Calculation of lining parameters for Cryostat designing

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Introduction

The word cryostat is made up of a word whose meaning is cold and stable. Mainly it is a container having a device kept at very low temperature and provides thermal insulation to that device. Sir Dewar was the first who has invented it with double-walled design [1]. Cryogenic liquids are filled inside the cryostat [2]. There should be no heat transfer into the cold vessel of the cryostat. In this paper, we discuss mainly a process to reduce thermal radiation into the cryostat [2] as well as the minimum required materials to achieve minimum heat load. The main function of the cryostat is to provide thermal insulation, mechanical housing of the cryogenic devices and cooling of the device for its safe and reliable operation [1]. Nowadays, large numbers of experiments are planned for various physics goals to operate at liquid N_2/He/Xe temperature. Therefore, it is necessary to study the minimum amount and layering of low activity material to get minimum amount of heat load. We are designing the cryostat considering the basic requirements of the LEGEND Collaboration – The Large Enriched Germanium Experiment for Neutrinoless Double-Beta Decay.

Requirements

Before designing a cryostat various parameters should be known such as working temperature range, shape, physics goals, order of vacuum, material constrains, internal shielding material and heat transfer [3]. Vacuum requirements are very important because of the gases within the two walls of the cryostat may produce conductive heat load. To avoid this type of heat load middle portion of double walled may be evacuated up to a pressure of 10^{-4} mbar [1, 2].

Copper: A most promising materials

Copper is mostly used in the cryostat designing because of its following peculiar properties:
(1) Low emissivity after proper polishing [1].
(2) Rare to brittle even at very low temperatures [3]. (3) Irradiation of Copper by cosmic rays will produce ^{60}Co, which is having shorter lifetime amongst all radio nuclei that are produced by any material other than Copper [3]. (4) Copper can be stored in an underground storage and quickly moved for processing site [3]. (5) The amount of cryogenic liquid used to cool 1 kg of Copper is less in comparison with other cryostat materials [3].

Heat transfer to the Cryostat

Opaque body is a good approximation for the cryostat designing. However, sometimes due to small cracks in the thermal shield, cryostat behaves like a Black Body [1, 4]. Thus, there is a possibility for solid conduction of heat because of the supporting system, convection due to non-perfect vacuum insulation and thermal radiation due to its double-walled design. These are the three ways in which one can consider the occurrence of heat transfer phenomena in the cryostat [1]. Solid conduction can be minimized by the choice of material. A simple method to eliminate convection or gaseous conduction is to remove the gas between the walls [4]. To minimize the thermal radiation one has to consider the radiation exchange between the two surface enclosures of the cryostat [1, 5]. After considering the heat exchange between two enclosures facing each other or oriented at some angle, which is given by the difference of radiation emitted by the first surface, absorbed by the second surface and emitted by the second surface, absorbed by the first surface [1]. The heat transfer between two enclosed surfaces; from one at temperature T_1 with an emissivity ε_1 and surface area A_1, to another at T_2 with ε_2 and A_2, of a cryostat can be given as [2, 4, 5]

$$Q_{12} = \frac{\sigma(T_1^4 - T_2^4)}{(1-\varepsilon_1/\varepsilon_1 A_1)} + \frac{1-\varepsilon_2}{\varepsilon_2 A_2} + \frac{1}{(A_1 F_{12})}. \quad (1)$$

Where σ is the Stefan-Boltzmann's constant: $\sigma = 5.67\times10^{-8}$ (W-m^{-2}-K^{-4}), and Q_{12} is the heat transfer rate from surface 1 to surface 2, F_{12} is the 'view factor', which is the fraction of the heat leaving surface 1 that is intercepted by surface 2. F_{12} is shape-dependent. So the value of heat transfer is also shape dependent i.e., parallel plate, cylindrical or spherical.

Using Eq. (1) and the concept of view factor one can have two Eqs. for these three shapes of the cryostat. For parallel plates $A_1 = A_2 = A$

$$Q_{12} = \frac{\sigma A(T_1^4 - T_2^4)}{(1/\varepsilon_1 + 1/\varepsilon_2 - 1)}. \tag{2}$$

For spherical and cylindrical cryostat [1, 2, 5]

$$Q_{12} = \frac{(\sigma A_2)(T_1^4 - T_2^4)}{(1/\varepsilon_2 + (1/\varepsilon_1 - 1)(A_2/A_1))}. \tag{3}$$

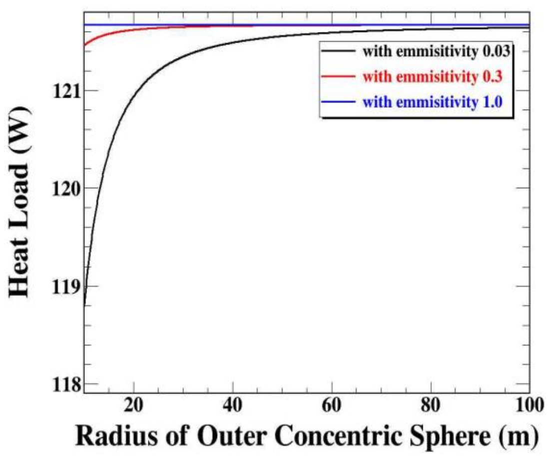

Fig. 1: Heat load in watt variation with respect to the radius of the outer concentric sphere in meter.

It is obvious from Eqs. (2) and (3) that the reduction of heat load on the cold vessel requires to reduce the size of A_1 i.e., the radius of the outer sphere. We studied the heat transfer rate variation with the outer radius and emissivity of the outer surface. As the emissivity of the outer sphere tends towards a blackbody condition heat transfer is increased accordingly. From Figure 1 we may conclude that one should use low emissivity material in the construction of the cryostat and the outer surface should very close to the inner surface [5]. Another way to reduce the thermal radiation load is to add a layer between two surfaces of the cryostat as shown in Figure 2. It shows that the spherical surfaces are more suitable for cryostat designing for holding liquid gases for longer time due to less heat load [5].

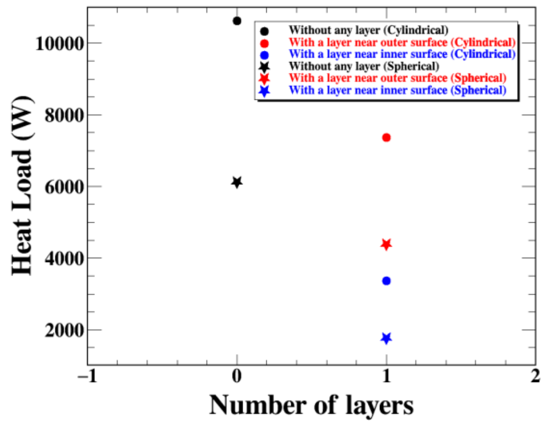

Fig. 2 Heat load variation in case of one layer or no layer in between two surfaces of spherical and cylindrical shaped cryostat.

We may also see that the lining near inner surface is reducing almost two times heat load than the lining close to outer surface.

Acknowledgement

Authors are thankful to the CSIR, DST and MHRD through SPARC, New Delhi for financial support.

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