Design of Low Heat Leak Liquid Helium Dewar

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ABSTRACT

IPR is planning to develop an indigenous Liquid Helium Refrigeration/Liquefaction Plant. So, at the end of the Liquefaction process when the Liquid Helium is produced, it is required to store. And from that it is supplied to different application and the generated He vapor is resupplied to the plant for the further liquefaction. So, to store this produced liquid helium, Liquid Helium Dewar will be used. The proposed Liquid helium (LHe) Dewar is to store liquid helium of 1000 liters. With design internal pressure 1.5 bar. This LHe storage Dewar should be designed such that the heat leak will as minimum as possible. To reduce heat leaks, it is planned to use vacuum and multilayer insulations (MLI). The vacuum space between inner and outer shell should have vacuum of 10^{-5} mbar or lower. The support structures of the inner shell which will carry the LHe need to be done considering the mechanical stress and low heat leaks. Appropriate ports should be included for vacuum pumping, safety valve and burst disc connection for pressure release, LHe and vapor process line connections, instrumentation connections. Requirements of these equipment also have to be worked out to find overall specs for procurement. The Dewar can be of vertical configuration having cylindrical shell with top and bottom dish ends.

Keywords: *Liquid Helium Dewar, MLI*

I. INTRODUCTION

IPR is Planning to design an indigenous Liquid Helium Plant which is used for production of liquid helium.During designing the helium liquefaction systems, the system constraints have to be properly designated and optimized (like mass flow rates, component efficiencies, purity of gas, temperature levels of components, pressures etc.), so that maximum liquid production is to be attained with minimum power input. A helium liquefier has to bring down the temperature of feed gas from normal room temperature to 4.2 K and this requires many steps of refrigeration or cooling components.

The process flow diagram for liquid helium production for which the LHe Dewar will be designed is shown in fig.1.

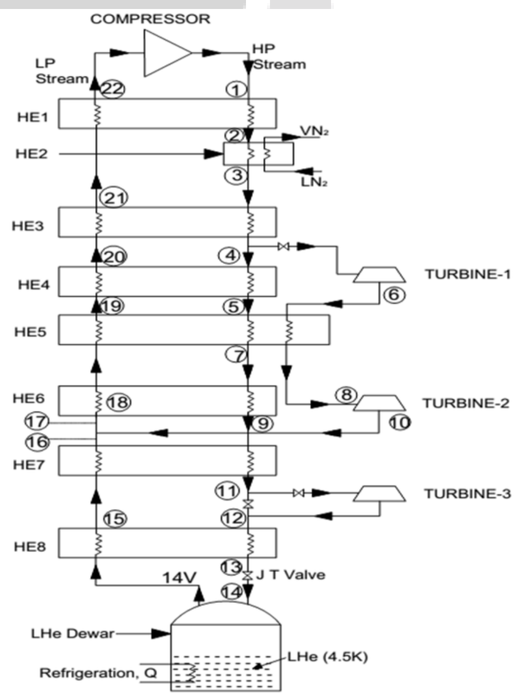


Fig.1. Process Flow Diagram of Indigenous Liquid Helium Plant (Courtesy: IPR)

In the proposed design of the HRL Plant at IPR, it will operate on either refrigeration or liquefaction or both. So, at the end of the process, when the helium will liquefied, it is required to store and from that it is supplied to different application and the generated He vapour is further supplied to the plant for the liquefaction. So, to store this produced liquid helium, Liquid Helium Dewar will be used. This LHe storage Dewar should be designed such that the heat leak will as minimum as possible because as the heat leak in the Dewar increases the boil off rate of Liquid Helium is also increases. And the Dewar should have the capacity of about 1000 litres. And it should have Heat In leak less than 10 W.

A. SELECTION OF SHAPE OF DEWAR

According to Ken Harrison [4], Deflection is the important factor for shape selection of the Dewar. The best shape to minimize both material and deflection is a sphere. . But the limitation to this shape is that some applications do not readily fit in it. The forming costs of the hemispheric halves can also be expensive for larger chambers So, The next best shape is a cylinder with length is equal to diameter. This is one of the most common shapes, and the rigidity is almost as good as a sphere. Because in this type of shape surface area is only 10% more than the sphere of same storage capacity

According to Saurabh Lawate, B. B. Deshmukh [5], The Surface Area of Torispherical heads is less than elliptical heads & hemispherical head. For Pressure (P<10bar), Deformations in elliptical & Torispherical dish ends are in close agreement And The forming cost of Torispherical heads is less than elliptical heads & hemispherical heads because of availability regular circular curves on the edges then a larger curve as it heads. So, For Pressure (P<10 bar), Torispherical shape is economical.

So, The Selected Shape of Dewar is Cylindrical with top and Bottom Torispherical head

B. SELECTION OF MATERIAL

According to George Behrens, William Campbell in "design guidelines for cryogenic systems"[7] For the Cryogenic application SS-304 L is widely used because it does not oxidize and can be heated to very high temperatures for bake-out to reduce the component of the gas load caused by diffusion (gases within the crystalline structure of the metal). Another reason for using S/S-304L is that it is easily electro polished, which provides a clean surface free of oxidation and contamination. Stainless steel is also easily welded with the (TIG) Tungsten Inert Gas (argon) method that is needed for producing vacuum tight welds for high and ultra-high vacuum operation.

According to Ken Harrison [4], to obtain a good leak proof chamber, the weldability of the material must be excellent. Stainless steel should be use for vacuum application because it has good weldability property.

According to C Gayari [9], Austenitic stainless steels of the I8/8 types (18% chrome, 8% nickel) offer a favourable set of properties which make them particularly well suited for the use of in the vacuum and cryogenic application because it has good welding properties , low effect of sensitization ,good mechanical properties.

According to Phil Danielson [10], choose that material for vacuum system design which has low outgassing rate. And the surface of the I8/8 type stainless steels is inert, passive and has a very low outgassing rate i.e. 6×10^{-9}(torr litre/sec/cm^2)

So, SS -304 L is selected for the inner vessel and Carbon steel SA-516 grade 60 is selected for outer vessel because it not subjected to cryogenic temperature.

C. SELECTION OF INSULATION

There are different types of insulations are used in the cryogenic application for reducing the heat in leak.

According to Holger Neumann [22], Multilayer Insulation is most effective for Low Temperature Application because it has low heat conductivity at low Temperature Range as shown in fig 2.

MLI consists of layers of alternate low emittance radiation shields separated either by low conductivity spacer or by crinkling or embossing the shields so that when placed against each other, the shields only touch at a few discrete points.

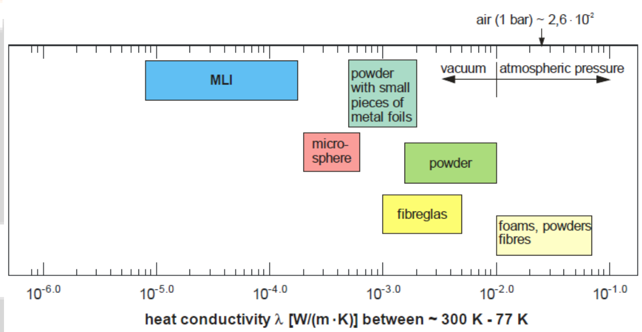


Figure 2. Range of heat conductivity of various insulation materials at different temperature. [12]

I. SELECTION OF SHIELD MATERIAL OF MULTILAYER INSULATION

According to Jacobus Henry Hodgman [13], a highly polished Aluminium film is most commonly used as shield material because it has very low emissivity i.e. 0.02.

So, aluminium film is selected for the shield material of multilayer insulation.

II. SELECTION OF SPACER MATERIAL OF MULTILAYER INSULATION

According to Jacobus Henry Hodgman [13], if polymer net is used in place of composite paper there is further reduction of heat transfer because the contact area through which heat is transferred from one shield to another is reduced and Nylon is most commonly used as spacer material because it has low thermal conductivity value.

So, nylon net is selected for the spacer material of the multilayer insulation.

D. SELECTION OF VACUUM LEVEL

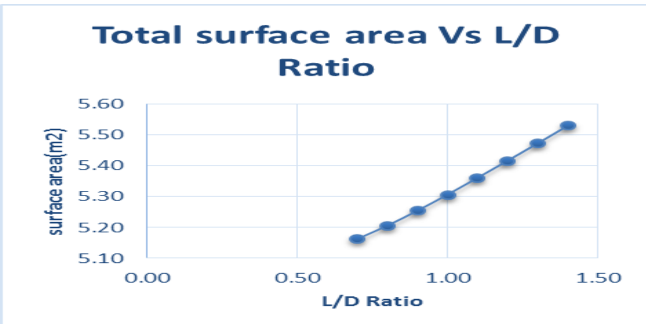
In liquid Helium Dewar we need to have least amount of heat load coming into the cryogen reservoir (at 4.5K). The maximum amount of heat load that can be transferred into the cryogen reservoir is due to natural convection from atmospheric condition (at 300K), hence to reduce this heat load we have to evacuate the space between vacuum jacket and cryogen reservoir.

The space is evacuated using pumping station to create vacuum in the range of>10-5 mbar. The pumping station consists of Turbomoleculer pump backed by a rotary or scroll pumps. This arrangement of pumps is enough to produce high vacuum of above 10-5 mbar.

II. DESIGN METHODOLOGY & DESIGN CALCULATION

A. INNER VESSEL DESIGN [3]

Here capacity of storage vessel is taken as 1000 litres and ullage volume is taken as 10%, so total volume of inner vessel is 1100 litres or 1.1 m^3.



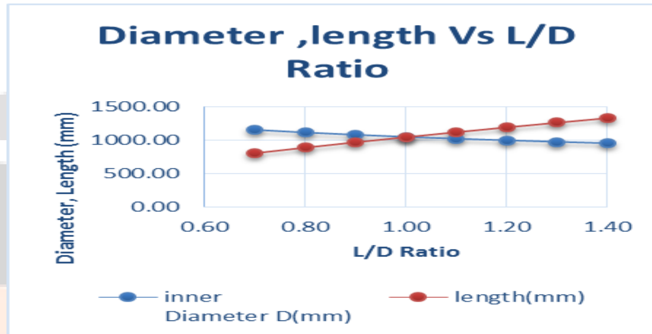


Fig. 3 Selection of L/D ratio for inner vessel

The plotted graphs for this case describe the L/D ratio decrease, the total surface Area of vessel is also decreases but the diameter of the inner vessel is increases so the required space is also increases. Considering the surface area and the required space we have select the L/D ratio is 1.

So, the total volume of vessel is given by,

$$V = \frac{\pi}{4} D^2 L + 2 \times \left(\frac{\pi D^3}{39.173}\right)$$

From this equation, the diameter of vessel comes 1057 mm and length is also 1057 mm. But from the standards and for being on the safe side it is preferable to take both diameter and length equal to 1060 mm.

I. THICKNESS OF CYLINDRICAL SHELL OF INNER VESSEL

According to the ASME Code, Section VIII, the minimum thickness of the inner shell for a cylindrical vessel is given by

$$t = \frac{PD}{2S_a e_w - 1.2P} = \frac{PD_o}{2S_a e_w + 0.8P}$$

P = design internal pressure (Pa) = 1.5×10^{-5}
D = inside diameter of shell (m) = 1.06
S_a = allowable stress (Pa) = 120.6×10^6
e_w = weld efficiency = 1

From this equation, the thickness of cylinder comes 0.66 mm.

CONSIDERING THE BUCKLING FOR CYLINDRICAL SHELL

Sometimes when we evacuating the He from the container, it is possible that there is an atmospheric pressure between inner vessel and outer vessel and there is a vacuum inside the inner vessel. So it is necessary that the inner vessel must be withstand against buckling or collapsing.

The collapsing pressure for a short cylinder subjected to external pressure is given by

$$P_c = \frac{2.42E(\frac{t}{Do})^{5/2}}{(1-\nu^2)^{3/4}[\left(\frac{L}{Do}\right) - 0.45(t/Do)^{1/2}}$$

P_c = Critical Pressure (Pa)
E = Young's modulus material (Pa) = 207×10^9
t = shell thickness (m)
L = Length of shell (m) = 1.06
D_0 = outer diameter of shell (m)
ν= Poisson's ratio for a shell material = 0.28

From the equation, for the different value of thickness the value of critical pressure is shown in following graph. From the graph we can say that the cylinder is safe against collapsing with thickness greater than 4 mm .so according to standard we have select the thickness equal to 5 mm for inner cylinder.

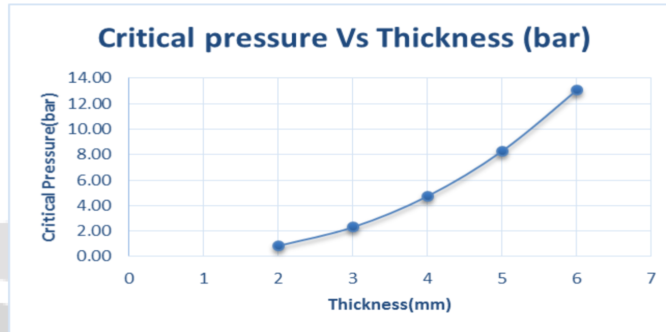


Fig. 4 Critical Pressure Vs thickness for inner vessel cylinder

II. INNER VESSEL HEAD

Now, we have select the torispherical head .so the different parameters according to ASME standard are inside diameter is 1060 mm, inside dish radius is 1060 mm , knuckle radius is 63.6 mm and straight skirt height is 38.1 mm .

THICKNESS OF INNER VESSEL HEAD

Now, the thickness of torispherical head according to ASME Sec. VIII is given by

$$t_h = \frac{0.885PL}{S_a e_w - 0.1P}$$

P = Internal Pressure (Pa) = 1.5×10^{-5}
L =Inside Dish Radius or Crown Radius (m) = 1.06
S_a = allowable stress (Pa) = 120.6×10^6
e_w = weld efficiency = 1
ν= Poisson's ratio for a Head material = 0.28

From this equation, the thickness of head comes 1.17 mm.

CONSIDERING THE BUCKLING FOR INNER VESSEL HEAD

The collapsing pressure for a Torispherical head subjected to external pressure is given by

$$P_c = \frac{0.5E(t_h/Ro)^2}{[3(1-\nu^2)]^{1/2}}$$

P_c = Critical Pressure (Pa)
R_o = Outside Crown Radius (m)
t_h = Thickness of head (m)
ν= Poisson's ratio for a Head material = 0.28

From the equation, for the different value of thickness the value of critical pressure is shown in following graph. From the graph we can say that the cylinder is safe against collapsing with thickness greater than 3 mm .so according to standard we have select the thickness equal to 5 mm for head.

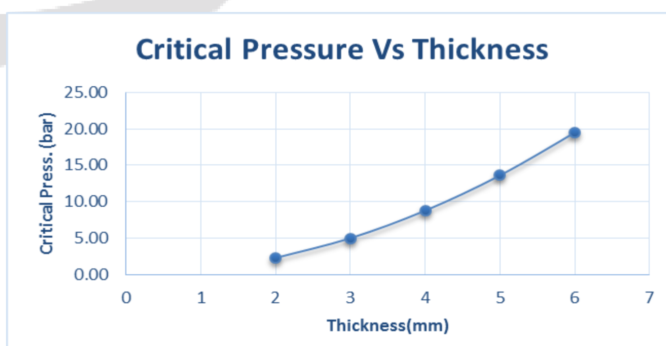


Fig. 5 Critical Pressure Vs thickness for inner vessel Head

B. OUTER VESSEL DESIGN [3]

Generally the annular gap between inner vessel and outer vessel is kept between 1/2 to 1/3 of the inner vessel diameter.so we will take annular gap is 1/3 of inner vessel diameter is equal to 360 mm. so the gap between inner vessel and outer vessel is 180 mm.

I. THICKNESS OF CYLINDRICAL SHELL OF OUTER VESSEL

From the U.S. Experimental Model Basin formula given by Wrinderburg and Trilling (1960), the collapsing pressure for a short cylinder subjected to external pressure is given by

$$P_c = \frac{2.42E(\frac{t}{Do})^{5/2}}{(1-\nu^2)^{3/4}[\left(\frac{L}{Do}\right) - 0.45(t/Do)^{1/2}}$$

Pc = Critical Pressure (Pa)
E = Young's modulus of shell material (Pa) = 200 × 109
t = Shell thickness (m)
L = Length of shell (m) = 1.430
D0 = Outer diameter of shell (m)
ν = Poisson's ratio for a shell material = 0.27

From the equation, for the different value of thickness the value of critical pressure is shown in following graphs. From the graph we can say that the cylinder is safe against collapsing with thickness greater than 4 mm .so according to standard we have select the thickness equal to 5 mm for outer cylinder.

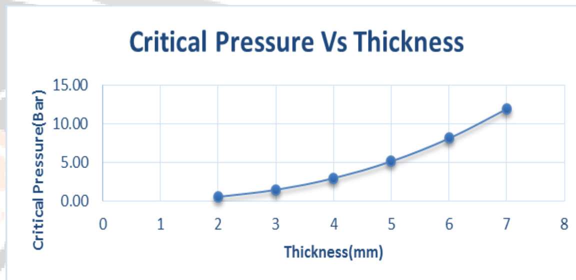


Fig. 6 Critical Pressure Vs Thickness of outer vessel

II. OUTER VESSEL HEAD

Now, we have select the torispherical head .so the different parameters according to ASME standard are inside diameter is 1430 mm, inside dish radius is 1430 mm , knuckle radius is 85.8 mm and straight skirt height is 51.3 mm .

THICKNESS OF OUTER VESSEL HEAD

The critical pressure for a Torispherical head is given by

$$p_c = \frac{0.5E(t_h/Ro)^2}{[3(1-\nu^2)]^{1/2}}$$

Pc = Critical Pressure (Pa)
Ro = Outside Crown Radius (m)
th = Thickness of head (m)
E = Young's modulus of shell material (Pa) = 200 × 109
ν = Poisson's ratio for a Head material = 0.27

From the equation, for the different value of thickness the value of critical pressure is shown in following graph. From the graph we can say that the cylinder is safe against collapsing with thickness greater than 4 mm .so according to standard we have select the thickness equal to 5 mm for head

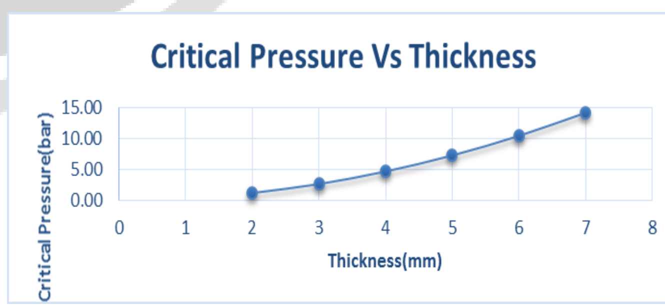


Fig. 7 Critical Pressure Vs Thickness of Outer vessel Head

C. DESIGN OF PIPING SYSTEM [3, 10]

I. DESIGN OF LIQUID HELIUM INLET PIPE

DETERMINATION OF PIPE THICKNESS:

The minimum wall thickness for piping subjected to internal pressure is given by,

$$t = \frac{PD_o}{2S_a + 0.8P}$$

P = 1.5 × 105 Pa

Sa= 120.6 × 106 Pa

From the equation, for the different standard outside diameter of pipe the required thickness of pipe is shown in the fig which is very small. So we have selected the standard thickness of pipe equal to 2.7686 mm

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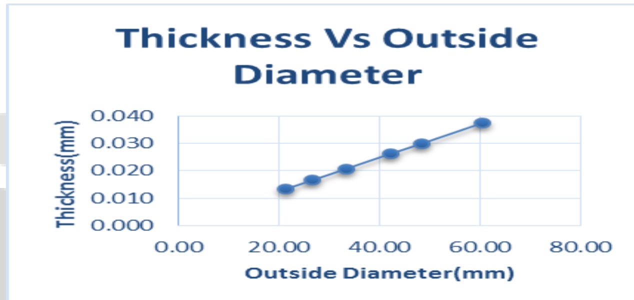


Fig. 8 Required Thickness Vs outside Diameter of Pipe subjected internal Pressure

DETERMINATION OF PIPE DIAMETER:

Now, for selecting the Diameter of pipe First We have to find Pressure Drop in the pipe and according to Allowable Pressure drop we have to select the Diameter.

Here in this pipe two phase flow is flowing. So two phase flow frictional pressure drop, using Martinelli and Lockhart correlation [3] following graph is obtained. And from that our allowable pressure drop is 0.1 mbar/m .so we will select the pipe having outside diameter 42.16 mm.

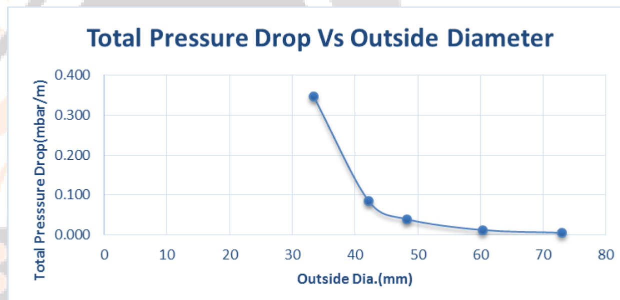


Fig. 9 Total Pressure Drop Vs outside Diameter of Inlet Pipe

II. DESIGN OF HELIUM VAPOUR OUTLET PIPE

DETERMINATION OF PIPE THICKNESS:

The Thickness of the Vapour outlet Pipe can be calculated same as inlet pipe. So the thickness of the outlet pipe is also taken 2.768 mm as per schedule 10 in Nominal Pipe size.

DETERMINATION OF PIPE DIAMETER:

Here in this pipe only vapour is flowing. So calculating pressure drop, using Darcy weisbach equation [10] following graphs are obtained. And from that our allowable pressure drop is 0.1 mbar/m .so we will select the pipe having outside diameter 60.33 mm.

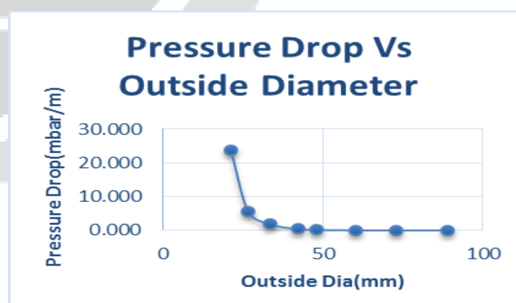
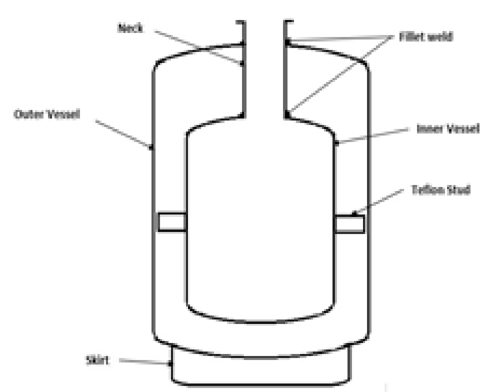


Fig 10 Total Pressure Drop Vs outside Diameter of

outlet pipe

III. DESIGN OF NECK

The outside diameter of neck we have to select according to the outside diameter of inlet and outlet pipe. So according to that we have select the outside



diameter of pipe equal to 168.26 mm (6 inch.) and the thickness of pipe is equal to 2.7686 mm.

D. DESIGN OF INNER VESSEL SUSPENSION SYSTEM

The arrangement of Suspension system is shown in fig. and according to this the inner vessel is suspended through neck. And four Teflon stud prevents the axial movement of inner vessel.

Due to this arrangement, there are two types of stress is produced. One is tensile stress in the neck and the other is shear stress at weld. So the tensile stress is 2.11 MPa. And the shear stress is 1.240 MPa. So both are less than the allowable stress.so our design is safe.

Fig. 11 Inner Vessel Suspension System

E. DESIGN OF PRESSURE RELIEF VALVE [3]

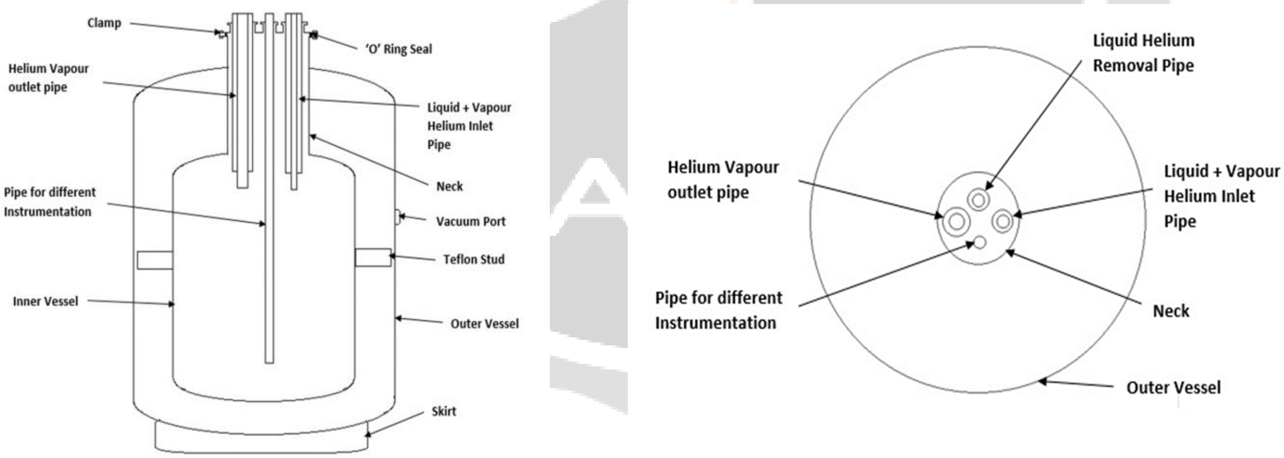
The required size of the safety valve is determined by the ASME Code is given by

$$A_v = \frac{m_g (R_u T / g_c M)^{1/2}}{C K_D p_{max}}$$

Av = Discharge area of valve (m2)
mg= Max. Mass flow rate through valve (Kg/s) = 0.07 Kg/s
Ru = Universal gas constant (J/ Kg-mol-K) = 8314.3 Nm/Kg mol-K
T = Absolute temperature of the gas at the inlet to the valve (K) = 4.5 K
gc = Unit conversion factor in Newton's Second law = 1 Kg-m/N-s2
KD = discharge co efficient = 0.65
Pmax = (set gauge pressure) (1.10) + (atmospheric pressure) = 1.54867 × 105 Pa

From this equation the required area of safety valve comes 92.714 mm^2 .

The final assembly of Dewar parts is shown in fig.12



Front View Top View

Fig.12 Assembly of different parts of Dewar

F. DESIGN OF MULTILAYER INSULATION [29]

The Heat flux through Evacuated MLI having Aluminium Foil as Radiation Shield and Dacron Fibre as a spacer is given by below formula:

$$q = \frac{C_s (N)^{2.56} T_m (T_H - T_C)}{N_s + 1} + \frac{C_r \varepsilon_{RT} ({T_H}^{4.67} - {T_c}^{4.67})}{N_s}$$

q = Heat flux (W/m2)
Cs = Solid Conduction Coefficient = 8.95 × 10-8
Cr = Radiation coefficient = 5.39 × 10-10
TH = Temperature of Hot side (K) = 300 K
Tc = Temperature of cold side (K) = 4.5 K
Tm = Mean Temperature (K) = (300 + 4.5)/2 = 152.25 K
N = Layer Density (Layers / cm)
Ns = No. of Radiation Shield
ƐRT = Emissivity of Shield Material

From this equation following graphs are obtained. From the graph of Heat flux Vs Layer density, we can see that the heat flux is minimum while layer density equal to 20 layers/ cm. and from the graph of Heat flux Vs radiation shied, we can see that The Heat Flux is first Drastically Reduced and After 40 Radiation Shield there is minor change in the Heat flux. So, we will select 40 Radiation Shield.

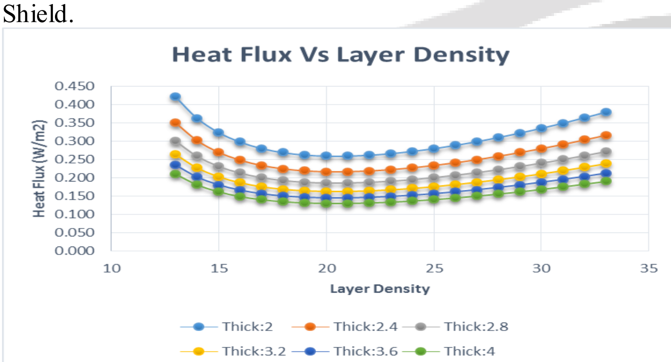


Fig.13 Heat flux Vs Layer Density

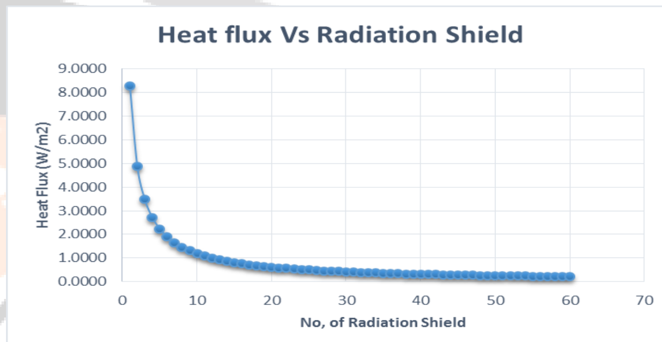


Fig.14 Heat flux Vs Radiation Shield

G. HEAT TRANSFER CALCULATION:

I. HEAT TRANSFER THROUGH MULTILAYER INSULATION AROUND INNER VESSEL:

The total mean surface area of the inner vessel is 5.6194 m2 and the heat flux through multilayer insulation is 0.1695 W/m2. So the total heat transfer through the insulation is 0.9524 W.

II. HEAT TRANSFER THROUGH RESIDUAL GAS CONDUCTION

The accommodation coefficient factor is given by,

$$F_a = \frac{1}{[\frac{1}{a1} + \left(\frac{A1}{A2}\right)\left(\frac{1}{a2} - 1\right)]}$$

a1 = accommodation coefficient for surface 1 = 1
a2 = accommodation coefficient for surface 2 = 0.9
A1 = Area of Surface 1 = 3.528 m2
A2 = Area of Surface 2 = 6.465 m2

From the equation the accommodation coefficient factor is obtain 0.943.

The residual gas conduction is calculated by the following formula

$$Q_{rgc} = [\frac{\gamma + 1}{\gamma - 1}] \left[\frac{R}{8\pi T}\right]^{\frac{1}{2}} F_a P A_1 (T_2 - T_1)$$

γ = Specific Heat Ratio = 1.4

R = Specific Gas Constant = 287
P = Pressure = 1×10-4 Pa
A1 = 3.528 m2
T1 = Temperature of inner surface =4.5 K
T2 = Temperature of inner surface = 300 K
From the equation Qrgc comes 0.1151 W.

III. HEAT TRANSFER THROUGH PIPING

Heat transfer through piping is determined from the Fourier rate equation considering the effect of variable thermal conductivity.

$$Q_2 = (K_{T1} - K_{T2}) \times \frac{A_c}{L}$$

A. HEAT TRANSFER THROUGH NECK

From above equation the heat transfer through neck comes 5.47 W.

B. HEAT TRANSFER THROUGH HELIUM INLET PIPE

From above equation the heat transfer through Helium inlet pipe is comes 1.33 W.

C. HEAT TRANSFER THROUGH LIQUID HELIUM REMOVAL PIPE

From above equation the heat transfer through Liquid Helium removal pipe is comes 0.7021 W

D. HEAT TRANSFER THROUGH PIPE FOR INSTRUMENTATION

From above equation the heat transfer through pipe for instrumentation is comes 0.5674 W.

So, the total heat load is 9.128 W. and the boil off rate of Liquid Helium is 1.4 % per Day.

IV. ANALYSIS IN ANSYS SOFTWARE

A. STATIC STRUCTURAL ANALYSIS OF DEWAR

I. EQUIVALENT (VON-MISES) STRESS ANALYSIS IN DEWAR

As shown in fig.15, the induced Equivalent stress inside the Dewar 80.392 MPa is less then Allowable stress of 120MPa [3]. So, the Dewar is on the safer side.

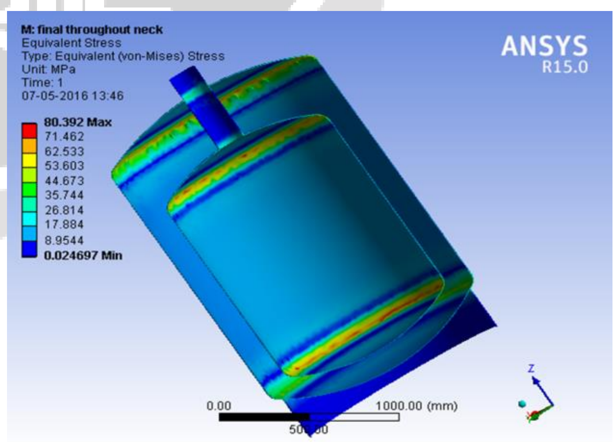


Fig. 15 Equivalent (Von-Mises) Stress in Dewar

II. TOTAL DEFORMATION IN DEWAR

From the fig.16, it indicate the total deformation occurs in the bottom inner vessel head is 1.234 mm, which is minimum and acceptable.

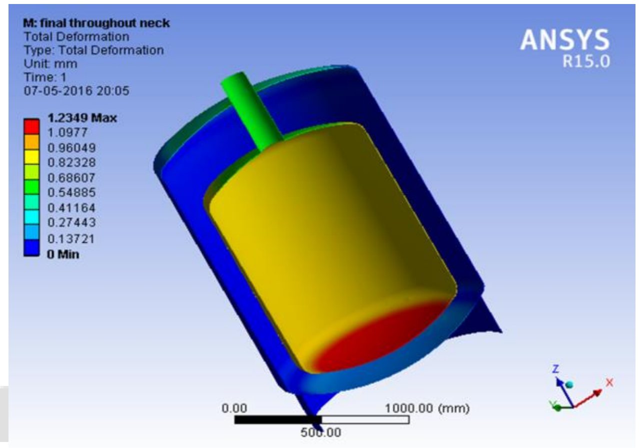


Fig.16 Total Deformation in Dewar

III. FACTOR OF SAFETY OF DEWAR

Fig. 17 shows the safety factor of the Dewar for the design parameters. Here, the safety factor is very high. So, the design of the Dewar is correct.

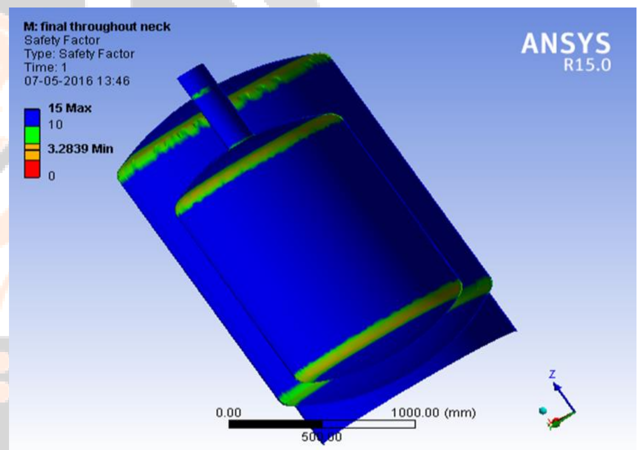


Fig.17 Factor of Safety of Dewar

B. STEADY STATE THERMAL ANALYSIS OF DEWAR

I. NECK

TEMPERATURE DISTRIBUTION IN NECK

Fig 18 shows that the temperature distribution in neck is uniform throughout its length. There is no uneven distribution.

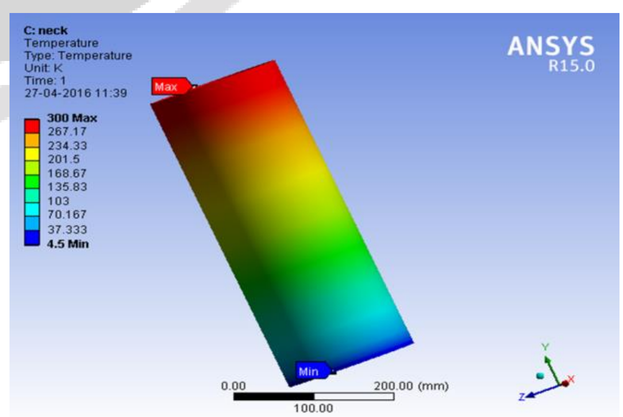


Fig. 18 Temperature Distribution in Neck

HEAT TRANSFER THROUGH NECK

Fig.19 shows that the heat flux Distribution through neck is uniform. And equal to 0.0068 W/mm2. So the total heat transfer through the neck is 5.475 W.

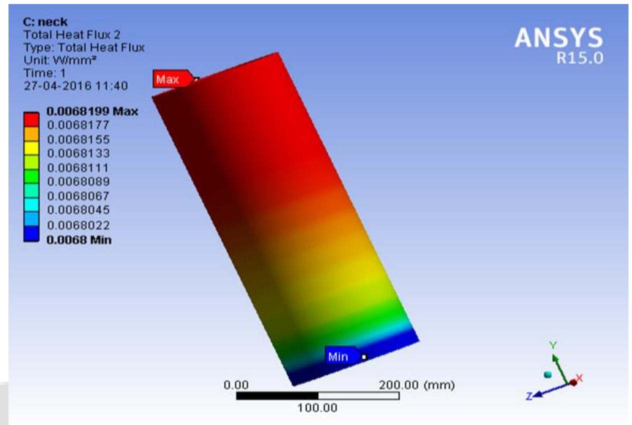


Fig.19 Heat Flux Distribution in Neck

II. LIQUID HELIUM INLET PIPE

TEMPERATURE DISTRIBUTION IN LIQUID HELIUM INLET PIPE

Fig. 20 shows that the temperature distribution in Liquid Helium Inlet Pipe is uniform throughout its length. There is no uneven distribution

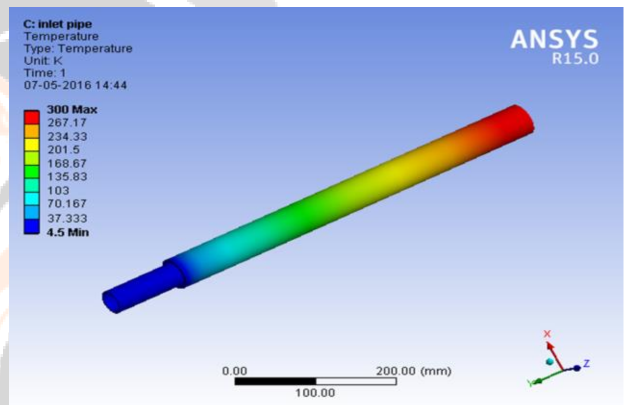


Fig. 20 Temperature Distribution in Liquid Helium Inlet Pipe

HEAT TRANSFER THROUGH LIQUID HELIUM INLET PIPE

Fig. 21 shows that the heat flux Distribution throughout Vacuum Jacket is uniform. And is changed as the cross section area is changed. So the total heat transfer through the Liquid Helium pipe is 1.376 W.

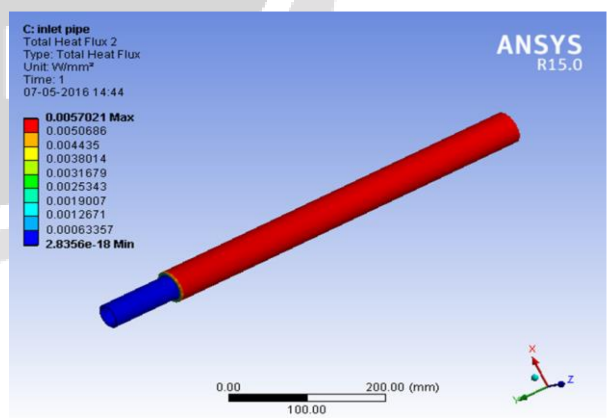


Fig. 21 Heat Flux Distribution in Liquid Helium Inlet Pipe

III. LIQUID HELIUM REMOVAL PIPE

TEMPERATURE DISTRIBUTION IN LIQUID HELIUM INLET PIPE

Fig.22 shows that the temperature distribution in Liquid Helium Removal Pipe is uniform throughout its length. There is no uneven distribution

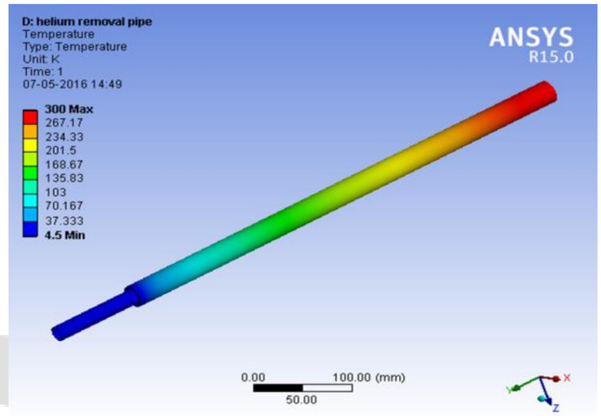


Fig. 22 Temperature Distribution in Liquid Helium Removal Pipe

HEAT TRANSFER THROUGH LIQUID HELIUM REMOVAL PIPE

Fig. 23 shows that the heat flux Distribution throughout Vacuum Jacket is uniform. And is changed as the cross section area is changed. So the total heat transfer through the Liquid Helium removal pipe is 0.739 W.

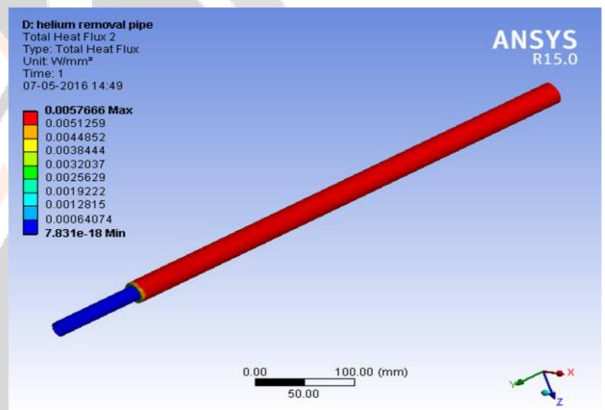


Fig. 23 Heat Flux Distribution in Liquid Helium Removal Pipe

IV. PIPE FOR DIFFERENT INSTRUMENTATION

TEMPERATURE DISTRIBUTION IN INSTRUMENTATION PIPE

Fig.24 shows that the temperature distribution in Liquid Helium Removal Pipe is uniform throughout its length. There is no uneven distribution

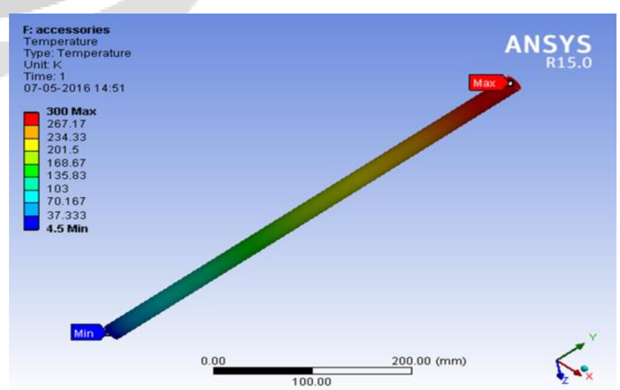


Fig. 24 Temperature Distribution in Instrumentation Pipe

HEAT TRANSFER THROUGH INSTRUMENTATION PIPE

Fig.25 shows that the heat flux Distribution through neck is uniform. And equal to 0.0055 W/mm2. So the total heat transfer through the neck is 0.5610 W.

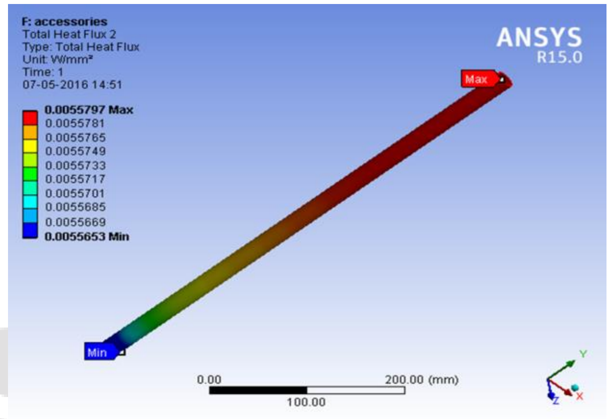


Fig.25 Heat Flux Distribution in Instrumentation Pipe

CONCLUSIONS

The current work is deals with the design of low heat leak liquid helium Dewar. In context to the design requirement it can be concluded that cylindrical shape with Tori spherical Shape is the best shape. The material for the inner vessel is SS304L and for the outer vessel is Carbon steel grade 60, are best material. The dimensions of the different parts of Dewar are determined from calculation.

Result of static structural analysis of Dewar in ANSYS shows that equivalent stress developed due to internal Pressure of Liquid Helium ,external atmospheric pressure and weight of equipment, is throughout the Dewar is 80.392 MPa, which is less than the allowable stress for material as shown in figure 6.7. And deformation is also negligible as shown in figure 6.8. So design is safe against internal pressure, external pressure and weight of equipment.

Dewar store Liquid Helium at very low temperature so reduction of heat load is necessary. For reducing residual gas conduction and radiation effects insulation must require. Multilayer insulation with aluminium foil as Shield material and Dacron Fibre as Spacer material is selected among all types of insulation system. Optimization is also done for MLI system for number of layers and layer density. And optimum layer density is 20 layers/cm and optimum radiation shield is 40 layers.

The total heat transfer to the Dewar due to all type of Heat transfer mode is 9.128 W. and according to the steady state thermal analysis in ANSYS the total Heat Transfer to the Dewar is 9.21 W. So, both are nearly equal. And boil of rate of Liquid Helium is 1.4 % per Day.

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