Design and Analysis of cold box and its internal component layout for kW class Helium Refrigerator/ Liquefier

A Thesis Submitted in Partial Fulfilment of the Requirements for

The Degree of

Master of Technology

In

Mechanical Engineering

By

Punit Kar
212ME5327

Department of Mechanical Engineering

National Institute of Technology

Rourkela

2014

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Under the supervision of

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<u>CERTIFICATE</u>

This is to certify that the dissertation, entitled

"Design and Analysis of cold box and its internal component layout for kW class Helium Refrigerator/ Liquefier"

is a bonafide work done by

Punit Kar

Under my close guidance and supervision in the Large Cryogenic Plant and Cryosystem Group

of

Institute for Plasma Research, Gandhinagar, Gujarat

for the partial fulfilment of the award for the degree of ***Master of Technology*** *in* ***Mechanical Engineering*** *with Specialization in* ***"Cryogenic and Vacuum Technology"*** *at*

National Institute of Technology, Rourkela.

The work presented here, to the best of my knowledge, has not been submitted to any university

for the award of similar degree.

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The thesis in my opinion has reached the standards fulfilling the requirement for the award of the degree of **Master of Technology** in accordance with regulations of the institute

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I shall be failing in my duty, if I don't express my thanks to **NIT ROURKELA** for providing me the financial help in the form of stipend and also encouragement to complete the study successfully.

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ABSTRACT

The indigenous HRL will have option for upgrading its cooling capacity upto ~2 kW at ~4.5 K. The cold box containing all cold equipment is designed considering strength and thermal aspects. Component layout inside the chamber is decided to maintain a temperature gradient through the length of the chamber. So, the chamber can be thought of two chambers; one 4 K part and another 20 K part. Stiffening rings, covers, gasket and saddle supports are designed. Strength analysis of the cold box is done in ANSYS. Piping and flexibility analysis is done in CAEPIPE. Support structures for all components are designed for permissible heat conduction as per HRL requirement. Entire cold box with its components is modelled in CATIA.

Chapter 1.

INTRODUCTION

1.1 Institute for Plasma Research:

Institute for Plasma Research (IPR) is the place where efforts are being made in one of the most challenging and necessary tasks of this century; “controlling nuclear fusion”. The idea is that energy can be obtained by fusing nuclei of light elements to produce heavier elements; which have been a process that occurs in the Sun where the fusion of hydrogen is the principle source of its energy. But the real challenge is to create this form of energy on earth by recreating the conditions of the Sun in the laboratory. The pursuit of this goal has been a worldwide effort over the last forty years. The Institute for Plasma Research located at the outskirts of Ahmedabad, is a recent entrant in this endeavour and is the prime expression of India's commitment to this futuristic energy source.

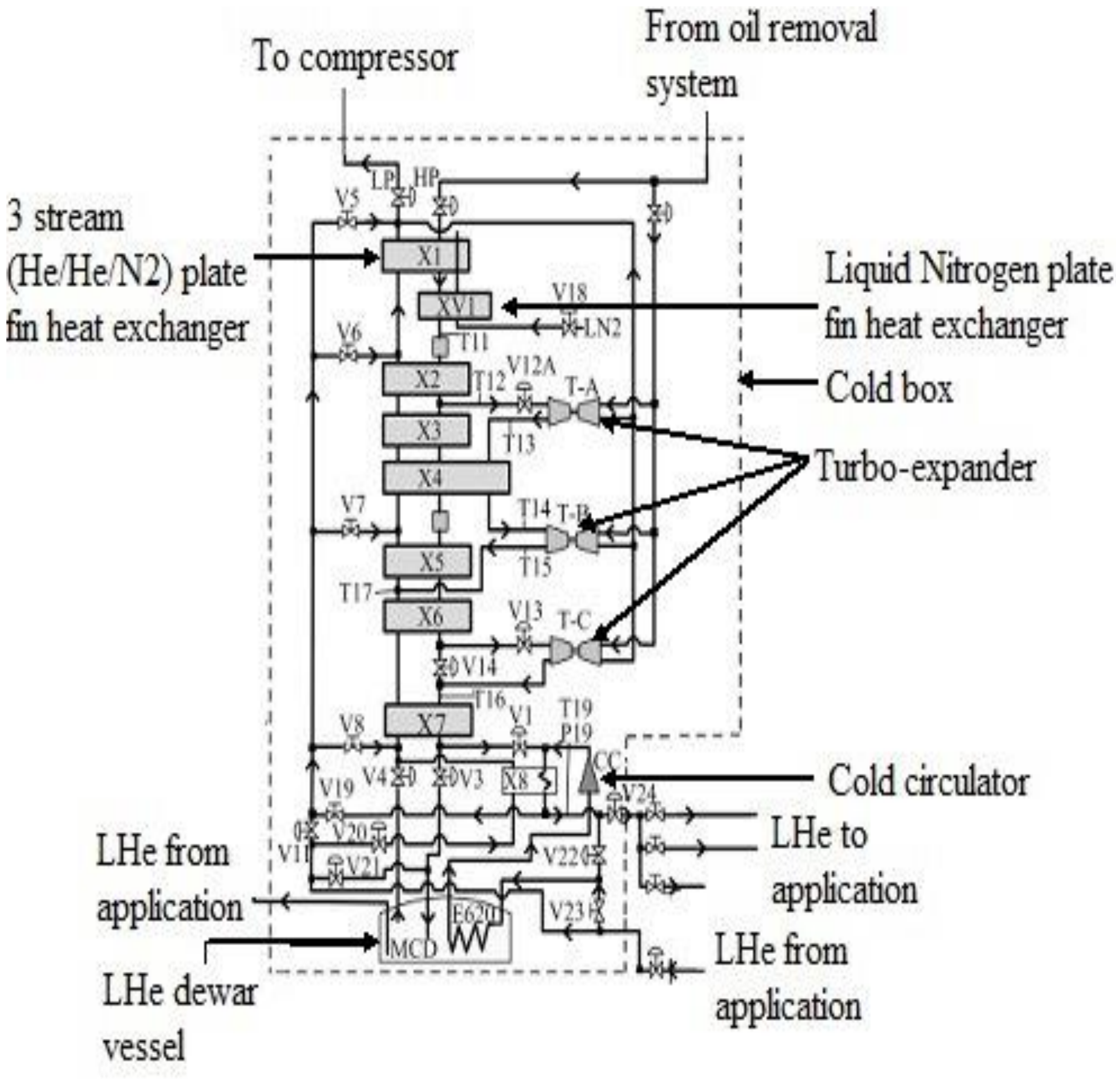
The Institute has a broad charter of objectives to carry out experimental and theoretical research in plasma sciences with emphasis on the physics of magnetically confined plasmas and certain aspects of nonlinear phenomena. The institute also has a mandate to stimulate plasma research and development activities in the universities and the industrial sector. This institute in its experimental activity has embarked on an ambitious project of building the first Indian Steady State Superconducting (SST-1) Tokamak. The helium refrigerator/liquefier (HRL) of 1.3 kW capacity for the Steady State Superconducting Tokamak (SST-1) is successfully operating independently along with integrated flow control and distribution system.

1.2 Cryogenics:

Demand for cryogenic temperatures (below 120K) has seen a peak since the middle of 20th century. Cryogenics has a very wide variety of applications in physics, chemistry, medicine, biology, biotechnology, engineering and industries. IPR has used the essential and advanced cryogenic technology of producing extremely low temperatures of 20K-4.2K, which uses liquid Helium as the refrigerant. Production of liquid helium led to production of superconductivity, which made a revolutionary change in science and technology. Liquid Helium and super fluid Helium are majorly used for the cool down of the Tokamak superconductor and proper conditioning of the superconductor operating environment.

1.3 Helium Liquefaction Process[1]:

Liquefaction of Helium is achieved by using “Modified Claude Cycle”.



(Figure 1: Typical process flow diagram cold box components of HRL of Tokamak superconducting magnet system)

The above schematic diagram is explained below. Pure helium is compressed by an oil injected screw compressor from atmospheric pressure (1.05 bar) to approximately 14 bar (dependant on the chosen compressor). The compression heat is being removed by an either air or water cooled after cooler. Oil removal system (ORS) is used to remove the residual oil in Helium to ensure that it does not contaminate the liquefier. The oil removal is performed in several stages by bulk oil separator, 3 coalescers and an oil vapour charcoal adsorber.

The pure high pressure gas is fed to the cold box (CB). The CB consists of 8 heat exchangers, 3 turbines, a CC (cold circulator) and 2 Joule-Thomson (J-T) valves to generate liquid Helium in the main process cycle. The heat exchangers X1 to X5 along with turbines A and B connected in series form a pre-cooling section for the downstream helium flow, while J-T valves (V1 & V3) provide J-T cooling effects at the cold end of cold box as shown in figure. The heat exchanger combination of X1-XV1 cools the downstream helium flow from 300 K to 80 K with help of liquid nitrogen (LN2). The turbines A and B reduce the temperature of downstream helium flow from about 40 K to 15.5 K bypassing the moderate flow from the total compressor flow. The heat exchangers X6 and X7 along with return flow from main control dewar (MCD) reduce the temperature of downstream helium flow up to ~6 K. If third turbine between these two heat exchanger is operated the temperature further comes down to about 5.3 K. The J-T valves on the downstream of the heat provides the J-T expansion to produce liquid and vapour mixture and two-phase fluid is separated in the MCD.

The return cold gas from the MCD through cold box LP to the compressor suction forms the closed loop as a refrigerator, while heat loads from SCMS is transferred to MCD to evaporate LHe using a heat exchanger inside the MCD in the circulating loop of the magnet helium cooling. The CC along with heat exchanger X8 and E620 completes the cooling loop for SCMS. The plate-fin heat exchanger X8 immersed in the LHe bath at 1.2 bar is used for stabilizing SHe temperature to 4.5 K , which is supplied to SCMS.

1.4 Function of Cold Box:

Cold box is a chamber vacuumed for the purpose of containing all the liquefaction equipment necessary for the LHe plant. Primary functions of the cold box are:

- House and support the cold equipment
- Protect them from unwanted pressure and heat loads
- Provide suitable vacuum environment
- Provide and control access to the materials, equipment and processes, which are located inside.
- House all piping and instrumentations for the equipment

The cold box of planned indigenous helium plant will constitute the following major equipment:

1. Three ultrahigh speed turbo heat expanders with their casing
2. Eight highly effective aluminium plate fin heat exchangers
3. Two charcoal adsorbers; one at 80K and another at 20K; to adsorb possible impurities in Helium gas
4. Two filter cartridges to check residual charcoal constituent in Helium
5. One liquid Nitrogen vaporizer to provide for the rapid cooling of Helium gas at the first heat exchanger.
6. One liquid Helium bath
7. One cold circulator
8. One ejector pump
9. Multilayer super insulation to check the heat leak into the chamber
10. LN2 cooled thermal shield to protect the equipment from radiation heat load
11. Supports for all the equipment
12. Process piping including high and low pressure piping
13. Cryogenic extended stem valves (J-T valves)
14. Instrumentation for measurement, monitoring and control of process parameters.
15. Auxiliary units such as pressure relief valves, cooling water supply for the turbo expanders and vacuum pumping units (rotary vane and oil diffusion or turbo molecular pump)

For proper liquefaction of Helium and effective performance of all the above devices, piping and support units; the entire system should be free from atmospheric disturbances such as heat load and pressure. That is why the need of an ultrahigh vacuum comes up. After installation of all equipment and structures, the cold box is vacuumed by using vacuum pumps; so as to satisfy the objective of providing for appropriate vacuum environment.

Cold box is used in many engineering applications, including low temperature storage tanks, air separation, petrochemical industries and ultra-low temperature system of hydrogen or helium liquefaction. Cold box is necessarily pumped to vacuum to save very high quality cold energy from spreading out to atmosphere. Atmospheric heat leak into the system takes place basically by a mixed effect of radiation and conduction. To avoid these, thermal shield is employed in the system periphery. Thermal shield is a thin cylindrical shell, cooled by liquid Nitrogen running in tubes on its surface in zigzag fashion. A major fraction of external radiation coming into the system is checked by the shield. Also, few layers of multilayer insulation can be used to enhance the performance of thermal shield.

1.5 Objective of thesis:

This thesis targets for the following objectives to be met:

1. Design and analysis of the vacuum chamber to accommodate all equipment
2. Optimization of chamber dimensions including saddle supports
3. Design and analysis of thermal shield
4. Internal layout of the equipment for the cold box; best possible arrangement of components so as to make good use of the space inside cold box
5. Design and layout plan for piping of all components
6. Flexibility analysis of piping system
7. Design, analysis and optimization of support structures inside cold box for all equipment
8. Making a model of the vacuum chamber, thermal shield, internal component layout, piping arrangements and support systems in a 3D modelling software.

Chapter 2.

Literature Review

This thesis has the primary aim of designing the cold box of the LHe plant at IPR. Therefore, the literature review commences from the study of the existing plant at IPR. Further literature have been studied for material requirement for all parts of the cold box, design procedure, fabrication aspects of cold chamber, piping prerequisites, thermal shield design and theory about Multilayer Insulation.

2.1 Existing HRL at IPR[1]:

Pradip Panchal, Ritendra Bhattacharya, B Sarkar, A K Sahu, et.al., have described the He cryogenic system for the SST-1 at IPR and it has four major units, namely; helium refrigeration/liquefier (HRL), warm gas management system (WGM), integrated flow distribution and control system (IFDCS) and current feeder system (CFS).

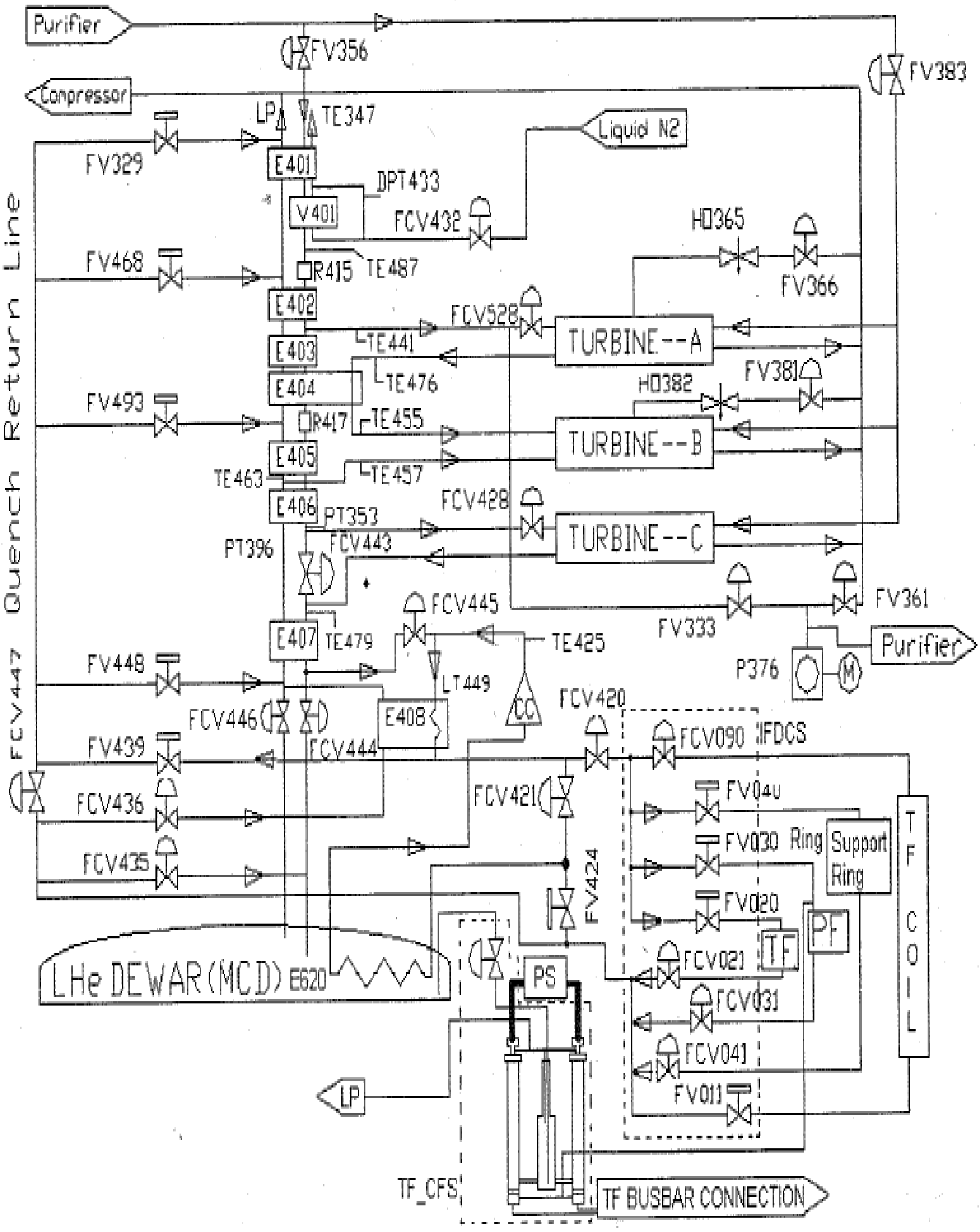
The HRL is a custom made system and has refrigeration capacity of 650 W at 4.5 K and simultaneous liquefaction capacity of 200 l/hr for current leads. Cold circulation (CC) pump provide 300 g/sec SHe at 4 bar and 4.5 K in a closed loop with SCMS. Major components of HRL are:

i. Compressor station,
ii. Oil removal system;
iii. Purifier system and
iv. Cold box with main control dewar (MCD).

Each subsystem is considered as standalone system equipped with individual programmable logic controller (PLC) to control the operarions.

2.1.1 Cold box with main Control Dewar (MCD):

The cold box consists of 8 numbers of very highly effective heat exchangers, 3 high speed turbo expanders, one liquid nitrogen vaporizer, cold circulator and Joule-Thompson valves for the main cycle. The liquefaction process used in the HRL is the modified Claude cycle. The entire cooling process in the HRL has been described previously in chapter 1.3. An expanded and more descriptive schematic diagram of the existing LHe plant is given in the following diagram.



(Figure 2: Schematic of cold box showing main components within it)

2.2 Need of vacuum in HRL[10]:

According to Phil Danielson, due to the fact that, vacuum has almost non existing thermal conductivity, it is one of the best methods of insulating a system. Vacuum greater than 10^{-3} mbar is required to provide for adequate insulation for processes at cryogenic temperatures and to improve the performance of MLI. Roughing vacuum pump like Rotary pump and high vacuum pump like diffusion pump are necessary to generate a vacuum environment of 10^{-3} mbar. Mechanical oil sealed rotary vane type of pump is generally used for roughing operation (up to 10^{-2} mbar) and turbo-molecular or oil diffusion pump is used to produce high vacuum (upto 10^{-3} mbar).

2.3 Shape Selection[3]:

According to ***Ken Harrison, VP of Engineering for the GNB Corporation in "Engineering a Better Vacuum Chamber";*** vacuum chambers have been built in many a shapes. The main concern while selecting the shape is to control deflection. An intrinsically strong shape or a weak shape with stiffening elements can keep deflection in limit.

To minimize deflection and material, spherical shape is the best shape. Also it is the best from maintaining cleanliness and evacuation point of view. But forming costs of these shapes are too high. Also many applications do not fit in. Ellipsoid shapes are also hard to build. The next best shape is cylindrical vessel with domed ends. Box shapes have the least rigidity.

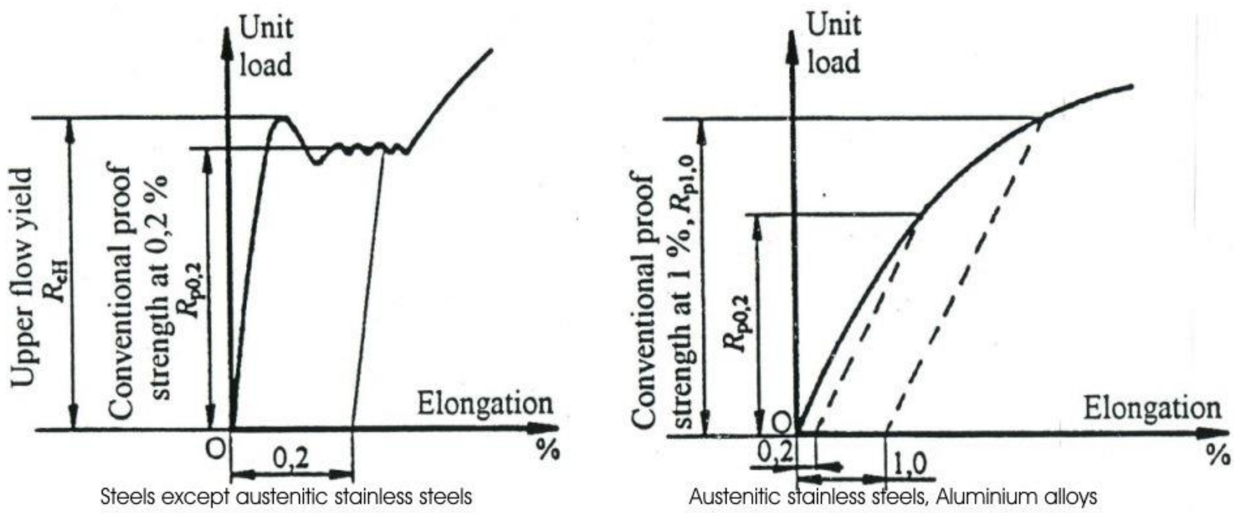
Rigidity of shapes	
Sphere	Most rigid
Ellipsoid	
Cylinder with domed ends	
Cone	
Geodesic ball (triacontrahedron)	
Diamond (octahedron)	
Cylinder with flat ends	
Pyramid (tetrahedron)	
Box	Least rigid

2.4 Materials and their properties:

The selection of materials for the components of the system is one of the most crucial stages of entire designing work. While choosing a material, function performed by the component takes precedence in consideration. Appropriate material, that has a suitable set of properties for the specified function of the component, is then looked for. Availability and material cost are then taken into account for the final selection.

2.4.1 Stainless steel for Vacuum chamber body:

According to ***C. Hauviller, CERN, Geneva, Switzerland*** in ***Design rules for vacuum chambers[4];*** while choosing a material for the vacuum chamber, wide discussions occur about all the properties of it. But, three main properties should be taken into account, namely, the modulus of rigidity (and Young's modulus), elastic limit and rupture limit. A simple analysis of traction or compression test is a common practice.



(Figure 3: Traction curve of steels)

Other parameters like creep data, fatigue limit, fracture toughness could be useful for specific design but usually complicate the choice of material.

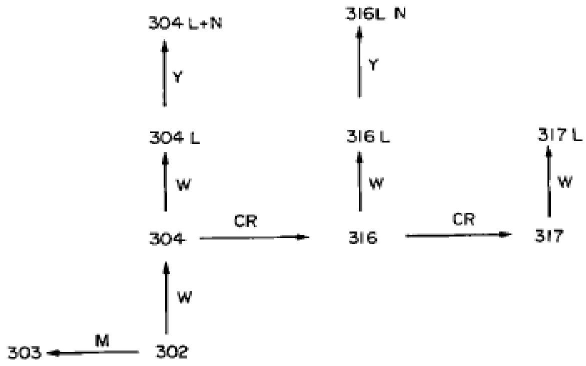
Among technological properties, a vacuum chamber should be absolutely free from leaks. To obtain a good leak proof chamber, the weladability of the material must be excellent; otherwise, it leads to the use of sophisticated and expensive techniques. Availability and cost criterion are the last to consider. It is strongly recommended to use materials of general use in industrial applications.

According to ***Ken Harrison, VP of Engineering for the GNB Corporation in "Engineering a Better Vacuum Chamber[3]",*** porous and dirty materials must be avoided. Common castings can be problematic. He has recommended using stainless steel for many purposes. Particularly, for cost compromise, SS should be used for components that are exposed to vacuum and mild steel and aluminium for other components.

According to ***George Behrens, William Campbell, Dave Williams, Steven White*** in "***Guidelines for the Design of Cryogenic Systems"[2],*** stainless steel is recommended to be used in vacuum applications, because at high vacuum SS304 does not oxidize. Also it can be heated to very high temperatures for bake out to reduce outgassing. SS304 is used because it can be easily electro-polished to provide clean and contamination free surface. Stainless steel has very good weldability, hence can be welded with TIG welding that is necessary for producing high and ultrahigh vacuum.

According to ***C Geyari*** in ***"Design considerations in the use of Stainless Steel for Vacuum and Cryogenic Equipment"[18];*** stainless steel of composition 18/8 (18% Chrome and 8% Nickel) is the best option for cryogenic and vacuum applications. A film of very small thickness is formed on the surface of any alloy that has chrome composition more than 13%; which is impermeable to most corrosive media. The film renews itself in the presence of oxidizing agents. Nickel is used to improve the corrosion resistance in weakly oxidizing media and to maintain the austenitic structure of steel. Nickel also helps in improving weldability and ductility of steel. 18/8 stainless steel is inert, passive and it has very low outgassing rate, which is 1×10^{-9} torr l s^{-1} cm^{-2} when degreased after 4 hours and 4×10^{-12} torr l s^{-1} cm^{-2} after bake out.

AISI (American Iron and Steel Institute) grade steels are used globally. As for vacuum and cryogenic applications, mechanical strength, weldability, corrosion resistance and machinability are the main properties to look for. In AISI family of steel, ***C Geyari*** has given the following comparison of different grades of steel.



(Figure 4: Comparison of different grades of steel)

C Geyari writes, the problem attached with welding of steel in vacuum and cryogenic application is micro cracks developing in the carbide rich zone and intercrystalline disintegration at very low temperatures. These problems are due to the phenomenon of carbon precipitation. This problem can be tackled in three major ways.

i. Selecting an alloy with very low carbon content
ii. Selecting a stabilized alloy
iii. Welding with minimum heat input.

Also, to increase the yield strength of steel following methods must be used:

i. Addition of nitrogen to the alloy
ii. Cold stretching
iii. A combination of the above two points.

2.4.2 Material for "O" ring Gasket:

According to ***Phil Danielson,*** in ***"Sealing material require a careful choice"***[13]***;*** there are basically three kinds of gasket material in use.

i. Elastomers
ii. Metallic gaskets
iii. A mixture of both the above gasket materials

Elastomers are organic polymers that have excellent elastic resilience which allows them to deform under compression to fill all curviness and waviness but with enough elasticity to prevent permanent deformation. Commonly used elastomers are Butyl, Buna-N, Fluoroelastomers, and Perfluoroelastomers. Elastomers have the disadvantage of increasing the gas load by outgassing and atmospheric permeation. Outgassing rate can reduced considerably by baking the ring upto 150^{o}c before installation. Also, permeation from atmosphere can be reduced by using two concentric "O" rings without any pump out space. These have the advantage of easy installation and good performance as far as sealing is considered. On the other hand, metallic gaskets have the advantage of no outgassing and no permeation of gas from atmosphere. But, metal gaskets are very hard hence less resilient. These cannot be reused. Hence, metallic gaskets are expensive. It is extensively advised to use elastomers, especially fluoroelastomers, for vacuum applications; but with proper baking. Viton is generally used in vacuum applications and cryogenic temperatures. The outgassing rate of Viton is given in the following table:

Vacuum Material	Outgassing Rate (torr litre/ sec/ cm)
Viton (Unbaked)	8×10^{-7}
Viton (Baked)	4×10^{-8}

(Table 1: Outgassing rate of Viton in baked and unbaked condition)

2.5 Design procedure and Fabrication aspect of vacuum chamber[5,6,7]:

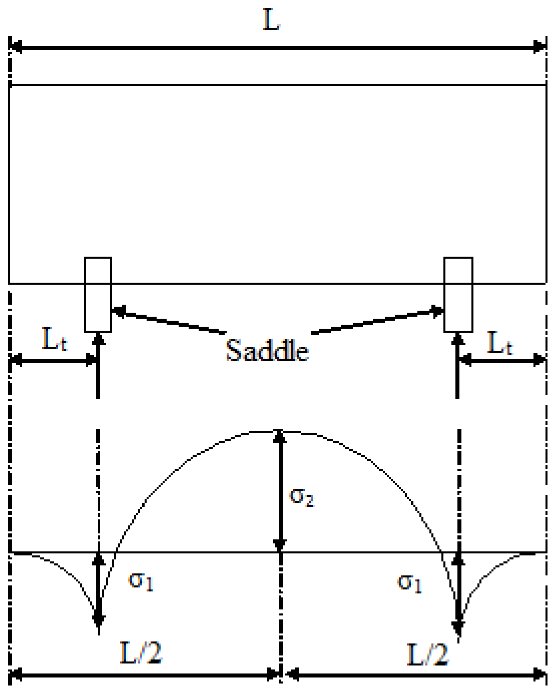
Many procedures and techniques for designing of pressure vessels have been developed for proper design, ensuring least cost and safety with required strength throughout the vessel length. These procedures are put into developing standards for vessel designing and these are used globally. In general, the ASME (American Standard for Mechanical Engineers), section VIII, Boiler and Pressure Vessel Code is used for vessel designing in many industries and laboratories.

According to ***Robert C. Gentliger*** and ***Kirk E. Christensen***[5], while designing for vacuum vessel of cold box, ASME Boiler and Pressure Vessel Code, section VIII should be followed as a guideline. First step is to choose a nominal thickness for the vessel under external pressure of 1 bar, using procedure given in UG-28 of the code. This paragraph of the pressure vessel code also provides design criteria for stiffening rings, which can be used to keep the thickness of vessel minimal but enough strength to withstand the external pressure. Next step is to design the manholes or openings, whose thickness can be found out by the same procedure followed for the pressure vessel. Paragraph UG-36 and UG-37 can be used to determine the opening size and reinforcement requirement. The last step for vessel design is the flange design for the openings and main vessel itself. The procedure for flange design can be followed from appendix-2 of the ASME code.

Welding takes utmost importance in vessel design. Welding is used to attach flanges, stiffening rings and saddle supports to the main body of vacuum vessel. Welding design specifications can be followed from UW-13 of ASME code. Moreover, welding practice should meet the standards of high vacuum application, satisfy code requirements and should be free from any virtual leak. Finally, pressure relief devices should be designed. These are incorporated in case of accidental pressurization inside the cold box or rupture of any pipe carrying coolant at cryogenic temperature.

According to ***Lloyd E. Brownell*** and ***Edwin H. Young*[16]*;*** horizontal vessels are supported on saddle supports. These vessels behave as simply supported beams. Zick has developed equations for the different stresses developed in the vessel and the saddle supports. These equations have constants determined experimentally. The different stresses developing in the vessel body as well as saddle supports are:

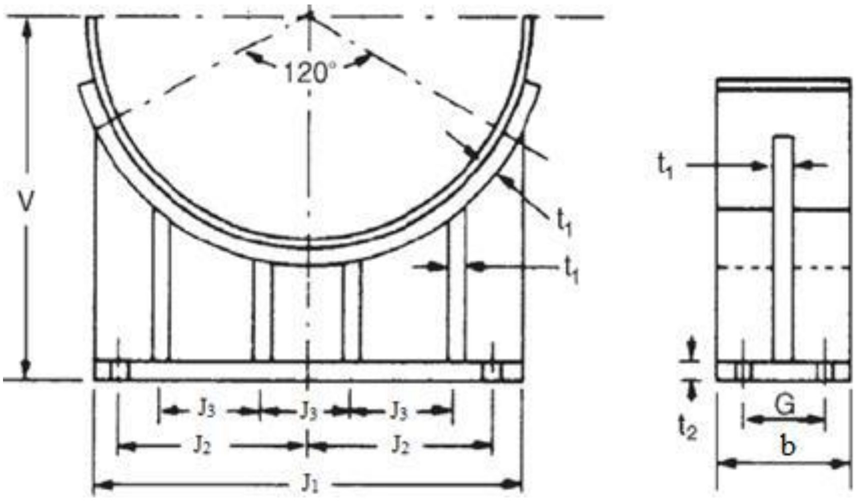
i. Longitudinal stress
ii. Tangential shear stress
iii. Circumferential stress at horn of saddle
iv. Additional stresses in the head or cover functioning as stiffener.



(Figure 5: Bending moment diagram of vessel on saddle support)

As a rule, the saddles should be positioned at a length not more than the radius of the vessel. To obtain stiffening effect from the head or cover plates, the saddle should be kept at length not more than 20% of the total length.

According to ***Gavin Towler*** and ***Ray Sinnott*** in ***"Chemical Engineering Design – Principles, Practice and Economics of plant and Process Design"*[20]*;*** standard saddles should be chosen according to vessel diameter and weight of the vessel. And finally, these dimensions should be checked against the stresses that will be coming, using Zick's formulae. The standard saddle sizes are given in the following table and figure.



(Figure 6: Notation for different dimensions for the saddle)

Vessel Diameter (m)	Maximum Weight (kN)	Dimensions (m)						Dimensions (mm)			
		V	b	J_1	J_2	J_3	G	t_2	t_1	Bolt Dia	Bolt holes
0.6	35	0.48	0.15	0.55	0.24	0.19	0.095	6	5	20	25
0.8	50	0.58	0.15	0.7	0.29	0.225	0.095	8	5	20	25
0.9	65	0.63	0.15	0.81	0.34	0.275	0.095	10	6	20	25
1	90	0.68	0.15	0.91	0.39	0.31	0.095	11	8	20	25
1.2	180	0.78	0.2	1.09	0.45	0.36	0.14	12	10	24	30
1.4	230	0.88	0.2	1.24	0.53	0.305	0.14	12	10	24	30
1.6	330	0.98	0.2	1.41	0.62	0.35	0.14	12	10	24	30
1.8	380	1.08	0.2	1.59	0.71	0.405	0.14	12	10	24	30
2	460	1.18	0.2	1.77	0.8	0.45	0.14	12	10	24	30
2.2	750	1.28	0.225	1.95	0.89	0.52	0.15	16	12	24	30
2.4	900	1.38	0.225	2.13	0.98	0.565	0.15	16	12	27	33
2.6	1000	1.48	0.225	2.3	1.03	0.59	0.15	16	12	27	33
2.8	1350	1.58	0.25	2.5	1.1	0.625	0.15	16	12	27	33
3	1750	1.68	0.25	2.64	1.18	0.665	0.15	16	12	27	33
3.2	2000	1.78	0.25	2.82	1.26	0.73	0.15	16	12	27	33
3.6	2500	1.98	0.25	3.2	1.4	0.815	0.15	16	12	27	33

(Table 2: Dimensions of saddle)

Saddle Angle (Degrees)	K_3	K_4	For $L_t/Ri>1$		For $L_t/Ri<0.5$		K_8	K_9
			K_6	K_7	K_6	K_7		
120	1.16	0.88	0.053	0.053	0.053	0.013	0.4	0.76
150	0.78	0.48	0.032	0.032	0.032	0.008	0.3	0.675

(Table 3: Value of empirical constants)

2.6. Piping Design and Layout[8]:

According to ***Mohinder L. Nayyar*** in ***"Piping Handbook";*** a proper piping layout is very necessary for any kind of system set up. Proper layout provides the advantage of non-interference with any equipment of the system. Also, a well analysed pipe routing leaves very less possibility of fabrication faults and it serves to control quality as well. The designer must understand the following aspects before going for piping design:

i. The system P&ID (Process and Instrumentation Diagram) which shows all equipment, their location, valves, free space etc. in sequence.
ii. Pipe dimensions, schedule or pipe thickness, loads on all pipes, material of pipe.
iii. Dimensions of all equipment and available space for pipe routing
iv. The project general arrangement or equipment location defining the interface and interfacing elements.

2.7. Thermal Shield and MLI:

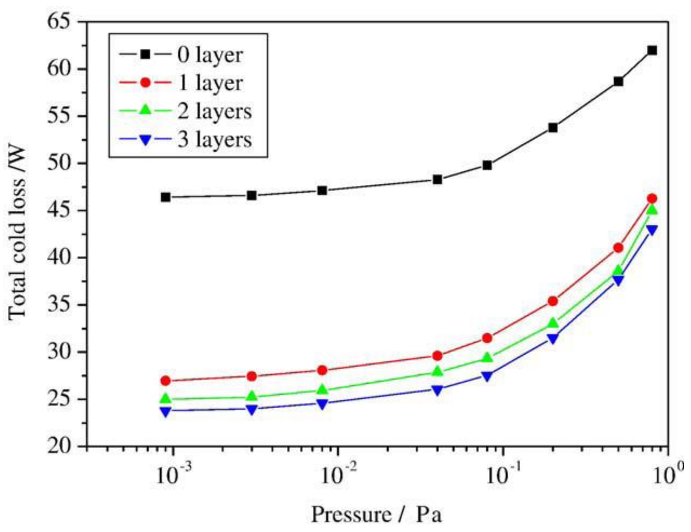
According to ***George Behrens, et. al.*** in ***"Guidelines for the Design of Cryogenic Systems"[2];*** thermal insulation shield must be built of a material that has a high value of thermal conductivity and low value of emissivity. Aluminium and Copper are the best options for the material of thermal shield. Properties of aluminium and copper at 70 K are shown below:

	Aluminium	Copper
Thermal conductivity (W/cm-K)	2.5	5
Emissivity	0.018-0.7	0.006-0.78

(Table 4: Properties of Aluminium and copper at 70 K)

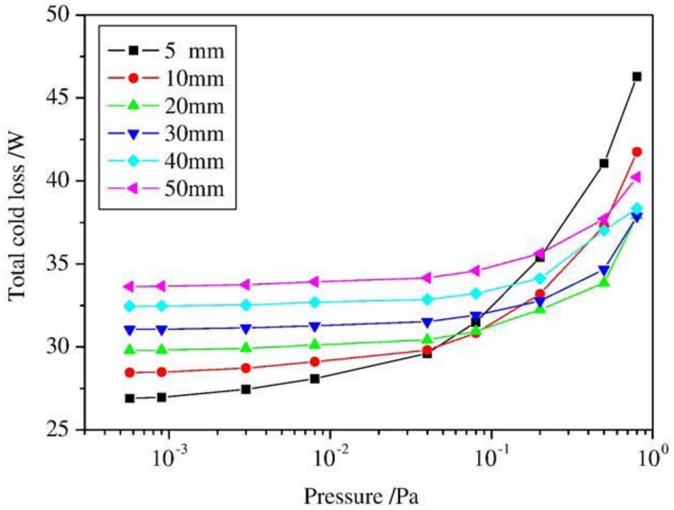
Thermal shield is used to reduce radiation effect from 300 K dewar walls to 15 K equipment walls. This is done by providing 70 K radiation intercepts, which are highly conductive. These act as thermal shields, which capture the radiation from 300 K walls and dissipate it out of the system. However there are chances of radiation load coming from these 70 K stations to the equipment. Therefore, it is necessary that the shield material must have high conductivity and low emissivity.

According to ***Feng Yu, Yanzhong Li and Yinhai Zhu*** in ***"Numerical and Experimental Investigation on Thermal Insulation Performance of low temperature Cold box"[21];*** cold box is generally used to store the high quality cold energy intact by reducing heat conduction and convection from atmosphere. Cold loss from the cold box occurs due to the combined effect of coupled conduction and radiation. Performance of the cold box can be improved by using thermal insulation shields. The authors have conducted experiment that shows thermal shield performance at different pressures, for different shield layers and different shield locations. Cold loss is less for very low value of pressure and gradually increases with increase in pressure.



(Figure 7: Total cold loss vs pressure for different no. of layers of thermal shield)

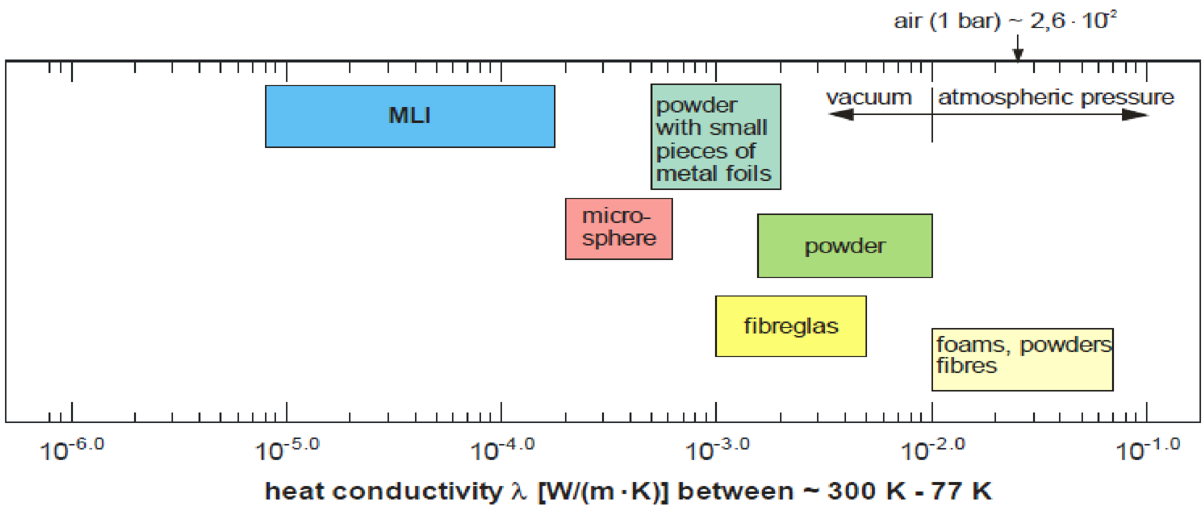
The above graph shows that cold loss substantially decreases when pressure is low. Again, with increase of number of layers of thermal shield, cold loss value decreases. But, adding more layers of shield to improve insulation performance becomes less effective when shield number is greater than 3. Hence, maximum number of thermal shield layers must not exceed 3. Also, the authors have investigated the dependence of cold loss with relative positioning of shield with internal cavity. The following graph illustrates the results.



(Figure 8: Total cold loss vs pressure for different position of shield)

The above figure shows that cold loss value remains very low when thermal shield is positioned very close to internal cavity and it increases with relative distance of shield, at very low pressure as low as high vacuum.

According to ***Thomas M. Flynn*[22]*;*** MLI or Multi Layer Insulation is used to prevent all kinds of modes of heat transfer to the cold box. MLI is composed of layers of alternate low emitting radiation shields separated by low conductivity spacers. These are so designed that, the layers touch only at a few discrete points, reducing conductive heat transfer. For cryogenic applications, generally 6 μm thick Aluminium shield are used as radiation reflecting materials. And for spacer materials glass fibre spacer, namely, Tissueglas or Dexiglas are used. A comparison for different insulating materials is given in the following diagram:



(Figure 9: Range of thermal conductivity of different insulating materials at temperature between 300 K and 77 K)

Chapter 3.

Design Procedures and Results

3.1 Introduction:

The cold box is a cylindrical vacuum chamber, which contains all equipment to generate liquid Helium. Design of the vacuum chamber is one of the most important objectives of the work. The design of requires an estimation of length and diameter of the chamber. This can be done only when the component layout inside the chamber has been perfectly defined. Hence, the project work starts with the internal component layout decision. Then after, cold box vacuum chamber is designed, including all parts of the vessel, such as, stiffening ring, gasket, cover plate etc. After these works have been completed, piping layout is done along with stress analysis of all pipes. Support system for the entire system is designed finally.

3.2 Internal Component Layout:

Equipment layout inside the cold box holds utmost importance in designing of the cold box. The major components housed inside the chamber are 8 heat exchangers, 3 turbo expanders, J-T valves, one liquid Nitrogen vaporizer, cold circulator, ejector pump and one liquid Helium bath to produce SHe. Besides the main equipment, pressure transducers, temperature measuring devices and other instrumentation are also present. The components cannot be arranged in a haphazard manner, as faulty design costs the whole system in the form of lapse in efficiency. That is why; internal component design must be done very carefully. The following important points were kept in consideration while designing the internal component layout.

i. Arrangement should be such that, there exists a temperature gradient along the length of the cold box. So that, the entire cold box length can be considered to be comprising of two parts; one 20 K part and another 4 K part.
ii. Maintain the temperature gradient helps in reducing the thermal shield length, i.e. thermal shield can be put in the 4 K side only.
iii. The layout should be used effectively, i.e. no space should be left unused or misused.
iv. The layout should be flexible to provide enough paths for process piping to be arranged properly; i.e. there should be sufficient space for the piping to go for without failure.
v. As the cold box houses all the support structures, there should be enough space at appropriate places to support every component easily.

vi. Space must be left blank for personnel to go inside the box for any kind of repairing or maintenance work.

vii. Arrangement must be done in such a way that the component assembly and disassembly can be done easily.

viii. Components should be kept close their actual neighbours in the process, to reduce piping length and hence frictional losses.

3.2.1 Dimensions of components given by IPR:

The dimensions of all components including their headers are given below:

Sl. No.	**Name of component**	**Dimension (in mm)**	**Size of header for heat exchanger (in mm) (Headers are semi cylindrical in shape unless mentioned)**			
			Hot stream		**Cold stream**	
			Inlet	**Outlet**	**Inlet**	**Outlet**
1	HE1	(1800*600*520)*2	Φ240*520	Φ144*520	Φ480 (semi spherical)	Φ480 (semi spherical)
2	LN_2 Vaporizer	950*300*250	Φ150*250	Φ150*250	Φ270*250	Φ270*250
3	HE2	1300*600*520	Φ190*600	Φ140*600	Φ500 (semi spherical)	Φ500 (semi spherical
4	HE3	500*400*500	Φ140*400	Φ140*400	Φ370 (semi spherical)	Φ370 (semi spherical)
5	HE4	1100*400*500	Φ140*400	Φ140*400	Φ370 (semi spherical)	Φ370 (semi spherical)
6	HE5	1100*400*250	Φ180*250	Φ140*250	Φ240*250	Φ240*250
7	HE6	1100*350*200	Φ180*200	Φ180*200	Φ300*200	Φ300*200
8	HE7	1100*350*200	Φ180*200	Φ180*200	Φ300*200	Φ300*200
9	Charcoal Adsorber 1	Φ500(1000+200)				
10	Charcoal Adsorber 2	Φ350(500+150)				

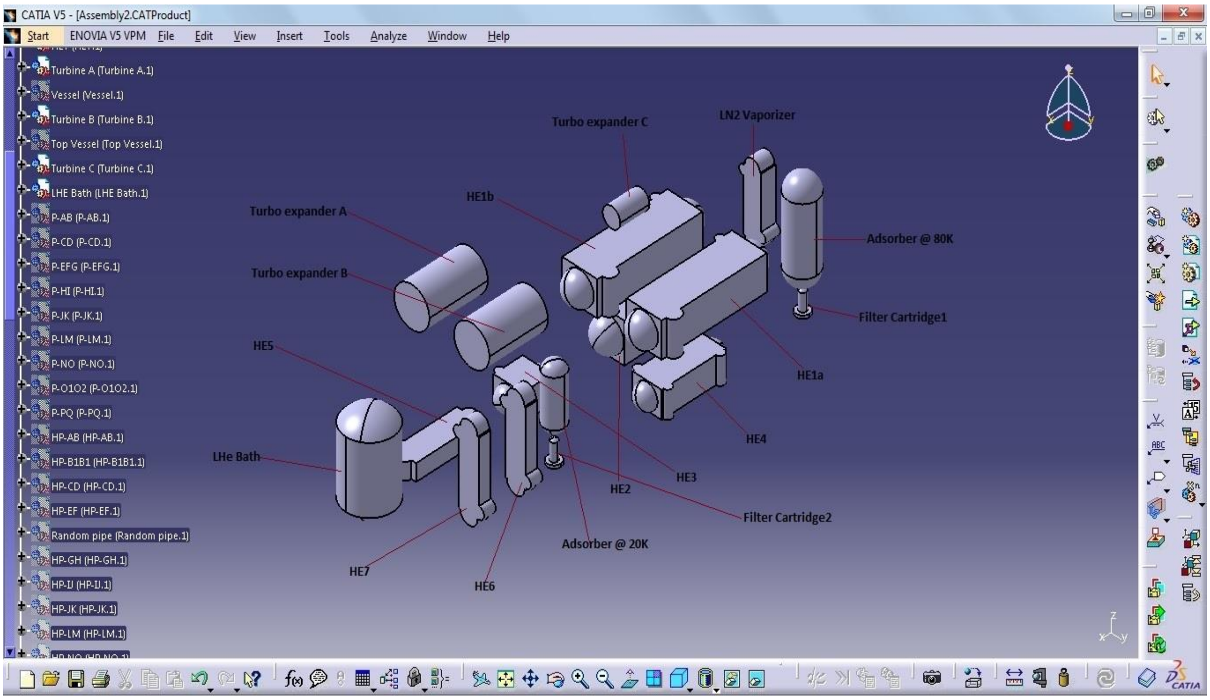
11	Turbo expander A	Φ600*1000				
12	Turbo Expander B	Φ600*1000				
13	Turbo Expander C	Φ300*500				
14	LHe Bath	Φ800(1000+200)				

(Table 5: Dimensions of all components and headers)

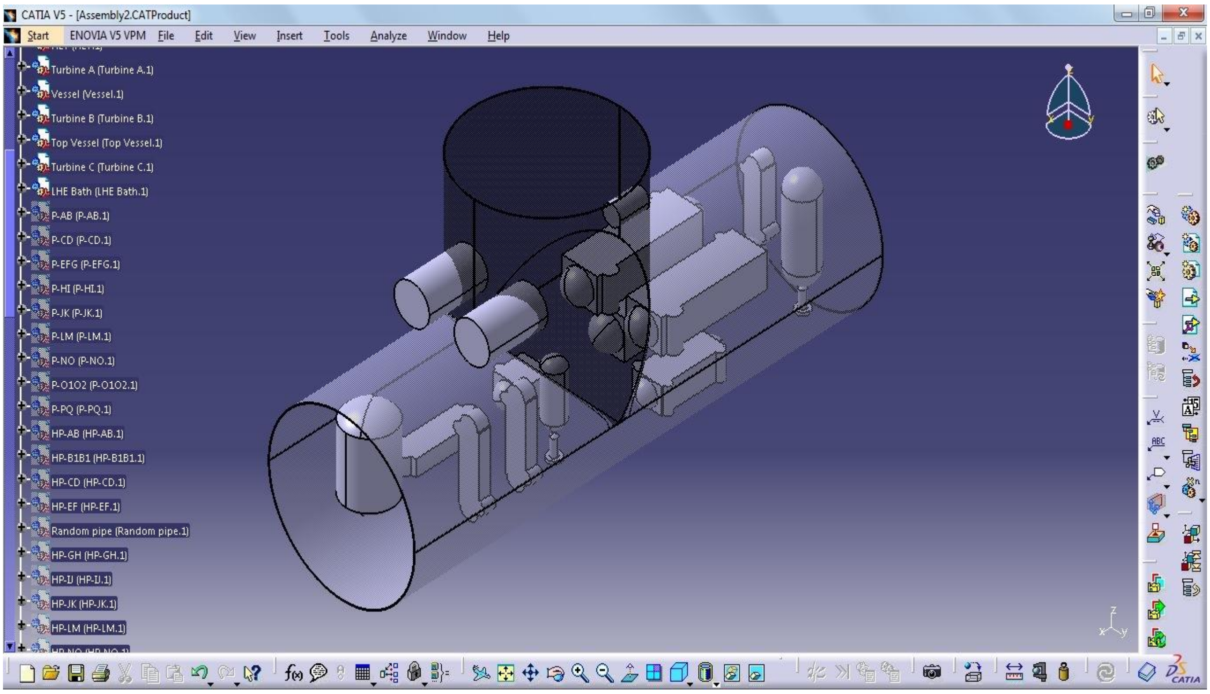
3.2.2 3D Modelling of Internal Component Layout:

The cold box along with all the constituent elements is modelled in 3D modelling software CATIA. Layout of the components is carefully done keeping in consideration all the above points.

The internal component layout looks like:



(Figure 10: Internal Component Layout along the length of the Cold Box)



(Figure 11: Internal Component Layout with Transparent Cold Box Vacuum Chamber)

3.3 Design Procedure of Vacuum Chamber and Results:

A cylindrical vessel under external pressure or a vacuum vessel has an induced circumferential compressive stress because of external pressure equal to twice the longitudinal compressive stress because of external pressure effects alone. Under such condition the vessel may collapse because of elastic instability caused by the circumferential compressive stress. The collapsing strength of such vessels may be increased by the use of uniformly spaced, internal or external circumferential stiffening rings. Form the standpoint of elastic stability such stiffener have the effect of subdividing the length of the shell into subsections equal in length to the centre-to-centre spacing of the stiffeners.

Long, thin cylinders without stiffeners or with stiffeners spaced beyond a “critical length” will buckle at stresses below the yield point of material. The corresponding critical pressure at which buckling occurs is a function only of the ratio of cylinder thickness (t) and diameter (Do), and the modulus of elasticity, E, of the material. If the length of the shell with closures Ls, or the distance between circumferential stiffeners, Ls, as the case may be, is less than the critical length, the critical pressure at which collapse occurs is a function of the Ls/Do ratio as well as of the Do/t ratio and E.

Design of vacuum vessel for the cold box follow procedure mentioned below:

i. Selection of material for all parts of cold box
ii. Design of cylindrical shell
iii. Design if stiffening ring
iv. Design of gasket
v. Design of bolt
vi. Design of flanges
vii. Design of cover plates or heads
viii. Design of saddle supports and
ix. Design of manholes

3.3.1 Material Selection:

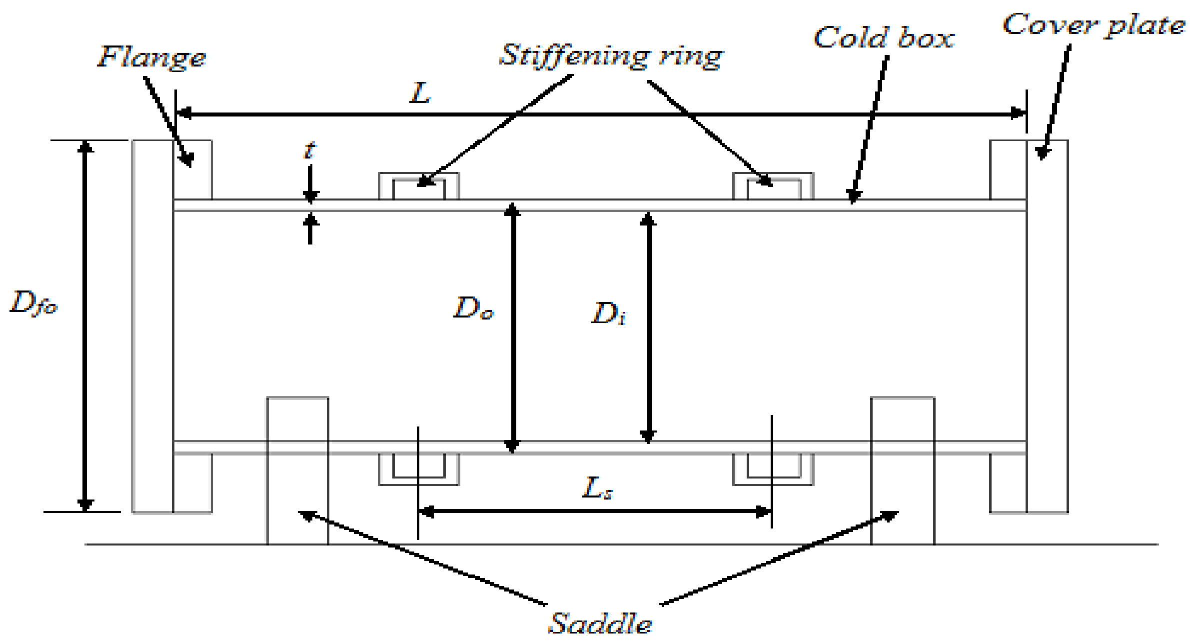
As explained in chapter 2; among many materials that are applicable in vacuum applications, stainless steel of 18/8 (18% chrome and 8% nickel) or SS304L shows a favourable set of properties like low outgassing rate, high corrosion resistance, very high yield strength and good weldability. Besides, SS304L is aptly available. Hence, for cold box body SS304L is the best material. Mild steel can be used for support system. Aluminium and copper are used as materials for heat exchanger body and thermal shield body.

One of the most important considerations is the material for gasket. In vacuum applications, “O” ring gaskets are used. The material for “O” ring gasket must have low outgassing rate and optimum resilience to seal the required equipment. It should be elastic enough to regain its original shape after the flange is removed, and plastic enough to flow into microscopic imperfections to seal. Outgassing rate of elastomer gaskets can be too high that exceeds the entire outgassing rate of other components. Properly treated gasket should be installed in vacuum applications. In general, elastomers, such as, Buna-N, Butyl and Fluoroelastomer Viton are used as gasket material. Viton has very low outgassing rate and less permeability to atmospheric gases with high compressive strength. Hence, Viton is the best choice for gasket material for the cold box.

Besides, saddle can be made up of stainless steel, not necessarily of 304 type; because, strength is the only desired property for saddle support. Piping material should have high thermal conductivity. Hence, SS is the best choice for this.

3.3.2 Design Procedure for Cylindrical Shell:

Elastic stability is main design parameter considered. The calculations are made by successive approximations, by using the following method as per UG-28 of Boiler and Pressure Vessel Code (BPVC), Section VIII, Division I, issued by American Society of Mechanical Engineers.



(Figure 12: Schematic diagram of the cold box vacuum chamber)

i. Estimate total length (L) and outside diameter (Do) of the vessel so as to meet the space requirement of internal component layout and provision for repair and maintenance work.
ii. Estimate number of stiffening rings and accordingly, calculate the critical length (Ls).
iii. Assume a thickness value (t).
iv. Determine the ratios “Do/t” and “Ls/Do”.
v. Find out the value of factor A, using the value of the ratios from figure G of subpart 3 of BPVC, section II, part D, issued by ASME.
vi. Using value of factor A, find out value of factor B from material chart of figure HA-3, in subpart 3 of BPVC, section II, part D, issued by ASME for the specified modulus of elasticity of SS304L.
vii. Calculate the allowable external pressure as follows:

If A falls to the left of material line,

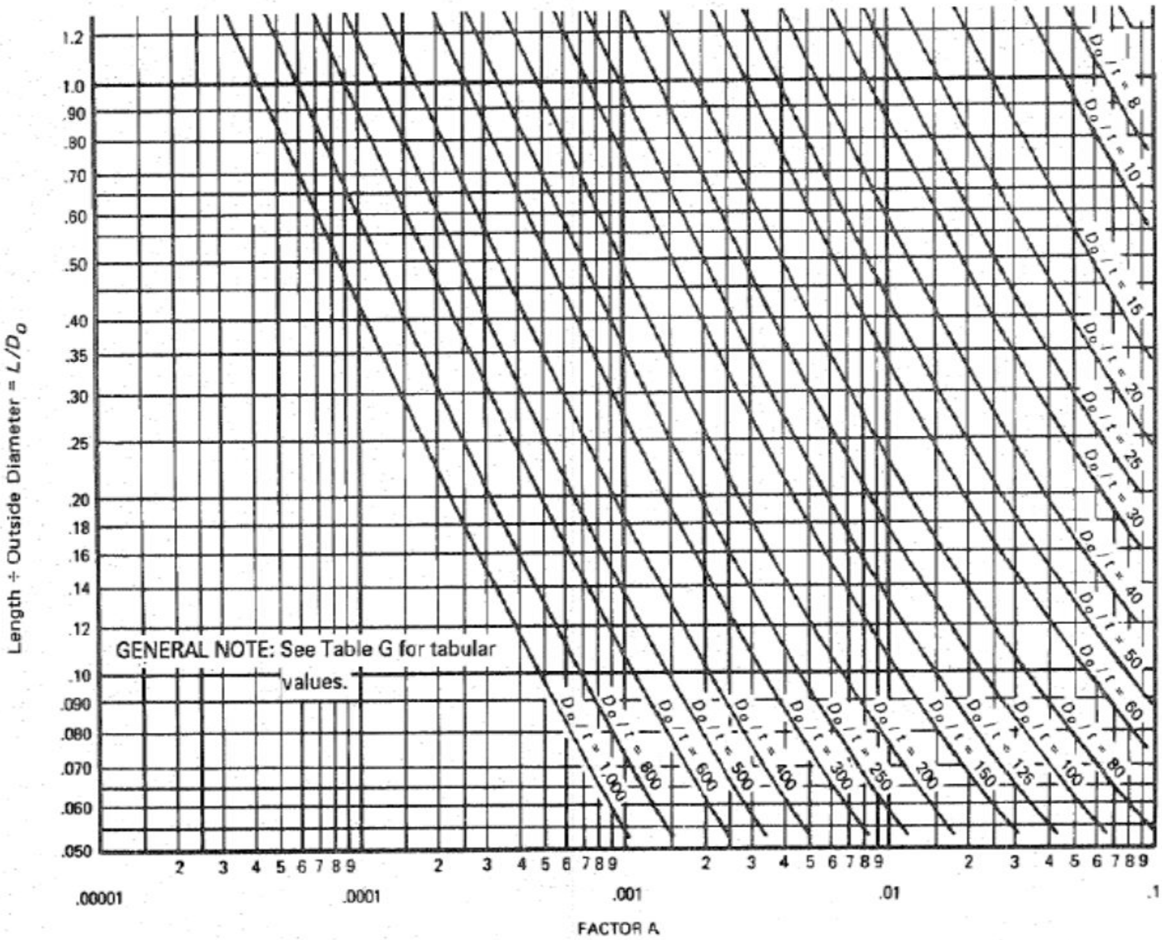
$$Pa = \frac{2AE}{(\frac{Do}{t})}$$

If A falls on the material line,

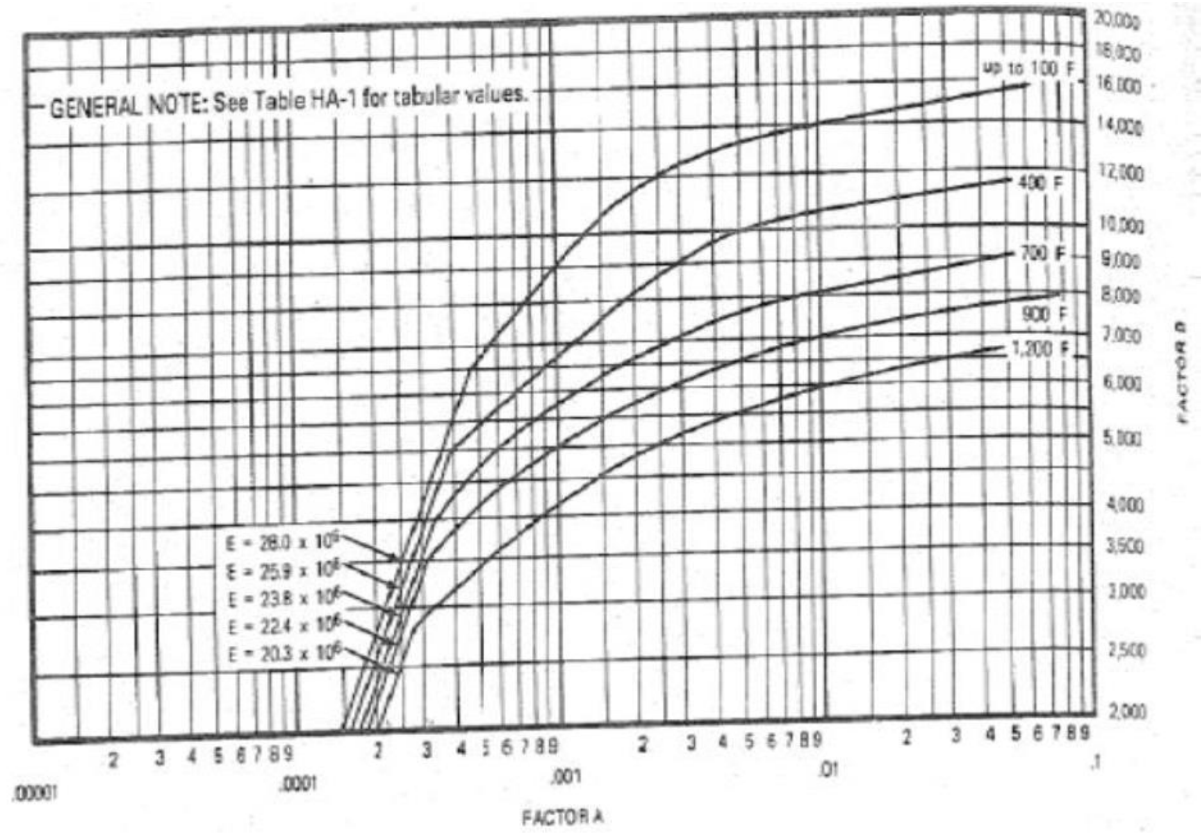
$$Pa = \frac{4B}{3\left(\frac{Do}{t}\right)}$$

viii. Compare the calculated value of "P_a" with external design pressure "P". If "P_a" is smaller than "P", select a larger value of "t" and repeat the procedure until "P_a" is greater than "P".

ix. Find out the Collapsing pressure, Pc = 4P. If, Pc>Pa, reduce "Ls" by increasing the no. of stiffeners and repeat the procedure until, Pc<Pa.



(Figure 13: Plot in BPVC, section II, Part D, to find factor A)



(Figure 14: Plot in BPVC, Section II, Part D, to find Factor B)

3.3.2.1 Sample Calculation:

Cold box material = SS304L

Estimated length = L = 8000 mm = 314.96 inch

Estimated outside diameter = D_o = 2500 mm = 2.5 m = 98.42 inch

External pressure = P_o = 1013 mbar = 14.94 psi

Internal pressure = P_i = 10^{-5} mbar

Modulus of elasticity = E = 28.5* 10^6 psi

Assumed thickness = t = 15 mm

Assumed no. of stiffening rings = n = 0

So, maximum unsupported length = L_s = 8000 mm

Now, $\frac{L_S}{D_O} = \frac{8000}{2500} = 3.2$

And, $\frac{D_O}{t} = \frac{2500}{15} = 166.667$

Using the plots from ASME, we get, A = 0.002 and B = 2800 psi

As this point falls on the material line,

$$Pa = \frac{4B}{3\left(\frac{Do}{t}\right)} = \frac{4*2800}{3*166.667} = 22.4\,psi$$

Now, comparing, 22.4 psi > 14.94 psi

Hence, this thickness is safe and can be used for further design.

Another thickness assumed = t = 13 mm

Assumed no. of stiffeners = n = 0

Maximum unsupported length = L_s = 8000 mm

Now, $\frac{L_S}{D_O} = \frac{8000}{2500} = 3.2$

And, $\frac{D_O}{t} = \frac{2500}{13} = 192.307$

Using the plots from ASME, A = 0.00016 and B = 2100 psi

As this point falls on the material line,

$$Pa = \frac{4B}{3\left(\frac{Do}{t}\right)} = \frac{4*2800}{3*192.307} = 14.56\,psi$$

Now, comparing, 14.56 psi < 14.94 psi

Hence, this thickness is not safe without stiffening rings. So, either thickness should be increased or no. of stiffening rings should be increased.

Similarly, the optimization process is carried on in MS-EXCEL.

Some of the best results obtained are shown below:

Sl. No.	Thickness (t) (mm)	No. of stiffening rings (n)	Maximum unsupported length (L_s) (mm)
1	16	2	3000
2	14	3	2000
3	12	4	1600
4	15	4	3000
5	12	5	1333.34

(Table 6: Best results obtained)

3.3.3 Design Procedure for Stiffening Ring:

The required moment of inertia of a circumferential stiffening ring can be calculated by the following procedure as per UG-28 of Boiler and Pressure Vessel Code (BPVC), Section VIII, Division I, issued by ASME:

i. Select a section to be used for the stiffening ring and find its cross sectional area "As" specified in Bureau of Indian Standard (BIS).

ii. Find factor B.

$$B = 0.75\frac{PDo}{t + \frac{As}{Ls}}$$

iii. Using the previous chart, find the value of factor "A".

iv. If, "A" can't be found out by chart, use the formula:

$$A = \frac{2B}{E}$$

v. Find out the required moment of inertia of the stiffening ring using the formula:

$$Is = \frac{Do^2 LsA\left(t + \frac{As}{Ls}\right)}{14}$$

vi. If "Is" is greater than the available moment of inertia for the section selected from BIS, a new section with a larger moment of inertia must be selected. If "Is" is smaller than the inertia for the section selected in step 1, that section should be satisfactory.

3.3.3.1 Sample Calculation:

Material for stiffening ring = SS304L

Let, vessel dimensions be; t = 13 mm, n = 3, L_s = 2000 mm, P = 14.94 psi, D_o = 2500 mm.

Let us selected section from BIS = MC75

Given: I = 785000 mm^4

A_s = 9.1 cm^2

Now, A_s/L_s = 0.455 mm and [t + (A_s/L_s)] = 13.455 mm

So, B = $0.75\frac{PDo}{t+\frac{As}{Ls}}$ = 2081.673 psi

And, A = 0.000111

Now, $Is = \frac{Do^2 LsA\left(t+\frac{As}{Ls}\right)}{14}$ = 1333486.607 mm^4

Comparing, I_s > I; so, MC75 section is not safe to use.

Now, another section from BIS is selected = MC100

Given: I = 1920000 mm^4

A_s = 12.2 cm^2

Using above formula, B = 2057.966 psi and A = 0.000109.

Now, $Is = \frac{Do^2 LsA\left(t+\frac{As}{Ls}\right)}{14}$ = 1324544.643 mm^4 < I

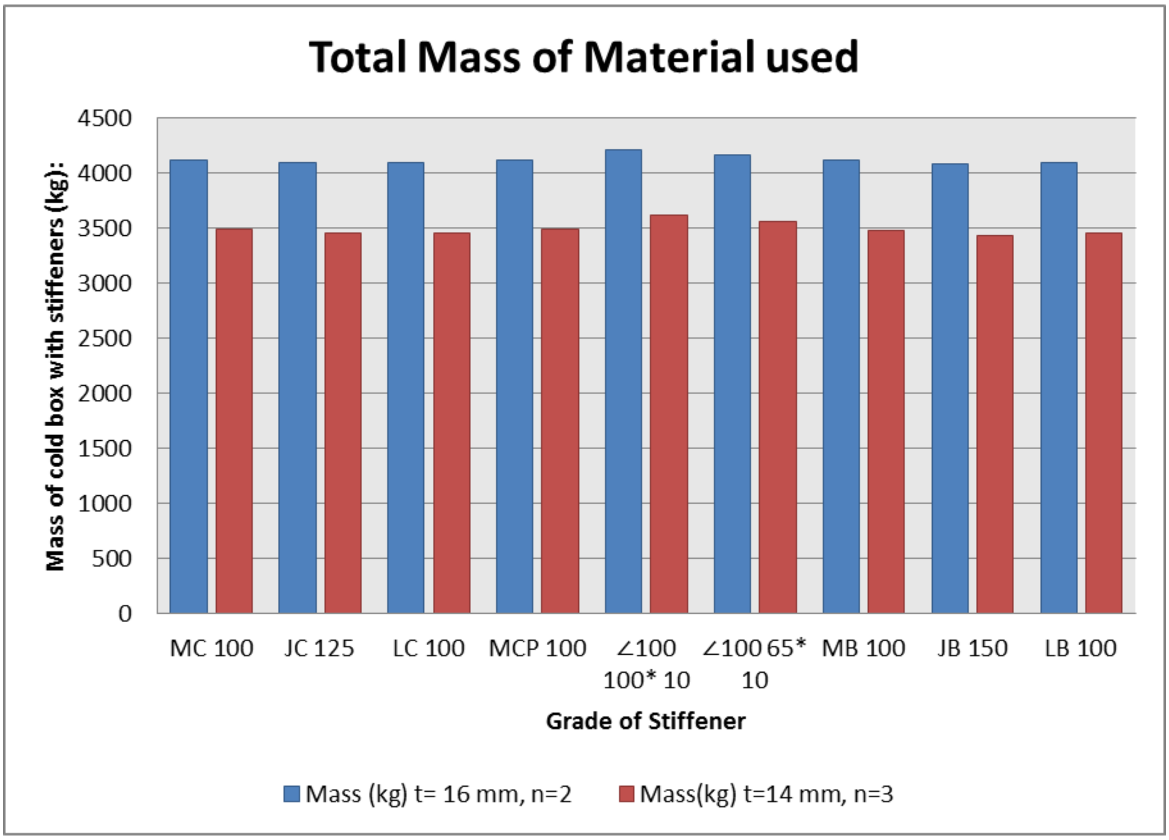
So, MC100 Section is safe to use and can be used for further calculations.

Similarly, using different sections are checked using above procedure.

The following are the sections that proved to be safe for our design:

MC 100, MCP 100, ∠100 65*10, LB 100, LC 100, JC 125, MB 100, JB 150.

An analysis for mass requirement for the cold box and stiffening ring is done for two specified cold box dimensions and different grades of rings. The results are as follows:



(Figure 15: A comparison of different grades of stiffening rings for two specified dimensions of cold box for material requirement)

It is concluded that, cold box thickness 14 mm with 3 stiffening rings of grade JB 150 uses the minimum material, thereby reducing material cost. Besides, LB 100 and JC 125 grades also give minimal use of material.

3.3.4 Design Procedure for "O" ring gasket:

As explained earlier, the gasket material should have enough plasticity to flow into the irregularities of flanges and provide a leak proof joint. Also, elasticity of this material should be enough to regain its original shape when load is removed, so that it can be reused time and again. The bolts must be tightened regularly, to ensure leak proof junction. Hardened gasket cannot be reused ones it is removed.

Elastomer Viton is used for gasket material in vacuum and cryogenic applications. Because Viton is an elastomer, the minimum design seating stress value and gasket factor are zero. The following procedure is followed, while designing "O" ring gasket:

i. The diameter at which the "O" ring is to be mounted must be selected. This diameter (D_{gm}) must be greater than the outside diameter of the vessel.
ii. Cross-sectional diameter (D_g) must be chosen from standard. If standard diameter is not given, it can be calculated considering the compressive failure of gasket because of external atmospheric pressure.
iii. Once, D_g is chosen, the groove dimensions can be found by using empirical formulae.
    Depth of groove = d = 0.72 D_g
    Width of groove = w = 1.15 D_g

3.3.4.1 Sample Calculation:

Let, Mounting diameter = D_{gm} = 2700 mm
Then, cross-section selected = D_g = 14 mm
So, depth of groove = d = 0.72 * 14 = 10.08 mm ~ 10 mm
Width of groove = w = 1.15 * 14 = 16.1 mm ~ 16 mm

Again, another mounting diameter = D_{gm} = 2600 mm
Let, Cross-section selected = D_g = 12 mm
Then, depth of groove = d = 0.72 * 12 = 8.64 mm
Width of groove = w = 1.15 * 12 = 13.8 mm

"O" ring design is also an iterative procedure. The groove and ring dimensions should be such that there is no leak from external atmosphere.

3.3.5 Design procedure for Cover plates:

Often used cover plates are flat head type. These are used as closures at all the openings of the vessel. For flat heads, blind flanges, and cover plates, the minimum thickness may be determined by following equation:

$$t_c = D_{gm}\sqrt{\frac{CP}{\sigma_a}}$$

Where, t_c = thickness of the cover plate or head

D_{gm} = Mounting diameter of the “O” ring

P = external atmospheric pressure

σ_a = Maximum allowable stress for the cover material

C = A constant for the type of enclosure and type of attachment with the flange, from UG-34 of ASME, section VIII, division I.

= 0.3 for flat heads and bolted joints.

3.3.5.1 Sample Calculation:

Let, D_{gm} = 2600 mm

We know, P = 1013 mbar = 1.013 bar = $1.013 * 10^5$ Pa = 0.1013 Mpa

σ_a = 115.15 Mpa = 115.15 N/mm^2

so now, thickness of the cover plate = $t_c = D_{gm}\sqrt{\frac{CP}{\sigma_a}}$

$$= 2600\sqrt{\frac{0.3\times0.1013}{115.15}} = 42.23 \text{ mm}$$

The thickness can be taken to be 42.5 mm, or for safety, 45 mm.

Here, C = 0.3 for flat cover plate and bolted joint.

Let, another, D_{gm} = 2700 mm

Then, $t_c = D_{gm}\sqrt{\frac{CP}{\sigma_a}} = 2700\sqrt{\frac{0.3*0.1013}{115.15}}$ = 43.86 mm

The thickness can be taken to be 45 mm.

Hence, 45 mm thickness of cover plate gives a safe design.

3.3.6 Design Procedure for Saddle Support:

Horizontal cylindrical pressure vessels are commonly supported by saddle supports. The saddle must be designed to withstand the load imposed by the weight of the vessel and its contents. The dimension of saddle support is first chosen from a standard available for particular vessel diameter. And then stress produced by the saddle of this dimension is checked against several stresses coming up, by using Zick's empirical relations for several stresses. Those stresses are:

1) *Longitudinal Bending Stress*:

Maximum longitudinal bending stress comes at the middle of the saddle, which is:

$$\sigma_2 = \frac{3K_2QL}{\pi R_i^2 t}$$

Where, Q = Total weight per saddle

L = Total length of cold box shell

R_i = Inner radius of the vessel

t = Thickness of the vessel

All units are to be taken in F.P.S. Unit.

Here,

$$K_2 = \frac{1 + 2\left[\frac{R_i^2 - h^2}{L^2}\right]}{1 + \frac{4h}{3L}} - \frac{4L_t}{L}$$

Where, L_t = Distance from tangent line to saddle

h = Depth of head

For flat cover head, h=0

For safe design, $\sigma_2 \leq 0.8\sigma_a$

2) *Tangential Shear stress:*

a) Unstiffened shell with saddles away from head:

Tangential shear stress occurs at horn of the saddle.

It is given by the equation:

$$\sigma_3 = \frac{QK_3}{R_i t}\left[\frac{L - h - 2L_t}{L + h}\right]$$

Where, all notations carry their usual meaning.

For safe design, $\sigma_3 \leq 0.8\ \sigma_a$

b) Shell stiffened by head:

When the saddle supports are located near the head, the tangential shear stresses are first carried from the saddle to the head. Then the load is transferred back to the head side of the saddle by tangential shear stresses which act on an arc of angle larger than the angle of contact of the saddle.

Tangential shear stress at the head is:

$$\sigma_4 = \frac{QK_4}{R_i t_c}$$

Tangential shear stress at the shell is:

$$\sigma_5 = \frac{QK_4}{R_i t}$$

Where, t_c = thickness of cover plate. and

K_3 and K_4 depend upon saddle angle.

For safe design, $\sigma_4 \; and \; \sigma_5 \leq 0.8\sigma_a$

3) *Circumferential stress at horn of saddle:*

a) Unstiffened shell:

Stress induced can be found out by:

when $L \geq 8R_i$

$$\sigma_6 = -\frac{Q}{4t(b + 1.56\sqrt{R_i t})} - \frac{3K_6 Q}{2t^2}$$

When $L < 8R_i$

$$\sigma_6 = -\frac{Q}{4t(b + 1.56\sqrt{R_i t})} - \frac{12K_6 QR_i}{Lt^2}$$

Where, all notations carry their usual meaning.

b) Shell stiffened by head:

When the shell is stiffened by the head, the shear stresses are carried across the saddle to the head, and then load is transferred back to the saddle. The circumferential bending moment is smaller in the shell stiffened by the head than in the unstiffened shell. Circumferential stress at horn of saddle for this condition is given by following formula:

When $L \geq 8R_i$

$$\sigma_7 = -\frac{Q}{4t(b + 1.56\sqrt{R_i t})} - \frac{3K_7 Q}{2t^2}$$

When $L < 8R_i$

$$\sigma_7 = -\frac{Q}{4t(b + 1.56\sqrt{R_i t})} - \frac{12K_7 Q R_i}{Lt^2}$$

Where, all notations carry their usual meanings.

K_6 and K_7 values depend upon saddle angle.

For safe design,

$$\sigma_6 or\ \sigma_7 \leq 1.25\sigma_a$$

4) *Additional stress at head used as stiffener:*

The stiffness of the head is often utilized by locating the saddles near the heads. When the saddle is close to the head, the horizontal component will cause tension across the entire height of the head as if the head were a flat disk. The maximum stress induced in the head by the horizontal component of the tangential shear is:

$$\sigma_8 = \frac{QK_8}{R_i t_c}$$

Where, all notations carry their usual meanings.

For safe design, $\sigma_8 \leq 1.25\sigma_a$

5) *Wear plate ring compression in shell over saddle:*

There are forces acting on the shell band directly over the saddle causing ring compression in the shell band. The ring compression stress may be reduced by attaching a wear plate somewhat larger than the surface of the saddle to the shell directly over the saddle. The ring compression stress can be calculated by:

$$\sigma_9 = \frac{QK_9}{t(b + 1.56\sqrt{R_i t})}$$

Where, all notations carry their usual meanings.

b = A dimension of the saddle given in figure no.

For safe design,

$$\sigma_9 \leq 0.5\sigma_a$$

3.3.6.1 Sample Calculation:

Outside diameter of the vessel = D_o = 2500 mm

So, the following dimensions from the standard (Table no.2) is chosen:

V = 1.48 m	J_3 = 0.59 m
b = 0.225 m	G = 0.15 m
J_1 = 2.3 m	t_2 = 16 mm
J_2 = 1.03 m	t_1 = 12 mm

For, a saddle angle of 120^o, the following empirical constants are chosen (Table no.3):

K_3 = 1.16	K_4 = 0.88
K_6 = 0.053	K_7 = 0.013
K_8 = 0.4	K_9 = 0.76

It is a general practice to place the saddle support at not more than 20% of the total length. That way, the heads can be used as stiffeners. So, we take it to be 15%.

Hence,

L_t = 0.15 * 8 = 1.2 m = 47.24 inch

$$\frac{L_t}{R_i} = \frac{1.2}{2.47} = 0.485 < 0.5$$

So, the heads can be used as stiffeners.

Q = $\frac{\pi}{4}[(2.5^2 - 2.47^2) \times 8 + 2.5^2 \times 0.03] \times 7920 \times 9.81 = 84228.20\, N$ = 18950 pound

$$K_2 = \frac{1 + 2\left[\frac{R_i^2 - h^2}{L^2}\right]}{1 + \frac{4h}{3L}} - \frac{4L_t}{L} = \left[1 + \frac{2 \times 2.47^2}{8^2}\right] - \frac{4 \times 1.2}{8} = 0.59$$

Now,

i. Maximum longitudinal stress:

$$\sigma_2 = \frac{3K_2QL}{\pi R_i^2 t} = \frac{3 \times 0.59 \times 18950 \times 314.96}{\pi \times 97.24^2 \times 0.59} = 602.76\, psi$$

$$\sigma_a = 120 \times 145 = 17400\, psi$$

So, 0.8 * 17400 = 13920 psi > 602.76 psi

Hence, the saddle chosen is safe.

ii. Tangential shear stress:

Tangential shear stress at the head is,

$$\sigma_4 = \frac{QK_4}{R_i t_c} = \frac{18950 \times 0.88}{97.24 \times 1.77} = 96.88\, psi < 13920 \text{ psi}$$

$96.88\, psi$ < 13920 psi

Tangential shear stress at the shell is,

$$\sigma_5 = \frac{QK_4}{R_i t} = \frac{18950 \ \times 0.88}{97.24 \times 0.59} = 290.667 \, psi$$

$$290.667 \, psi < 13920 \, psi$$

Hence, the chosen saddle is safe.

iii. Circumferential stress at horn of saddle:

$8R_i$ = 8 * 2.47 = 19.76 m > L

So, $\sigma_7 = -\frac{Q}{4t(b+1.56\sqrt{R_i t})} - \frac{12K_7 Q R_i}{Lt^2}$

$$= -\frac{18950}{4\times0.59(8.85+1.56\sqrt{97.24\times0.59})} - \frac{12\times0.013\times18950\times97.24}{314.96\times0.59^2}$$

= -388.54-2621.91

= -3010.45 psi

Now, 3010.45 psi < 1.25 σ_a

Hence, the chosen saddle is safe.

iv. Additional stress in head used as stiffener:

$$\sigma_8 = \frac{QK_8}{R_i t_c} = \frac{18950 \times 0.4}{97.24 \times 1.77} = 44.04 \, psi$$

Now, 44.04 psi < 1.25 σ_a

Hence, the chosen saddle is safe.

v. Wear plate ring compression in shell:

$$\sigma_9 = \frac{QK_9}{t\left(b + 1.56\sqrt{R_i t}\right)}$$

$$= \frac{18950 \times 0.76}{0.59\left(8.85 + 1.56\sqrt{97.24 \times 0.59}\right)}$$

$$= 2065.844 \, psi$$

$$0.5\sigma_a = 0.5 \times 17400 = 8700 \, psi$$

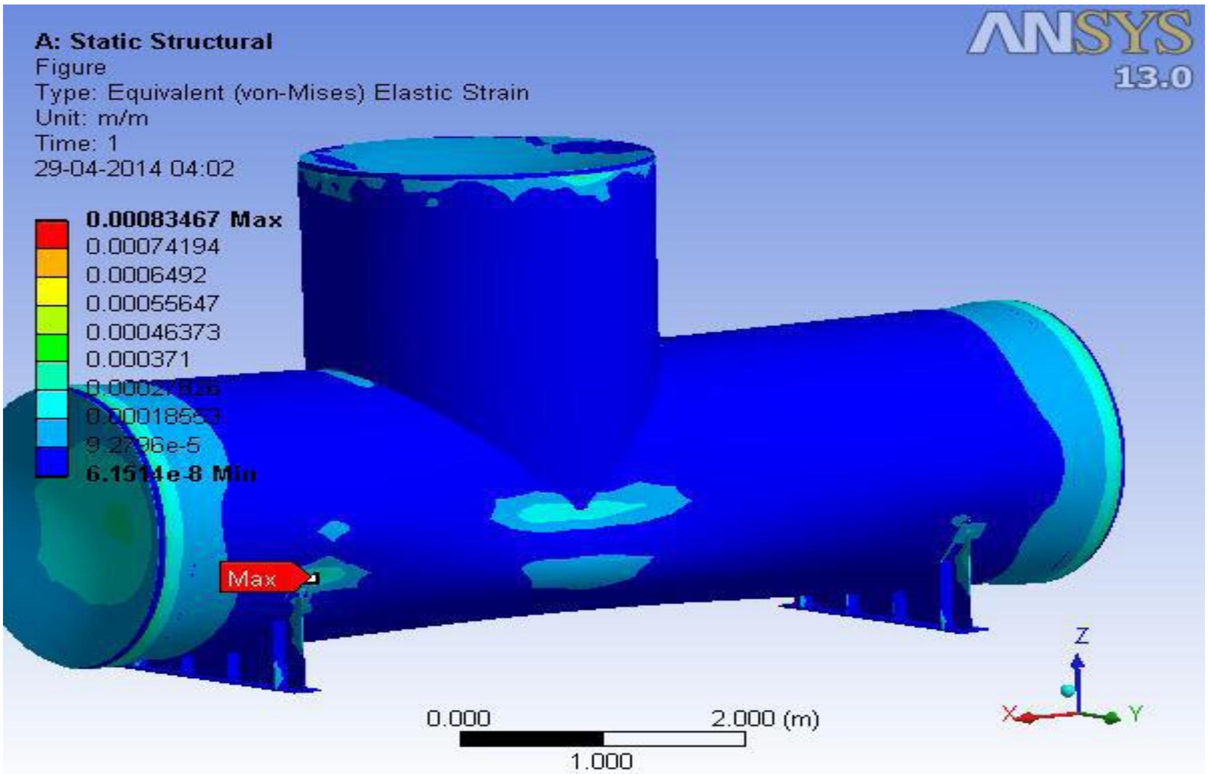
So, 2065.844 psi < 8700 psi

Hence, the chosen saddle is completely safe.

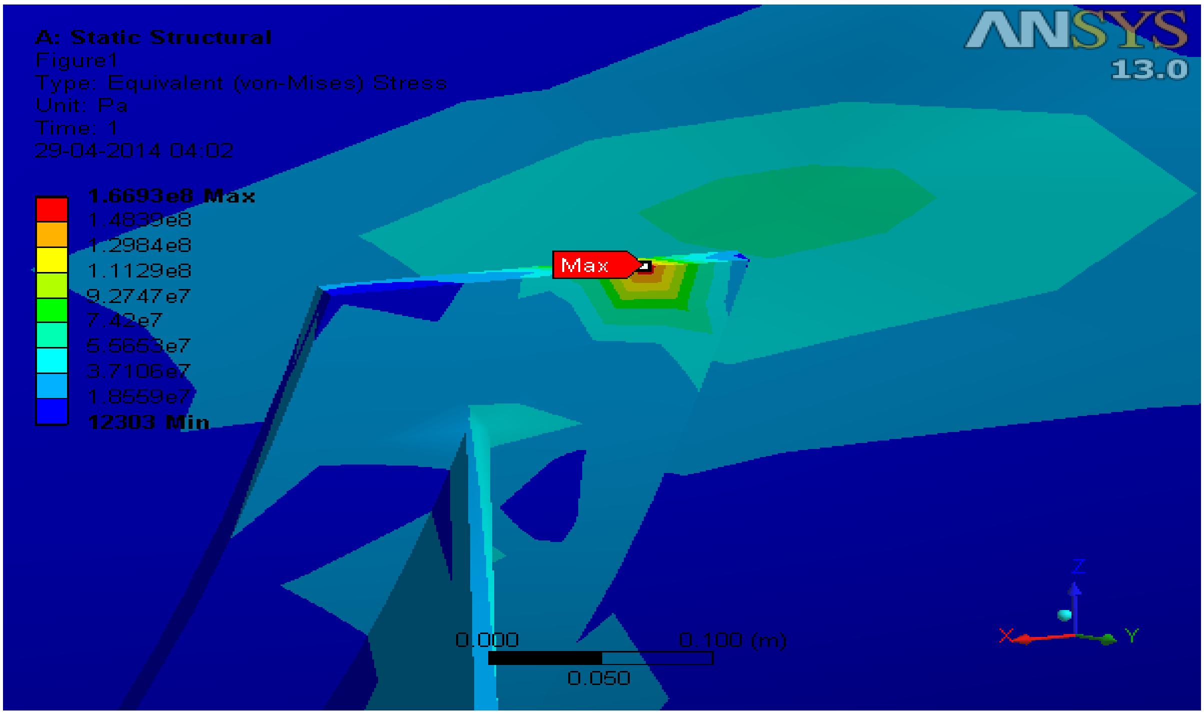
So, it is demonstrated that, all the stresses are within limit, using Zick's formulae. Hence, the saddle support chosen from standard is safe.

3.4 ANSYS analysis of the cold box:

The cold box is drawn and analysed in ANSYS. The following results were obtained:

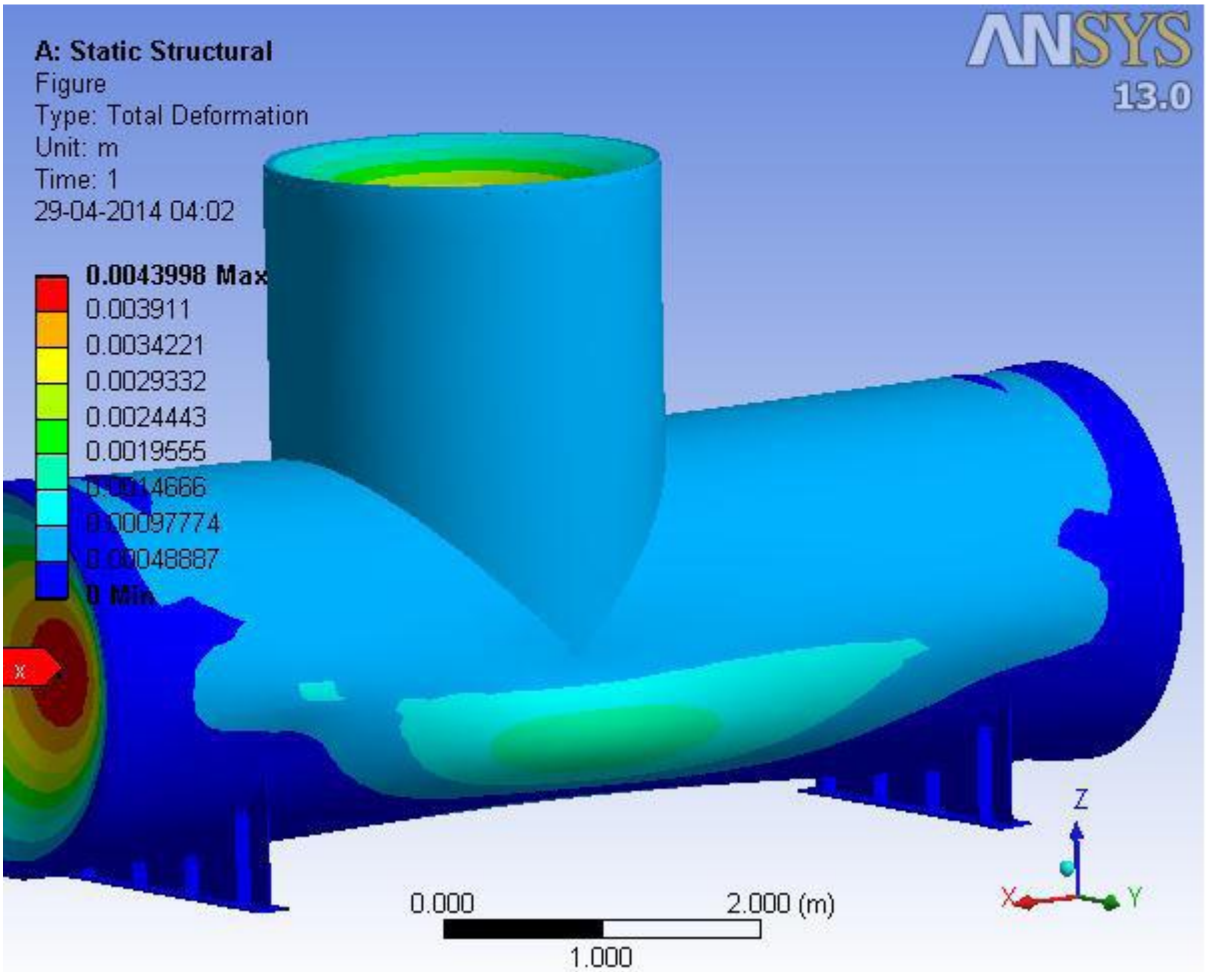


(Figure 16: Equivalent (Von-Mises) elastic strain of the cold box)



(Figure 17: Zoomed view of maximum stress)

It is shown in the above diagram that maximum stress occurs at the junction between saddle support and cold box. But this maximum value of stress is also less than the yield strength of SS304L. so this design is safe.



(Figure 18: Total deformation)

It is shown in the above diagram that, maximum deformation occurs at the centre of the cover plates. Maximum deformation is not more than 4 mm. this deformation can also be avoided by using bending resisting sections or beams.

Chapter 4.

Piping Analysis & Support System Design

4.1 Piping Analysis:

Piping and flexibility analysis can be divided in two parts. The first one is design step. In this step the approximate lengths of all pipes are estimated. Then possible frictional losses are calculated to finally find out required diameter for the specified temperature and pressure condition. The second part is the flexibility analysis, which is most important in cryogenic applications.

4.1.1 Design Procedure:

In pipe flow, there are two basic kinds of head losses; major head loss and minor head loss. Major head loss constitute the main share of the overall head loss. This is due to frictional losses throughout the length of the pipe. And minor losses are due to uneven sections in the pipe, entrance, exit, larger cross-section or smaller cross-section coming in the way of the flow. The magnitude of minor loss is very small compared to major losses. Hence, while designing pipes, we consider only major head losses due to friction.

Before designing the pipes, temperature and pressure conditions are needed to be known. So that, viscosity and other properties can be found out.

Head loss in a tube is given by the following formula:

$$h_f = \frac{4fLv^2}{D * 2g}$$

Where, f = Fanning's friction factor.

= 16/Re (for Re < 2000)

= $0.079/Re^{0.25}$ (for 2000 < Re < 10^6)

L = length of the pipe

v = velocity of flow

D = Diameter of the tube (because the cross-section is circular)

g = Acceleration due to gravity = 9.81 N/m^2

The following procedure is adapted to design the pipes:

i. Consider the flow to be laminar.
ii. Note down he temperature and pressure conditions and the corresponding properties.
iii. Estimate the approximate length of the pipe.

iv. Find out the diameter of the pipe by the following formula derived from head loss formula:

$$d = \left[\frac{128\mu L m}{\pi\rho\Delta P}\right]^{0.25}$$

v. Using the diameter, find out Reynold's number to check whether it falls in the laminar region.

vi. If, Reynold's number falls in the turbulent region, find the diameter by the formula:

$$d = \left[\frac{0.241 * L * m^{1.75} * \mu^{0.25}}{\rho\Delta P}\right]^{0.2105}$$

4.1.1.1 Sample Calculation:

Allowable pressure drop = ΔP = 0.1 mbar/m

Mass flow rate for the low pressure line = m = 90 g/s

By above mentioned procedure, flows along all lines are found to be turbulent.

So, considering the LP line from HE7 to HE6:

Estimated length = L = 2 m

Property values are obtained from "Hepack" added in MS-EXCEL.

Density = ρ = 11.982 kg/m^3

Viscosity = μ = 1.54*10^{-6} Pa-s

So, diameter of the line can be found out by,

$$d = \left[\frac{0.241 * L * m^{1.75} * \mu^{0.25}}{\rho\Delta P}\right]^{0.2105}$$

$$= \left[\frac{0.241 \times 2 \times 0.09^{1.75} \times 1.54 * 10^{-6^{0.25}}}{11.982 \times 0.2}\right]^{0.2105}$$

= 0.05512 m

= 55.12 mm

Hence, for this particular piping, nominal diameter of 50 mm can be used with schedule 5s.

In this manner the diameters of all the pipes including low pressure pipes, high pressure pipes and pipes going to the turbines are calculated.

4.1.2 Flexibility Analysis:

Piping flexibility analysis is an important design consideration because the large difference between ambient and cryogenic temperature will result in significant pipe shrinkage. The effects of the thermal contraction of pipe and fittings as a result of system operating temperature changes cannot be overlooked during the layout and routing of any pipe system. The function of piping flexibility or stress analysis is for the most part delegated to the computer particularly in the case of low temperature piping systems.

Cryogenic piping is routed so that the piping configuration provides adequate flexibility. This means that the pipe is routed so that there are Z, L and U bends, to take up the pipe thermal contraction while keeping the stresses within the allowable range. When additional flexibility is required, flexibility can be increased by the addition of expansion loops (U bends).The addition of flexible metal hose and bellows should be minimum as these can lead to leaks after certain cycles of operation.

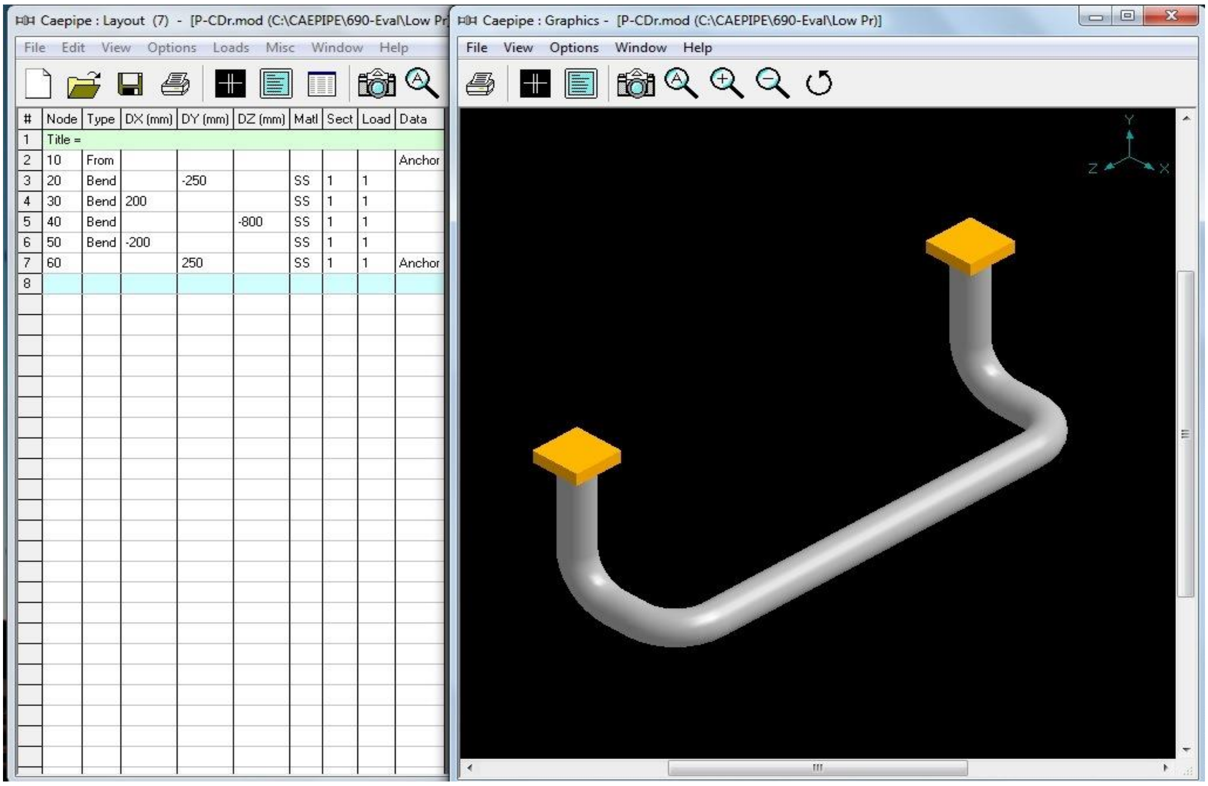
Codes and standards usually set forth minimum requirements for design, materials, fabrication, erection, test, and inspection of piping systems. For cryogenic systems, there are some codes and standards, but not well established like for room temperature piping system. In most of the cryogenic process, the severity occurs at room temperature as the material has lower strength at this temperature compared to that at cryogenic temperature. So, ASME B31.3 can be used for design along with piping flexibility analysis for low temperature thermal contraction. This ASME code also lists piping materials by ASTM specification number. The allowable stress as a function of design temperature is listed for each material.

Flexibility and stress analysis is carried on by using piping analysis software CAEPIPE.

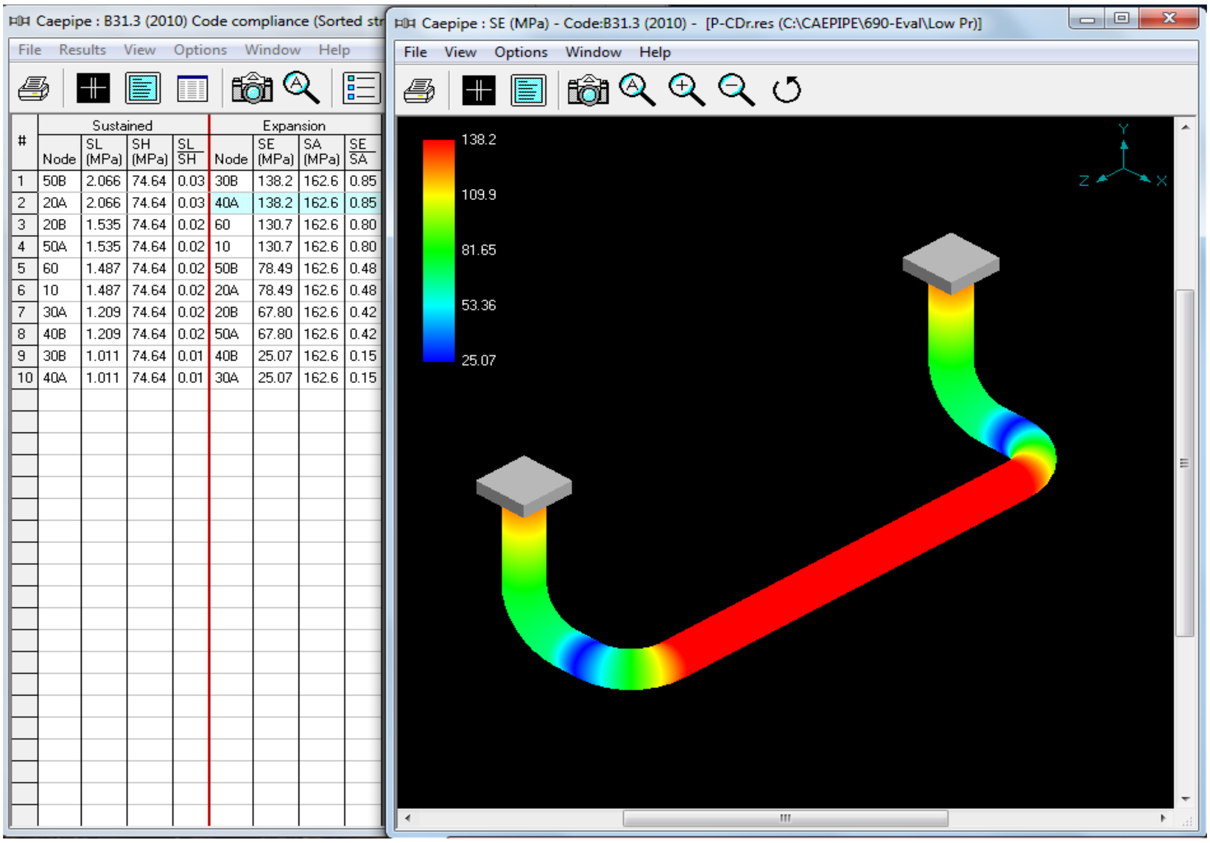
4.1.3. Analysis Results:

CAEPIPE is a piping analysis software, which analyses stresses on the pipes, including mechanical and thermal stresses at the given load conditions. It has option for material selection for the pipe and vibration analysis. Analysis for all pipes is carried on and screen shots of the result windows are taken. Among many pipes, few are presented in this article.

4.1.3.1 LP line from HE7 to HE6:

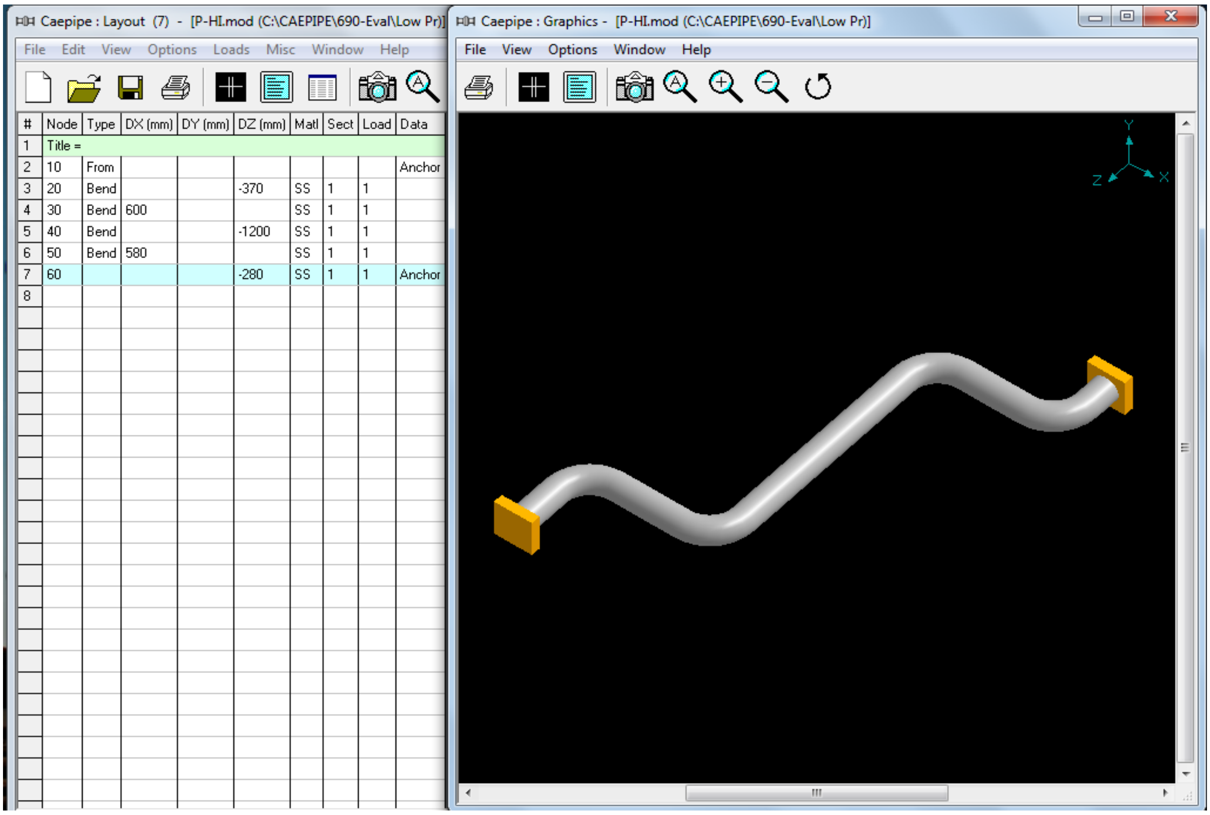


(Figure 19: Appearance of the LP line from HE7 to HE6)

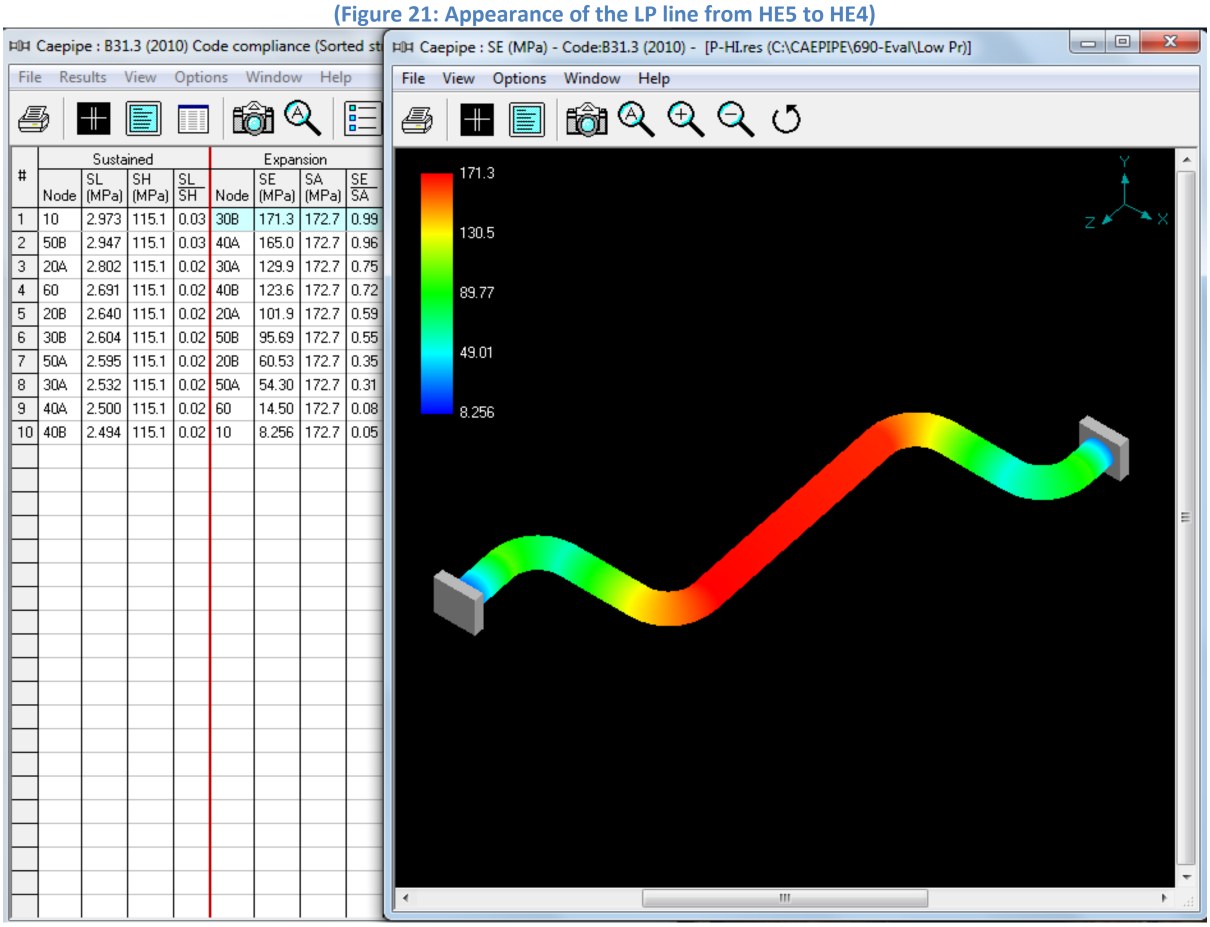


(Figure 20: Stresses distribution for the LP line from HE7 to HE6)

4.1.3.2 LP line from HE5 to HE4:

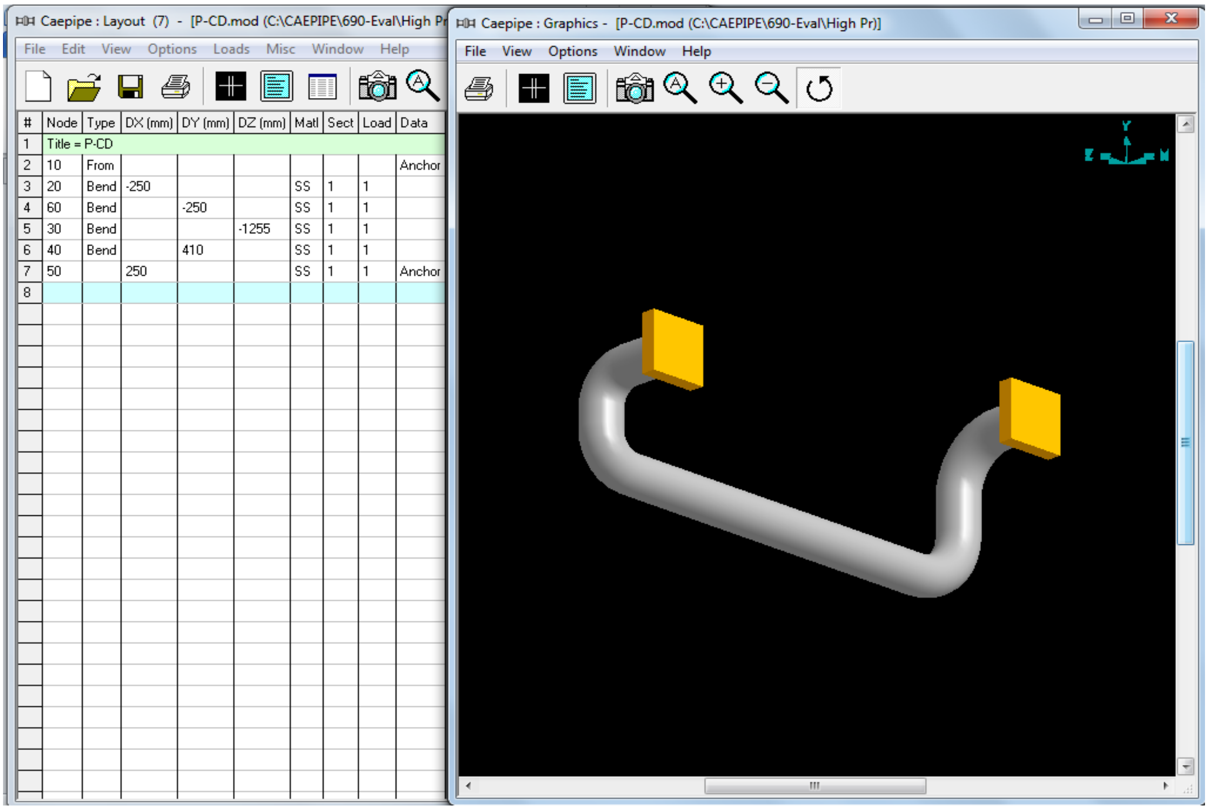


(Figure 21: Appearance of the LP line from HE5 to HE4)

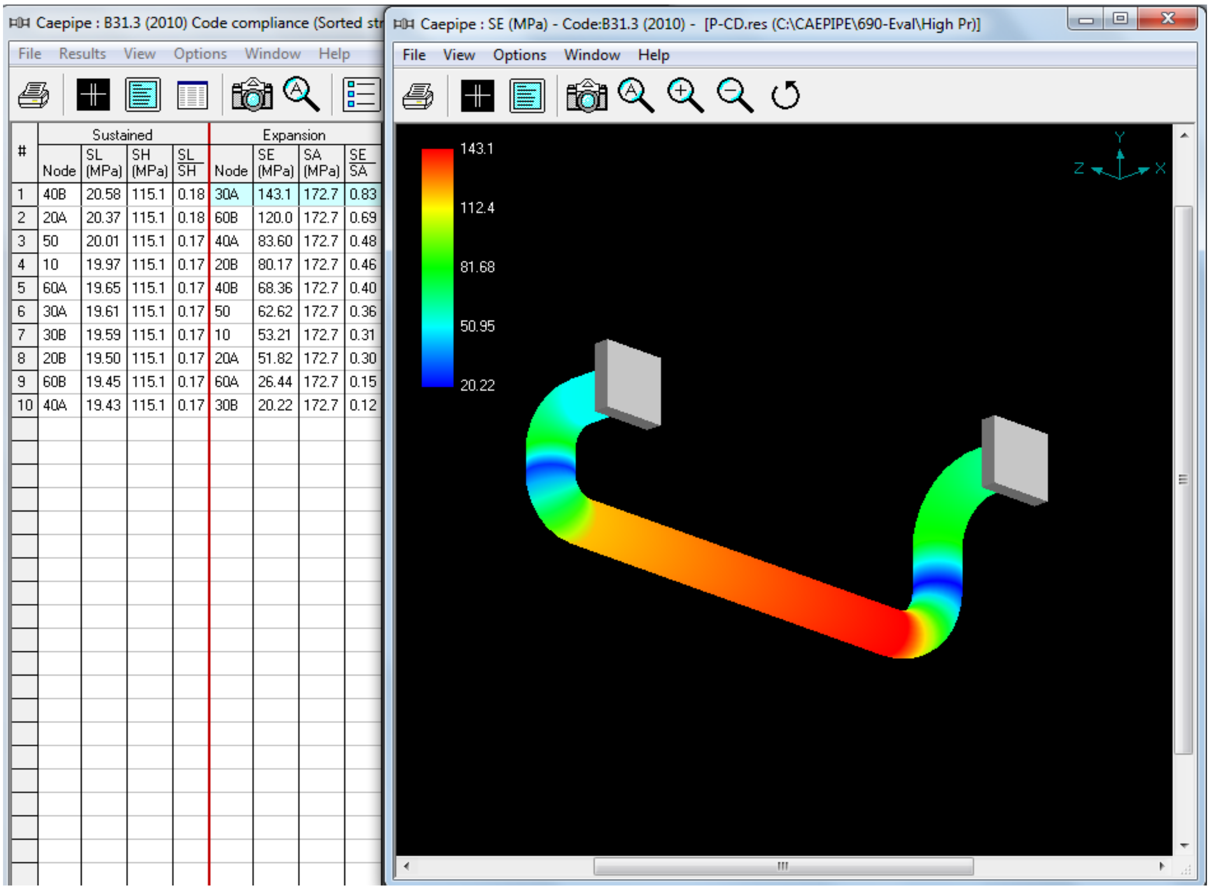


(Figure 22: Stress distribution for the LP line from HE5 to HE4)

4.1.3.3 HP line from HE1b to LN2 Vaporizer:

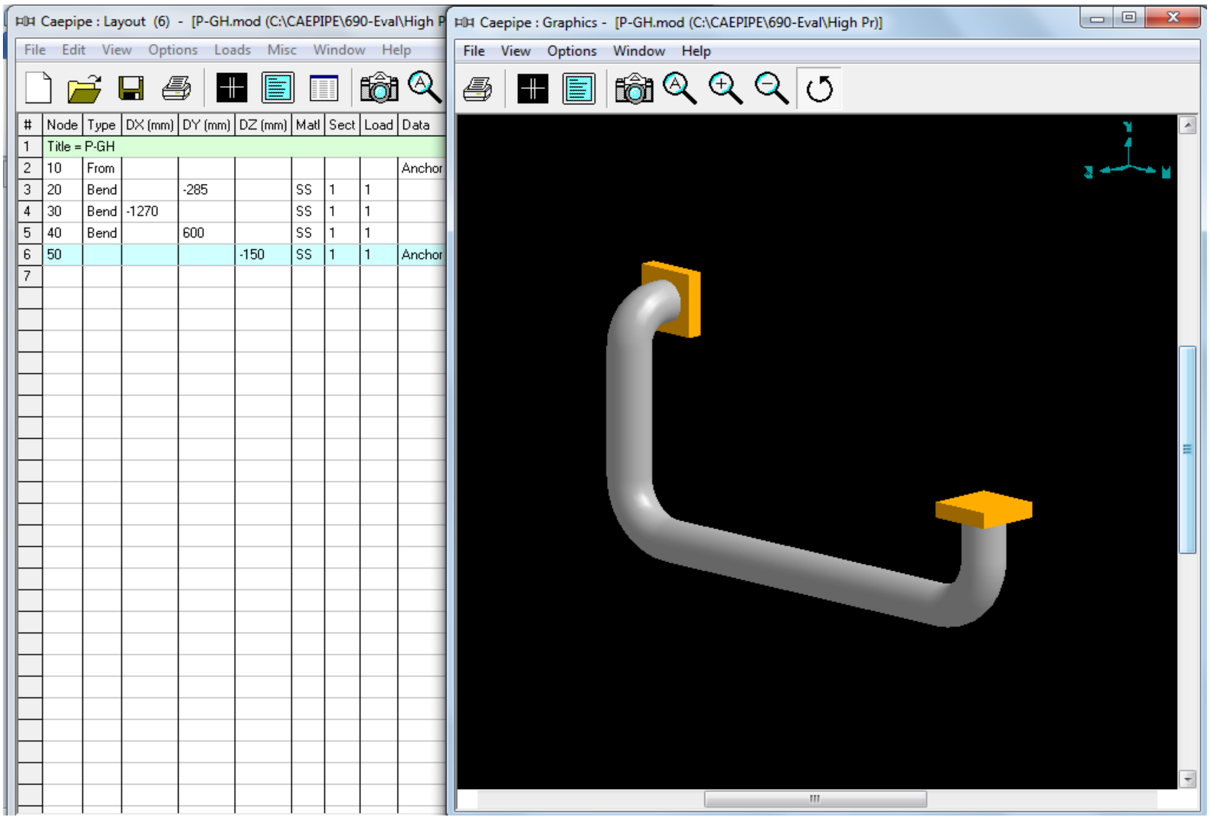


(Figure 23: Appearance of the HP line from HE1b to LN2 Vaporizer)

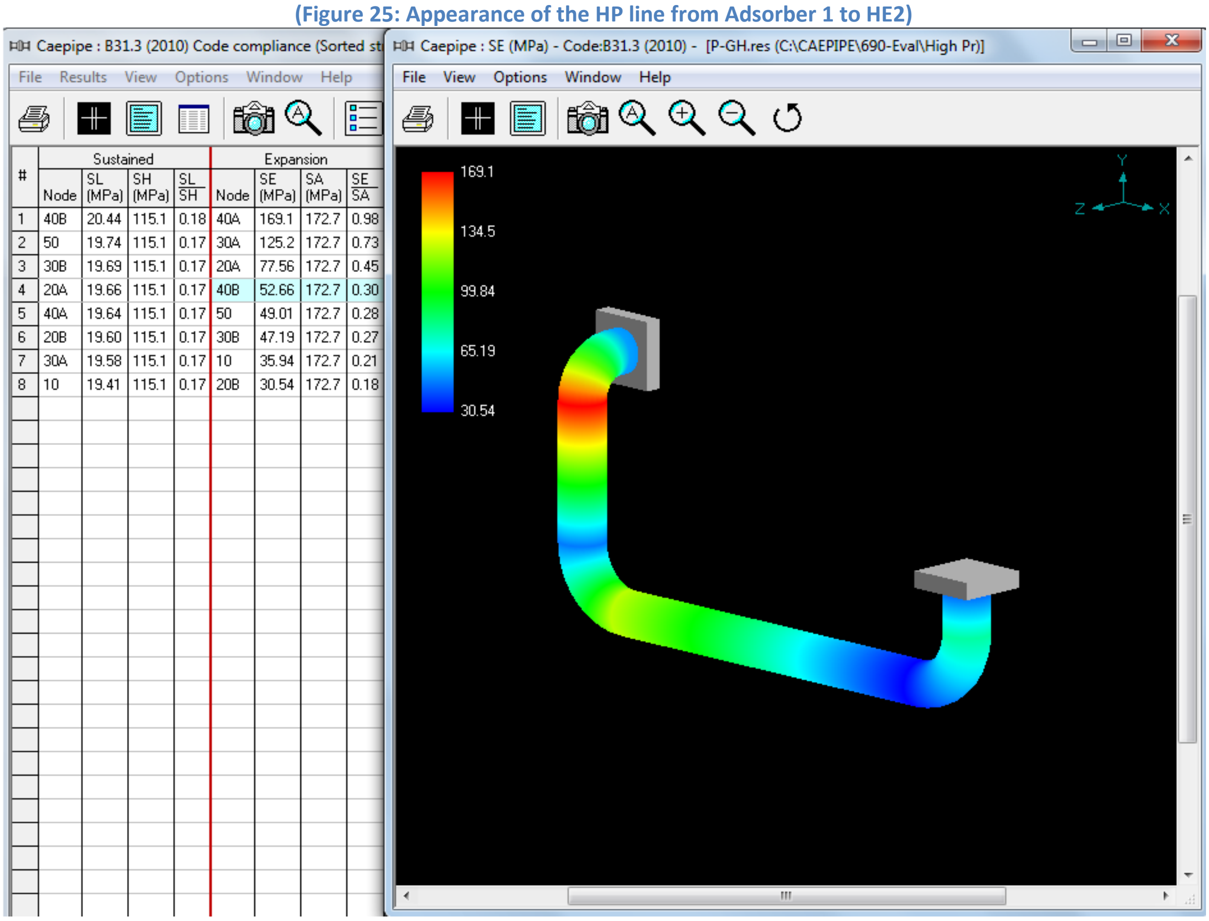


(Figure 24: Stress distribution for the HP line from HE1b to LN2 vaporizer)

4.1.3.4 HP line from Adsorber 1 to HE2:



(Figure 25: Appearance of the HP line from Adsorber 1 to HE2)



(Figure 26: Stress distribution for the HP line from Adsorber 1 to HE2)

4.2 Support System Design:

Hanging supports can be used to accommodate significant pipe movement in both the lateral and axial directions. Sliding style pipe supports can be used to accommodate large axial pipe movement. When the amount of pipe movement exceeds the capability of a hanger or roller pipe support system, a fixed support located in the centre of the pipe span can be effective in reducing the amount of movement. Heavy equipment should have fixed support on one side and should be allowed to thermally contract from other side. Roller support can also be used when assembly method demand that. When an uninsulated cryogenic line and equipment are supported, a portion of the pipe support will be at cryogenic temperature. The lower temperature should be considered when selecting the materials for the pipe support and its hardware.

Equipment with heavy weights will need supports of thicker cross section leading to higher heat loads. To reduce this, optimization of support type is necessary. G10 material is a suitable material for that. Equipment having temperature near to 4.5 K should have thermal intercepts (cooled by 80 K thermal shield) for its supports to reduce the heat loads. Supports should be designed to have low thermal and mechanical stress.

In our system, a discussion arises about whether to use supports from bottom or from top of the cold box. Supports from bottom require more material and larger cross-section is also needed. This increases the material cost as well as conduction heat load from atmosphere. The joints are more rigid. But our system does not have much space left at the bottom. So, this arrangement may not be fruitful. Also, installation of this kind of support is cumbersome. On the other hand, if the equipment are hanged from top, it requires much less material, thereby reducing material cost. Also, low cross-section of support ensures very less conduction heat load. Installation is easier compared to supports from bottom. This arrangement can be fruitful, as there is considerable amount of space left on the top of the vessel. The biggest advantage of support from top is that, this is very flexible; hence responds effectively to temperature fluctuation without failure.

Support system design includes deign for stresses arising and conduction heat load from 300 K atmosphere to 20 K equipment or 4 K equipment.

For supports, the most sought property is strength. This is why, stainless steel is chosen for this purpose, as it has very high yield strength of about 120 MPa.

Following procedure is adapted while designing supports:

i. A cross-section is chosen from Indian Standards for wire rods.

ii. Induced tensile stress is found by: $\sigma = \frac{W}{A}$; where, W is the weight of the equipment and A is the cross-sectional area of the particular support.

iii. By comparison, if, $\sigma < \sigma_y$; the section is safe and can be taken for further calculation.

iv. If, $\sigma > \sigma_y$; the section is not strong enough to withstand the stress. So, cross-sectional area of the support must be increased.

v. Conduction heat load is found by: $Q = \frac{A}{L}\int_{T_1}^{T_2} K.dT$; where, $\int_{T_1}^{T_2} K.dT$ is the thermal conductivity integral for the material.

vi. Finally, heat load is checked whether it is within limit or not. If not, either system is changed, so that the length of the support is increased or diameter is decreased.

Limits of heat loads for different equipment are given as follows:

Temperature range (K)	Allowable Heat load/ support (W)
Upto 80 K	$\leq$ 0.5 W
At 20 K	$\leq$ 0.2 W
At 4 K	$\leq$ 0.1 W

(Table 7: Allowable heat loads per support for different temperature zones)

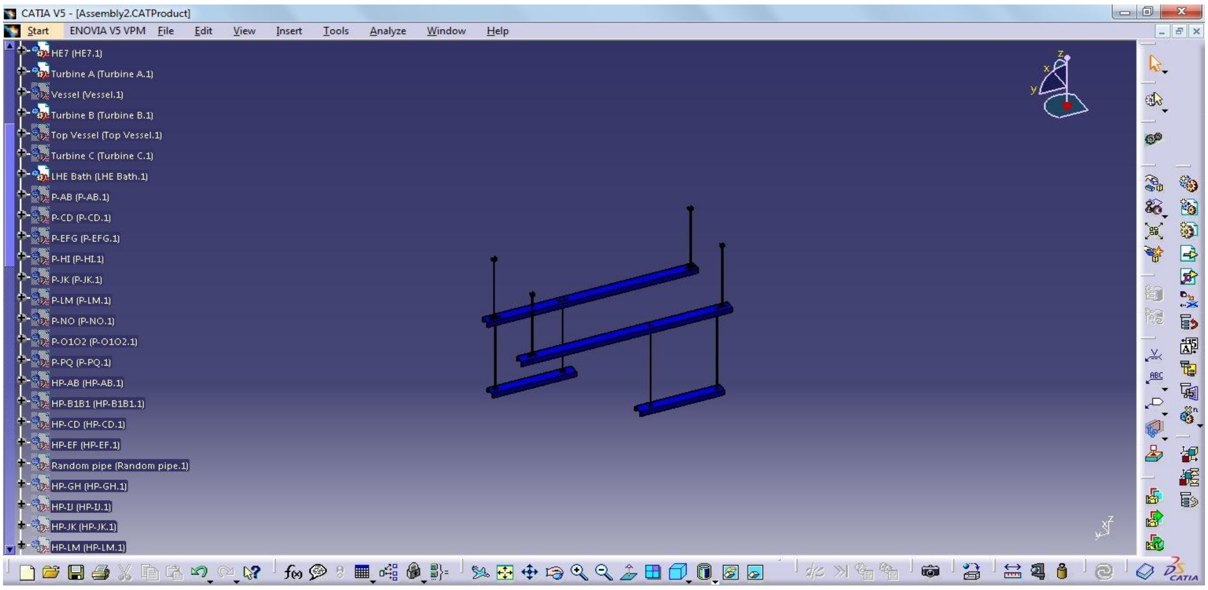
Approximate masses of different equipment are given as:

Equipment	Approximate Mass (kg)
HE1a	500
HE1b	500
HE2	360
HE3	90
HE4	200
HE5	100
HE6	70
HE7	70
Adsorber1	100
Adsorber2	50
LN_2 Vaporizer	70

(Table 8: Approximate masses for all equipment)

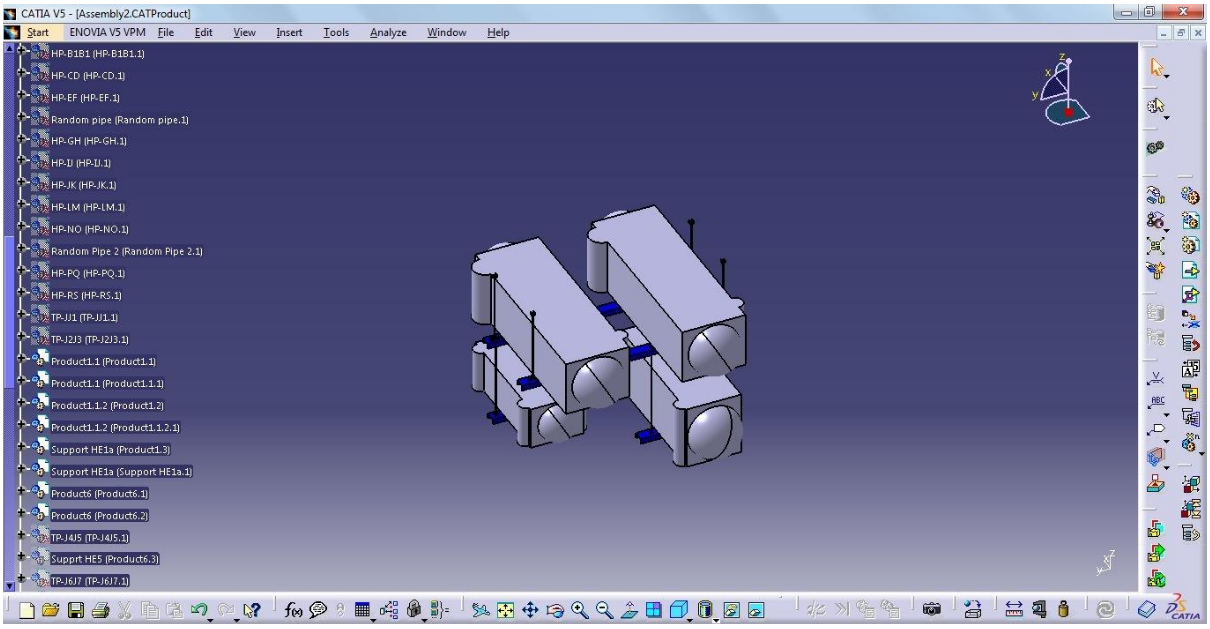
4.2.1 CATIA Modelling of Support System:

Entire support system is modelled in CATIA. Few of the major systems are shown:

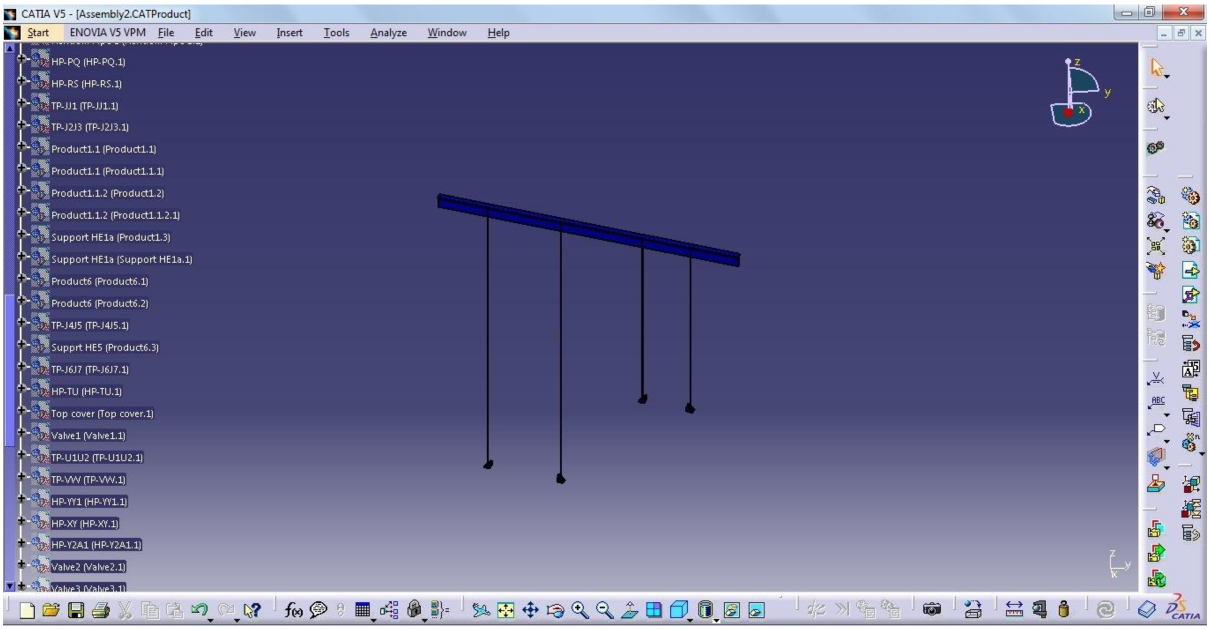


(Figure 27: CATIA model showing support for HE1a, HE1b, HE2 and HE4)

The heat exchangers are put on these Channel beams and the beams are then hanged from top. This arrangement provides excellent flexibility for disassembly purpose and minimize heat load. The beams are kept just below the centre of gravities of HE2 and HE4 to avoid tilting. And these are further given support from top beams. When the heat exchangers are mounted on this, it looks like:

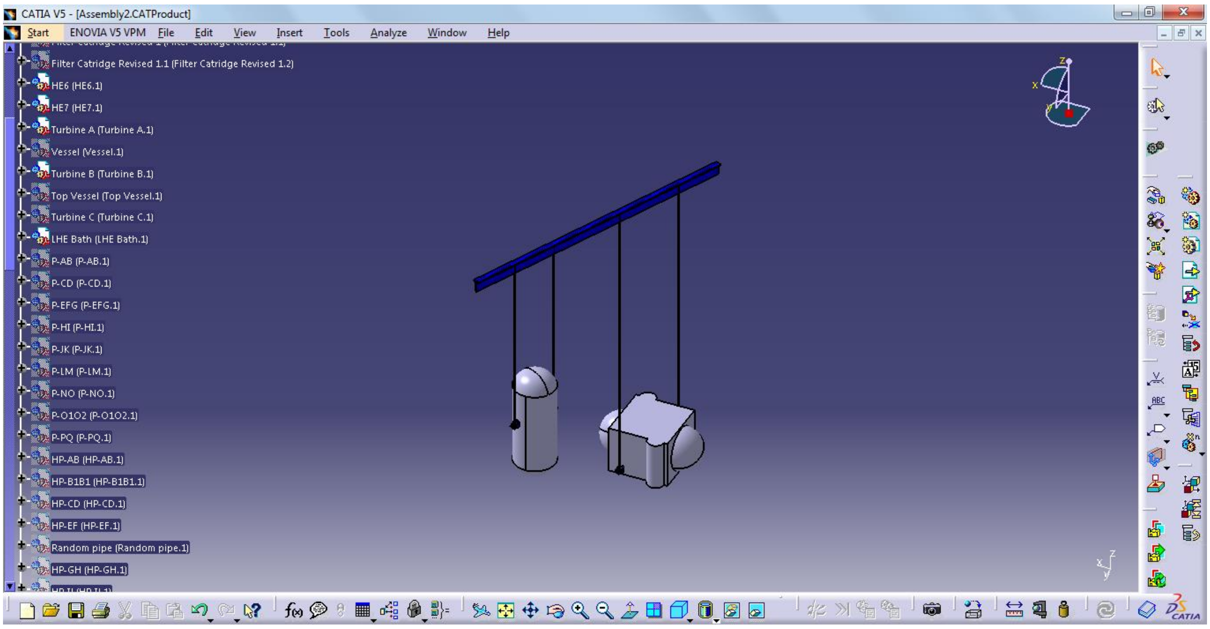


(Figure 28: Heat exchangers mounted on the support)



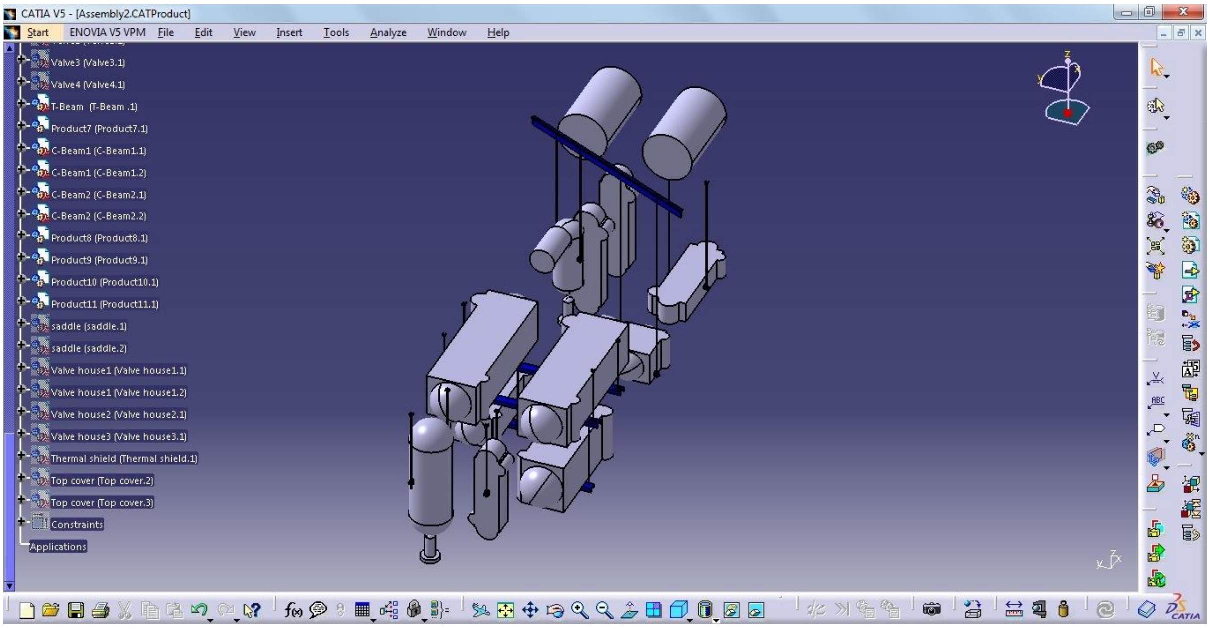
(Figure 29: Support system for HE3 and Adsrber 2)

As the third heat exchanger and the adsorber at 20K stand just below the top vessel, they cannot be hanged from roof top. Hence, a T-Beam is welded to the walls of the top vessel and rods are hanged from it to give support to the corresponding equipment. When the components are mounted, it looks like:



(Figure 30: HE3 and Adsorber 2 mounted on the support)

The entire system while mounted on supports looks like:

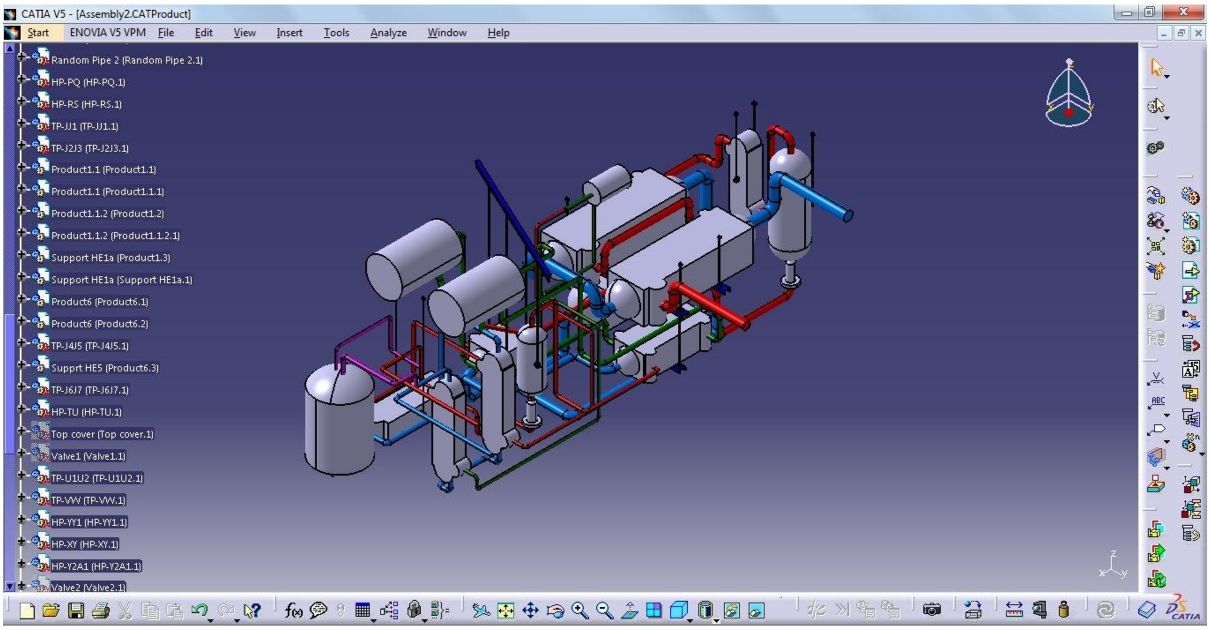


(Figure 31: Entire system on support)

Chapter 5.

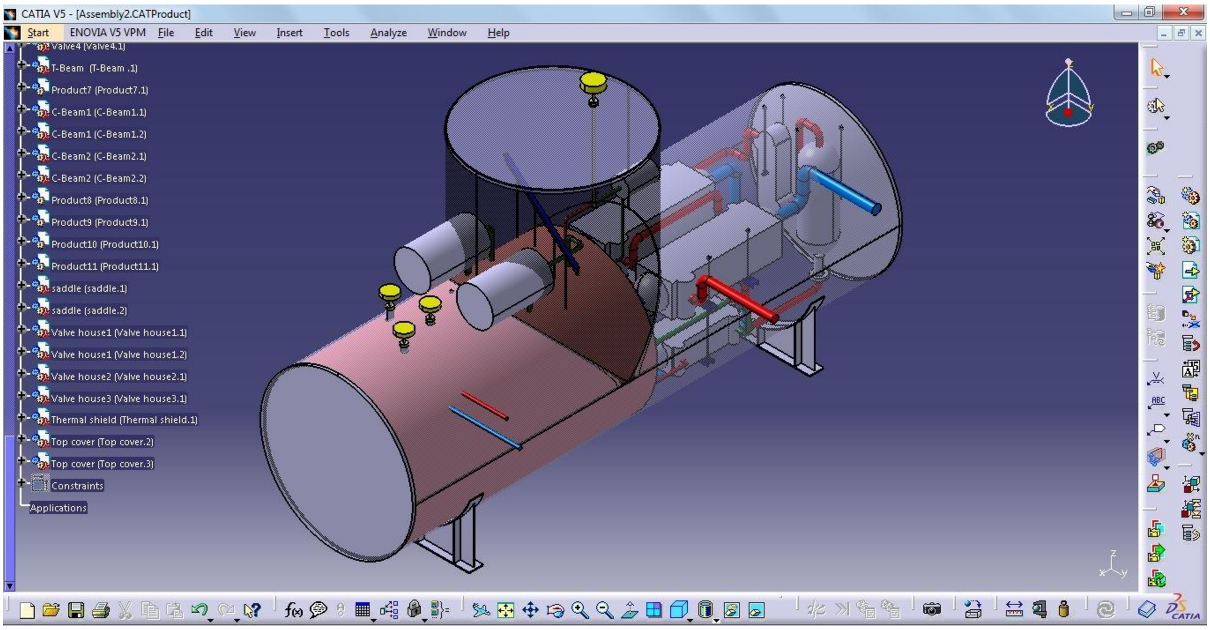
CATIA Modelling of Entire system

Entire system including the cold box, components, major valves, cover plates, saddle supports, all pipe lines and support systems are modelled in CATIA. These CATIA models are shown below:



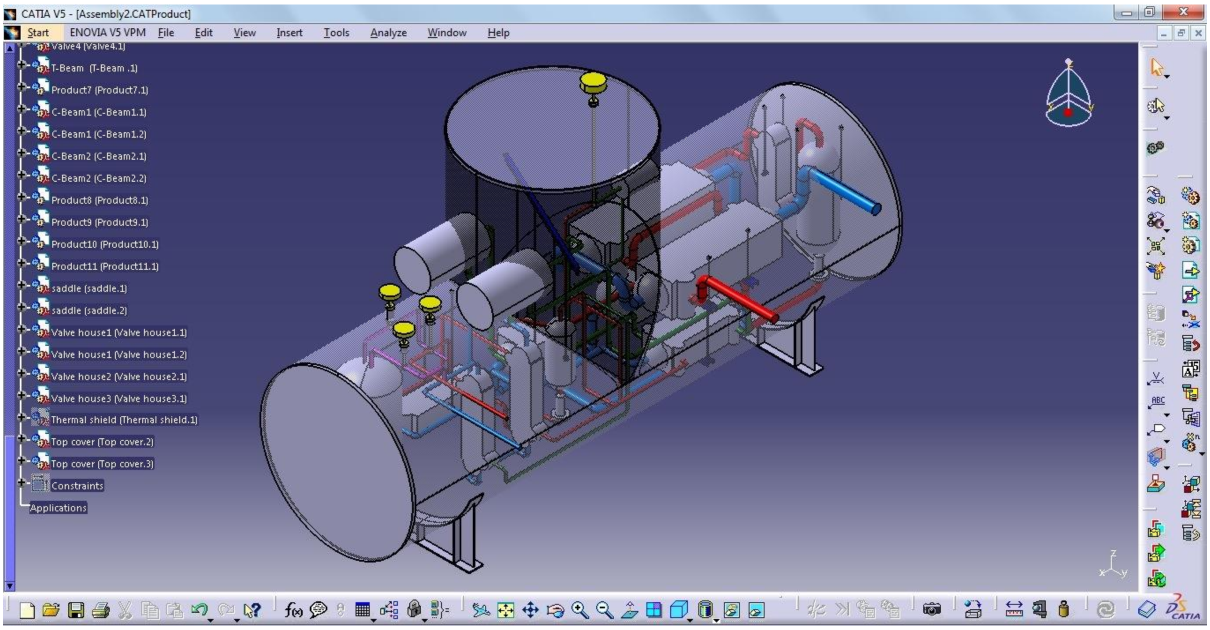
(Figure 32: All components with the piping system and supports)

Here, LP lines are coloured blue, HP lines are coloured red, lines going to the expanders are coloured green and lines going to the LHe bath are coloured pink.



(Figure 33: Entire system with thermal shield)

In this model, the pink cylinder is the thermal shield. It is shown that, the thermal shield is placed only upto half the length, i.e. only the 4 K side.



(Figure 34: Entire system with transparent cold box)

This model shows all components inside the cold box with the piping arrangement and support system; cold box being transparent.

Chapter 6.

Conclusion

As a concluding remark, this thesis satisfies the objectives it was meant to meet. Designing work for all aspects of the cold box is done satisfactorily. Optimized dimensions were found out. Strength, safety and cost were considered while designing the elements.

While deciding the internal component layout, through discussions were made. CATIA software was extensively used to get a clear picture of the cold box and its internal components. Finally, it was decided to keep the components as they are in the process flow diagram along the length, so as to maintain the temperature gradient.

ASME standard was thoroughly followed for design purposes. After deciding the internal component layout, the required length and diameter for the cold box was determined to be 8000 mm and 2500 mm respectively. With a thickness of 14 mm, the mass of cold box would be ~7420 kg. and including the components, the mass of system becomes ~9500 kg. The design for every element of the cold box was done carefully. A vessel of 14 mm thickness with 2 stiffening rings provides ample strength to withstand the external atmospheric pressure.

All piping for the system was done in CATIA and analysed in CAEPIPE. The results are satisfactory. But, the pipes may face some fabrication problems. Hence, the project can be extended for this purpose. Support systems were optimized in MS-EXCEL and drawn in CATIA. The average heat load by conduction through the supports for 20 K side is found to be 0.1-0.3 W. and that for the 4 K side is found to be 0.05-0.15 W. these heat loads are within permissible limit.

Overall the following targets were achieved:

- The cold box vessel was designed as per rules of ASME, Boiler and Pressure Vessel Code, section VIII.
- Saddle supports and stiffening rings are chosen effectively from standards.
- The components were arranged in proper layout maintaining a temperature gradient.
- Piping for all components was designed and flexibility was tested by CAEPIPE. And it is found that stresses and deflections are within limit.
- CATIA model of entire cold box system is generated, which shows all components within the cold box including valves, LHe bath and all heat exchangers with turbo expanders.
- A report of all the above work has been made.

The future scope for this project is:

i. Designing of thermal shield and MLI
ii. Analysis and discussion of fabrication issues for all the pipes
iii. Planning of assembly and disassembly for the cold box components and
iv. Designing of rectangular manhole on the top of the cold box.

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