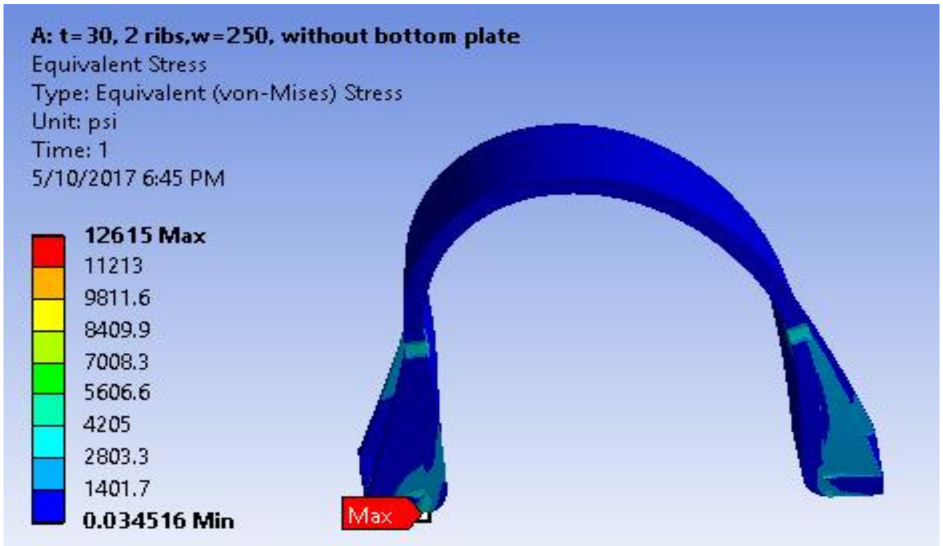


Fig 4.38: Stress Distribution for case2, t= 30, t1= 30, w= 250mm with 2 ribs.

The maximum deformation here is 0.10735 mm.

Case 3: 18 mm thick plate with rib of 18 mm and width of clamp equal to 200 mm.

In the above 2 cases, the configuration with minimum thickness is always selected but for case 3 it is reversed. This is because ordering a 16mm separate plate would be uneconomical and therefore a common plate of 18mm will be ordered which can be used for both case1 and case 3.

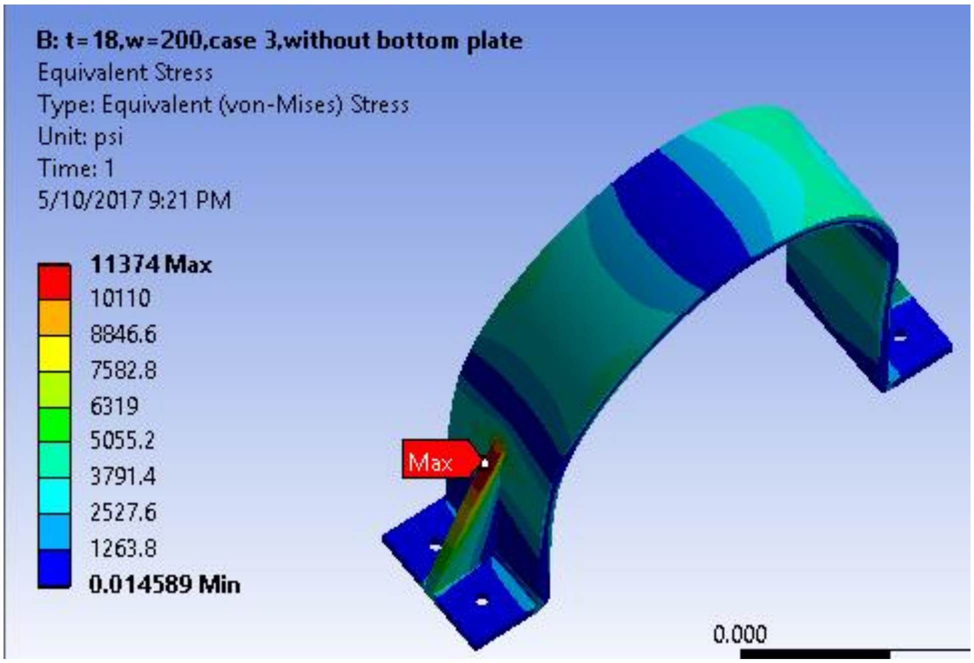


Fig 4.39: Stress distribution for case 3, t=18, t1=18, w= 200mm

The maximum deformation is 0.59031 mm.

The summary of loading condition 2 is as follows:

Table 4.8: Stress analysis for loading condition 2.

	Fx	Fy	Fz	Deformation (mm)	Stress (ksi)
Case 1	26213	31495	1343	0.57958	12.763
Case 2	15543	18675	39596	0.10735	12.615
Case 3	22761	19700	1169	0.5907	11.374

CHAPTER 5

Conclusion

5.1 Conclusive remarks for Pipe Design

The pipe is to be designed for the connection between the condensate storage tank and the demoralized water storage tank. Initially the thickness is calculated after which the procured thickness is estimated based on various cross checks and the standard thicknesses available in the market.

Input Conditions:

- Internal gauge pressure = 34.46 kg/cm2g
- Design temperature = 27`c
- The corrosion allowance is 1.2 mm. [4]

Final thickness for various diameter sizes. The material is A106 Type-B.

Table 5.1: Procured thickness for pipes.

Diameter	O.D	Procured thickness
NPS(in)	mm	
0.5	21.3	2.11
6	168.3	7.11
24	610	22.83
42	1067	37.73002012
56	1422	49.88386937
64	1626	56.86805316

After calculating the thickness of the pipe, the piping system was modeled in Caesar II from the isometric.

The code stress ratio or the allowable stress to the induced stress ration determines whether the pipe is safe or not. The code stress should always be greater than 1.

For Sustained loading: Code stress ratio = 6.1 and hence safe.

For hydro test: Code stress ratio= 4.9

Hence the pipe support arrangement is safe for design.

5.2 Conclusive remarks for clamp design

The clamp is to be designed and optimized for three loading conditions. The hold down clamp is to be used for holding the pipe designed above.

Initially the lateral force was given as line load and the loads were directly taken as the forces calculated. The results are as follows:

Table 5.2: Summary of Loading Conditions for thickness of plate = 18mm

Loading Condition	Fx (KN)	Fy (KN)	Fz (KN)	Deformation	Stresses Induced
1	1.343	26.213	31.495	0.4088mm	13.033 ksi
2	29.596	15.543	18.675	0.30354 mm	13.311 ksi
3	1.169	22.761	19.7	0.36478mm	12.383 ksi

On a later stage, another type of configuration was to be calculated for which the lateral load was to be applied over a large surface area. Also the load application was to be converted into pressure loading for all the remaining loads. The table below reflects the finalized configuration for the new loading conditions.

Table 5.3: Stresses for all the three cases with loading condition 2,

t= thickness of plate, t1= thickness of ribs

thickness	Case 1	Case 2	Case 3
t=16, t1=16,w=200	-	-	15.496
t=16, t1=18,w=200	-	-	15.439
t=16, t1=18, w=250	16.377		**13.030**
t=18, t1=18,w=200	14.426	-	**11.374**
t=18, t1=18,w=250	**12.763**		
t=18, t1=20,w=200	13.904	-	
t=20, t1=20,w=200	**11.446**	45.128	
t=20, 2 ribs, t1=20 ,w=200		25.865	
t=25. 2 ribs, t1=25,w=200		18.867	
t=30. 2 ribs, t1=30,w=200		14.504	
t=30. 2 ribs, t1=30,w=250		**13.256**	
t=25. 3 ribs, t1= 25,w=200		17.591	
t=30. 3 ribs, t1=30,w=200		**13.184**	

The reason for such an increase in thickness can be accounted as the lateral force application has been changed and therefore because of the increase moment there is an increase in stresses induced.

5.3 Future Work

Another aspect where there is a large need for optimization is material optimization. A change in material can bring large dimensions changes if its allowable stresses are very high.

Also, a study can be carried out to find out whether the increased cost of a much harder material can compensate for the increased dimensions when a lower grade of material is used. If results are conclusive then it can play a major role in cost optimization.

Another area of future work is designing different innovative ribs. Currently in the thesis, the failure in case 1 and case 3 is happening on the ribs itself. This can be accounted by designing ribs by varying the thickness across the ribs. As in, give more thickness where failure is occurring and vice versa. If the manufacturing of such component is possible, large cost reductions are possible.

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