Thermosyphon and Bath Cooling Techniques for SHe/LHe Heat Exchange in Cryogenic Aluminum Plate-Fin Heat Exchanger

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Heat Exchange in Cryogenic Aluminum Plate-Fin Heat Exchanger

Major Project Report

Submitted in partial fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering (Thermal Engineering)

By

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MAY 2016

Declaration

This is to certify that

1. The thesis comprises my original work towards the degree of Master of Technology in Thermal Engineering at Nirma University and has not been submitted elsewhere for a degree or diploma.

2. Due acknowledgement has been made in the text to all other material used.

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Undertaking for Originality of the Work

I, **Roshan R. Tandel**, Roll. No. **14MMET27**, give undertaking that the Major Project Part 1 entitled "**Thermosyphon and Bath Cooling Techniques for SHe/LHe Heat Exchange in Cryogenic Aluminum Plate-Fin Heat Exchanger**" submitted by me, towards the partial fulfillment of the requirements for the degree of Master of Technology in **Mechanical Engineering (Thermal Engineering)** of Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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This is to certify that the Major Project Report Part 1 entitled "**Thermosyphon and Bath Cooling Techniques for SHe/LHe Heat Exchange in Cryogenic Aluminum Plate-Fin Heat Exchanger** " submitted by **Mr. Roshan R. Tandel (14MMET27)**, towards the partial fulfillment of the requirements for the award of Degree of **Master of Technology in Mechanical Engineering (Thermal Engineering)** of Institute of Technology, Nirma University, Ahmedabad is the record of work carried out by him under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination. The result embodied in this major project, to the best of our knowledge, has not been submitted to any other University or Institution for award of any degree.

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ROSHAN R. TANDEL

Abstract

Aluminum Plate-Fin Heat Exchangers (PFHE) found many applications in the cryogenic process industries, petrochemical plants and refrigeration units ranging from Liquefied Natural Gas (LNG), Air Separation Units (ASU) to Helium Cryogenics. Its compact design offers high heat transfer area to volume ratio and thus great thermal performance. Aluminium PFHE's unique capabilities to provide minimum temperature approach between the streams and accommodate many streams in the same unit gives flexibility in process flow design and small fluid inventory with consequent improvement in control safety and product quality. In the present study the various cooling modes, i.e., bath cooling and natural convection through thermosyphon effect will be studied in detailed for Supercritical Helium (SHe), a hot stream, to Liquid Helium (LHe), a cold stream, heat exchange of 10 kW class dynamic heat load at 4 K temperature level. Thermo-hydraulic design of Aluminium PFHE under the various geometrical parameters like fin type, pitch, number of passes etc. with temperature dependent fluid properties have been carried out. A comparative analytical study have been made for thermosyphon cooling with bath type cooling and the results were compared.

Contents

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Nomenclature

Ah	Effective heat transfer area on SHe side per unit height of heat exchanger
Au	Heat transfer area per unit area of parting plate
B	Breadth
Cp	Specific heat at constant pressure
D	Hydraulic diameter
f	Friction factor
Gm	Mass flux
g	Acceleration due to gravity
H	Height
h	Heat transfer coefficient
j	Colburn factor
Kc	Pressure loss coefficient at entrance of heat exchanger
Ke	Pressure loss coefficient at exit of heat exchanger
k	Thermal conductivity
L	Length
Nu	Nusselt Number
P	Pressure
Pr	Prandlt number
p	Pressure of fluid
Q	Total heat transferred between LHe and SHe
q	Boiling heat flux
q^*	Peak nucleate boiling heat flux
R	Thermal resistance
Ra	Rayleigh Number
Re	Reynold's number
Re^*	Critical Reynold's number
S	Pitch
T	Temperature
ΔT	Log Mean Temperature Difference (LMTD) between LHe and SHe
t	Thickness
W	Width
α	Fin geometrical parameter
δ	Fin geometrical parameter
γ	Fin geometrical parameter

ρ	Density of fluid
τ	Surface tension
λ	Latent heat of vaporization
η	Dynamic viscosity
φ	Un-perforated fraction
σ	Ratio of minimum free flow area to frontal area

Subscripts

a	Average
c	Core
F	Film Boiling
f	Fin / Fin Material
i	Inlet
L	Liquid Helium
l	Liquid
N	Nucleate Boiling
o	Outlet
P	Parting Plate
S	Supercritical Helium
sh	Static Head
T	Total
v	Vapor
X	Heat Exchanger
1	Primary
2	Secondary

Abbreviations

Al	Aluminium
ASME	American Society of Mechanical Engineers
Cu	Copper
DAE	Department of Atomic Energy
GHe	Gaseous Helium
HX	Heat Exchanger
IN-DA	Indian Domestic Agency
LHe	Liquid Helium
LMTD	Log Mean Temperature Difference
OFHC	Oxygen Free High Conductivity
PNBF	Peak Nucleate Boiling Flux
SHe	Supercritical Helium
SS	Stainless Steel
VHe	Vapor Helium

Chapter 1

Introduction

1.1 ITER

ITER, is an experimental fusion reactor being developed presently at Cadarache, in the South of France. It is a step towards production of electricity from fusion energy in future. Nuclear Fusion is the process in which the Sun and the stars produce the energy by fusing light nuclei of hydrogen. ITER will produce at least 10 times more energy than the energy required to operate it. In future with the help of commercial reactors based on fusion, this energy can be converted to electricity.

ITER-India is the Indian Domestic Agency (IN-DA) which is delivering India's contributions towards ITER Project. It is an empowered project within the Institute for Plasma Research, which is an autonomous institute under the Department of Atomic Energy (DAE), Government of India. ITER will be built mostly through in-kind contributions from the different participant countries in the form of components manufactured by these parties and delivered or installed at ITER.

1.1.1 ITER Partners

ITER is a unique collaboration involving more than half of the global humanity. The ITER partners are European Union, India, Japan, China, Russia, South Korea and the United States.

1.1.2 ITER Project

ITER will have a device of 6 metre radius, a magnetic field of 5 Tesla and plasma of volume 840 cubic metre. It will contain 100,000 km of niobium tin superconducting strands and weigh 23,000 tonnes. Niobium tin becomes superconducting at 4.5K (about -269 $°C$), and the superconducting strands will be cooled by superconducting helium (SHe) at 4K. It will keep the superconducting strands just a few degrees above absolute zero. ITER is expected to start generating its 1st plasmas in 2020. But the burning plasma experiments will begin after 2027. One of the huge encounters will be to see whether these self-sustaining plasmas can really be maintained and created without damaging the plasma facing wall or the high heat flux divertor target.

1.2 TOKAMAK

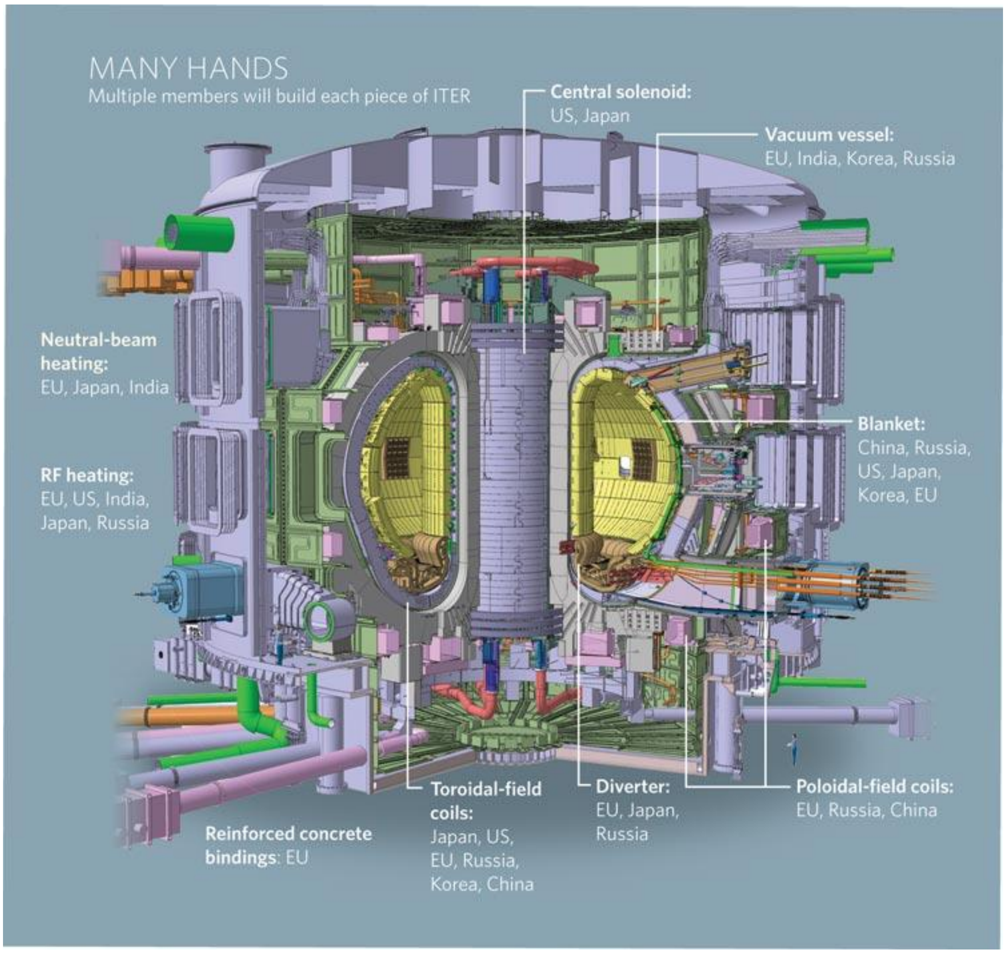

Figure 1.1: Tokamak[1]

- The Tokamak design has been widely adopted because scientists believe it provides the best environment in which to safely and efficiently run a fusion reaction.

- Tokamak is a doughnut-shaped magnetic confinement reactor that suspends a fusion reaction inside the doughnut within electromagnetic fields.

- This is necessary because no solid material has yet been found or created that could withstand the extremely high temperatures, greater than 150 million$°C$ or 15 keV, of fusion plasma.

- Strong magnetic fields are used to keep the plasma away from the walls; these are produced by superconducting coils surrounding the vessel.
- The fuel which is a mixture of deuterium and tritium, is heated to temperatures in excess of 150 million$^{\circ}C$, forming a hot plasma. Strong magnetic fields produced by superconducting coils surrounding the vessel, and by an electrical current driven through the plasma are used to keep the plasma away from the walls.

1.3 Cryogenic System for Fusion Grade Tokamak

The sophisticated cryogenic system is required for proper cool down and temperature control of cryopumps as well as superconducting electro-magnets used in fusion grade tokamak machine. Torus as well as cryostat occupies very huge space. So vacuum pumps used should be reliable and with very high capacity essential for continuous and economical operation of fusion grade tokamak machines. Cryopump is one of the most favourable options for that.

1.3.1 Cryopump

The ITER machine includes three large cryogenic high-vacuum pumping systems.

1. Evacuation and maintenance of the required pressure levels in the torus.
2. Generation of the required vacuum conditions in the neutral beam injectors (NBI) which are used to heat up the plasma by injection of highly energetic accelerated neutral H and D particles.
3. Provision of the insulation vacuum in the cryostat, which houses the superconducting coil system.

It has the advantages over other types of vacuum pumps are as follows.

- 10^{-6} mbar is required in cryostat and 10^{-8} mbar is required inside torus of tokamak machine, cryopump can be used to achieve vacuum of that order
- Pure clean operation without any risk of oil contamination
- Very high pumping capacity
- No moving parts : high reliability and Low maintenance

The cryopump essentially consist of cryopanels which are maintained at very low temperature in the range of 4.5 K during normal operation. In order to achieve such a low temperature, cryo-panels are usually cooled with helium gas. The 80 K shield is also used to cover up 4.5 K cryopanels and to reduce heat load on it. The temperature of 80 K shield is maintained using either liquid nitrogen or 80 K helium gas, depending on the application. The evacuation effect in cryopump is achieved by condensation, solidification and adsorption of the gases present in the space to be evacuated. As gas molecules comes in contact with cryo-panels, its temperature decreases. The condensation of gas molecules followed by solidification on cryo-panels of the cryopump results in decrease in volume and pressure of gas present in the space under the evacuation. If the temperature of cryo-panels is controlled such that it is lower than freezing point of all gas constituents inside the space, then very high vacuum can be achieved. At 4.5 K, except helium, all gases can be condensed. Therefore, helium is most preferred coolant for cryopumps. In order to simplify the cooling process and increase the reliability of the cryopump operation, a single phase coolant fluid is always preferred over two phase fluid. Therefore, supercritical helium (SHe) is predominantly used in cryopumps with carefully controlled pressure, temperature and flow rate for a given heat load to ensure single phase SHe throughout the cooling loop of cryopumps.

1.3.2 SHe to LHe Heat Exchanger

In order to remove heat load from for all cryopumps, SHe to LHe heat exchanger is used. SHe carries heat load and to cool that SHe, Liquid He is used.

For that Bath cooling and Thermosyphon cooling techniques are used. The LHe bath assembly at ITER plant is shown in figure below.

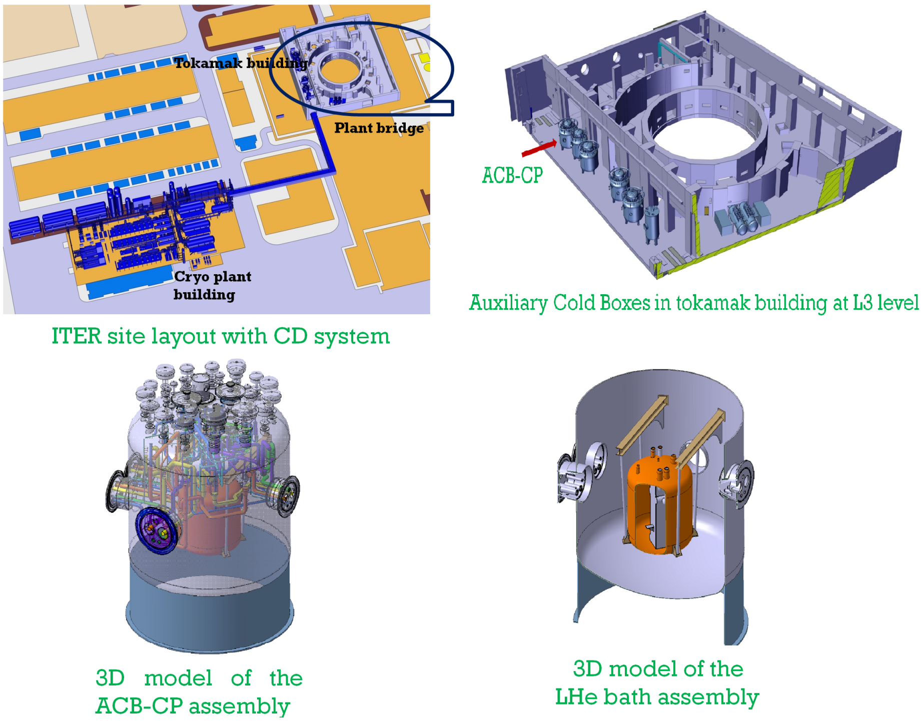

Figure 1.2: LHe bath assembly at ITER plant[2]

1.3.2.1 Bath Heat Exchanger

All heat load from SHe is dumped in the LHe bath. Evaporation of LHe takes place due to heat load received from SHe. For continuous operation the level of LHe bath is required to be maintained. LHe is supplied directly from LHe Dewar and LHe level is maintained as per the requirements.

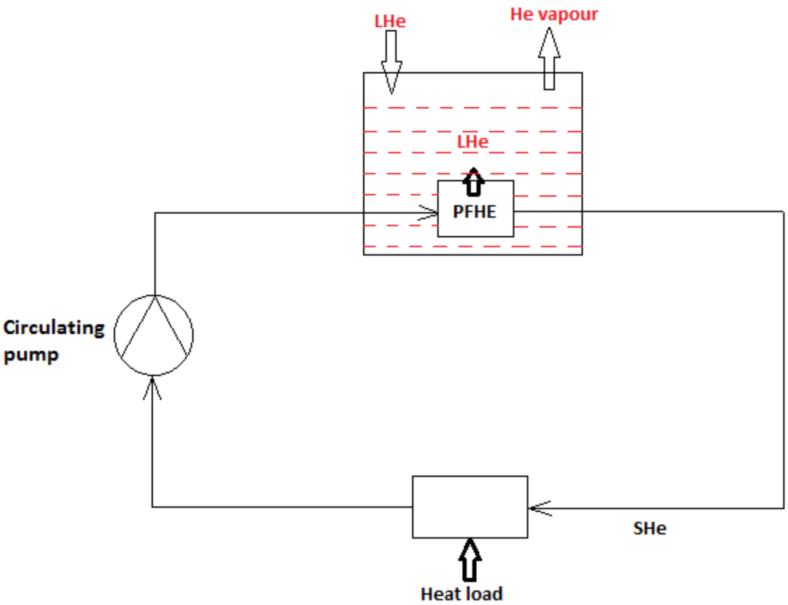

Figure 1.3: Bath Cooling

1.3.2.2 Thermosyphon Cooling

Thermosyphon refers to a method of cooling based on natural convection. LHe takes the heat load from SHe in PFHE by absorbing latent heat and converted into vapour. Thus due to density difference natural circulation of LHe is possible which is called thermosyphon flow. There is no requirement of mechanical pump for the circulation of LHe.

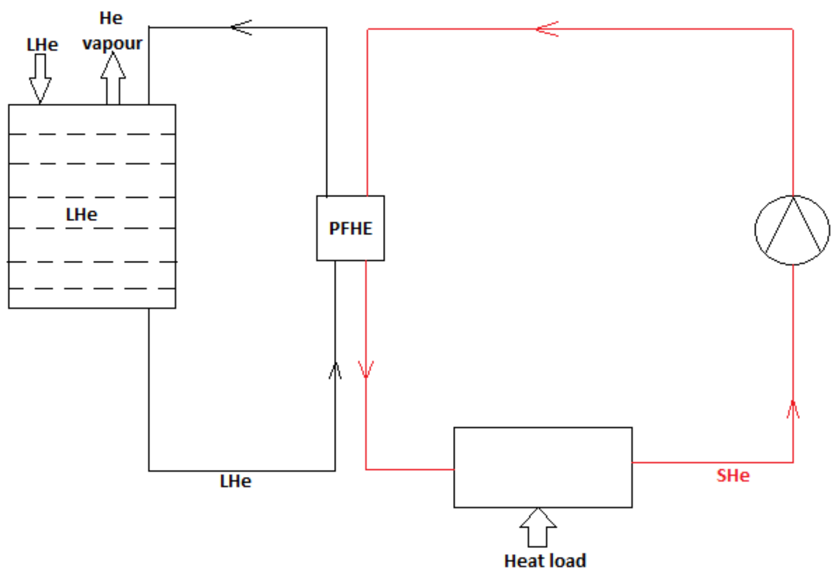

Figure 1.4: Thermosyphon Cooling

1.4 Motivation for Study

Despite more than 60 years of work, researchers have failed to achieve a fusion reaction that produces more energy than it consumes. The ITER project aims to show that nuclear fusion-the power source of the sun and stars-is technically feasible as a source of energy. It is designed to produce 500 MW of fusion power from 50 MW of input power. There is a requirement of SHe to LHe heat exchanger to cool SHe with the help of LHe taking into account the all design requirements. The iterative programming in MS Office Excel has been developed for designing the heat exchanger, perform heat transfer analysis and pressure drop analysis. The helium property data is integrated in the MS Office Excel using software HePak©[3] . There are two options available for cooling SHe : 1. Bath cooling and, 2. Thermosyphon cooling. It is very important to find the best possible option for heat transfer due to the space limitations as it is used in the tokamak machine.

1.5 Objective of Present Work

- The objective of the present work is to develop a design methodology for SHe to LHe heat exchanger for cooling circuit for 10kW class heat load at 4.5K temperature.
- Apply suitable correlations to carry out heat transfer and pressure drop analysis and find out the size the heat exchanger considering design constraints.
- Analytical study of two cooling techniques, first is bath cooling and second is thermosyphon cooling used in SHe to LHe heat exchanger.
- Investigate, the best possible option for cooling by comparing their designs.

Chapter 2

Literature Review

The literature survey has been carried out for the heat exchanger used in cryogenic applications and different heat transfer phenomena for heat exchange from SHe to LHe .

2.1 Literature Review for SHe to LHe heat exchanger

- The cryogenic heat exchangers work at temperatures typically less than 123 K. The major applications of cryogenic heat exchangers are in air separation plant, LNG plants, helium liquefiers, cooling systems for cryopumps and superconducting magnets, cryogen vaporisers, space applications, food industries, medical appliances etc. Cryogenic heat exchangers for helium applications are made compact and operated at high NTU conditions. They need to work with very high effectiveness. In many applications, they are equipped inside vacuum chamber to reduce ambient heat load. They are designed with special materials having desirable properties at extreme low temperatures.

- Typical cryogenic heat exchangers working at LHe temperatures having small temperature approaches and hence, very compact design with highest possible efficiency is required. Considering the functional requirements of helium heat exchangers, priority is often given to quality of material, compactness and highly efficient design than cost. Very few industries worldwide having idea how to design and manufacture cryogenic heat exchangers which shows the complexity of cryogenic heat exchangers for helium application.

- The literature survey has been carried out to investigate the past references available for SHe to LHe heat exchangers. It has been observed that though numbers of references are available for cryogenic heat exchangers, very few references are available for SHe to LHe heat exchangers. Also, it was found that some literature is reported regarding the investigation of heat transfer parameters in LHe for different types of surfaces, various correlations for nucleate boiling, PNBF and film boiling in LHe under different experimental conditions as well as specific correlations for thermo-hydraulic and heat transfer analysis in SHe.

- No proper reference is available for complete design of SHe to LHe heat exchanger in the literature due to following reasons.
  - It has limited application in the present scenario and limited availability of LHe and its very high cost, no R&D activities related to design and manufacturing of these heat exchangers are carried out.
  - Very few references are available for heat transfer in LHe and She, which are important for design and optimization of heat exchanger.
  - Few number of such heat exchangers exists in the world.
  - Few available references clearly indicate preference for plate fin heat exchanger over other types of heat exchangers for similar type of application.
  - SHe to LHe heat exchangers are designed and manufactured by very few industries (e.g. Nordon Cryogenie) worldwide and their designs are not reported in the literature by these industries.

2.2 Literature Review for Heat Transfer Phenomena in Exchange of Heat from SHe to LHe

- T Kato et al.[4] has investigated heat transfer characteristics of plate fin type Supercritical liquid Helium heat exchanger. The heat exchanger was aluminium plate fin type heat exchanger. For theoretical estimation of heat transfer co efficient, heat transfer correlations were used by T Kato et al are as follow:
  - For single phase flow of SHe and LHe, heat transfer co efficient is expressed by Dittus Boelter equation.
  - Schrock-Grossman's (film boiling and forced convection vapour) equation for two phase flow with quality less than 0.2 .
  - Schrock-Grossman's (forced convection vapour) equation for two phase flow with quality range of 0.2 to 0.75 .
  - Miropolskii's (mist flow) equation used for two phase flow with quality more than 0.75 .

  - The numerical simulation of heat exchanger was also carried out which shows good agreement with experimental data. Effective heat transfer coefficient for supercritical helium was calculated as constantly 20% higher than given by Dittus Boelter equation.

- B. Jager et al.[5] has investigated the heat transfer characteristics in LHe for industrially manufactured cryogenic plate heat exchanger. The heat flux on surfaces with and without fins was investigated. In the experiment measurement of temperature difference between LHe bath and plain surface as well as between LHe bath and finned surface of plate fin heat exchanger was taken. Electric heaters were used to control the heat flux to bath.

  - In experiment, nucleate boiling regime was observed up to 0.2 W/cm^2 where transition took place to film boiling. For smooth surface, heat flux was estimated using correlations by Kutateladze, Been and Westewater, which gave satisfactorily agreement with experimental results for fluxes less than 0.2 W/cm^2.
  - For LHe-I at a pressure of 1 atm. and temperature difference up to 0.4 K (flux value up to 0.2 W/cm^2), there was no difference in heat transfer between smooth surface and finned surface.
  - However, for temperature difference of more than 0.4 K or flux more than 0.2 W/cm^2, use of finned surface improved heat transfer compared to smooth surface.
  - All the experiments performed by B. Jager et al. have been used as design data for some of the heat exchangers required for the cryomagnetic system of TORE SUPRA Tokamak.

- V. A. Grigoriev et al[6] presented an experiment on pool boiling heat transfer in LHe for SS and copper cylindrical geometries with diameter 8 mm and the surface roughness of cylinders was 0.7 and 7 micron.

  - The thermal properties of the heating surface influenced the helium nucleate and transition boiling.
  - There was no nucleate boiling zone on the surface with low degree of roughness(0.7 micron) and experimentally validated the fact that the rough surfaces favours better heat flux compared to smooth surfaces in nucleate boiling heat transfer applications.

- Dinaburg L. B. et al[7] has investigated the LHe boiling heat transfer for vertical copper cylinders of 25 mm diameter and 120 mm length. The temperature range for the experiment was 0.01 K to 0.7 K and the values of critical heat flux was observed in the range of 0.42 W/cm^2 to 0.48 W/cm^2 for Cu and Al cylinders.

- M. Jergel et al[8] has investigated the heat flux characteristics in LHe bath for surfaces with different orientations. For vertical surface made of OFHC copper, the PNBF observed was approximately 0.65 W/cm^2 while for Aluminium surface it was close to 0.9 W/cm^2.
  - For the finned surface, the PNBF value is close to 0.45 W/cm^2 which is almost two times PNBF value for smooth surface which shows the advantage of finned surface over smooth surface for the LHe pool boiling heat transfer applications.
  - Kutateladze correlation fairly estimates the value of heat flux.
  - PNBF has been estimated using correlation by Zuber et al.
  - For stable film boiling, Breen and Westwater correlation can be used with good accuracy.
- Steven W. Van Sciver[9] has recommended
  - equation by Kutateladze for nucleate pool boiling.
  - equation by Zuber et al for estimation of PNBF.
  - equation by Breen and Westwater for stable film boiling.

2.3 Literature Review for Thermosyphon Cooling Technique

From the basic concept of natural convection, the idea of transporting a large amount of thermal energy from one location to another was first applied by Schmidt[10] during second world war. He applied this idea for cooling down gas turbine rotor blades, having inside radial holes, closed at one end and opened at another forming an open thermosyphon.

To earlier investigators, the open system had been more attractive, because it was capable of producing relatively high heat transfer rates. Lighthill[11] theoretically examined the flow regimes existing in an open thermosyphon. Martin and Cohen[12] have reported that the interface between hot and cold fluid streams moving in opposite directions, decrease the heat transfer rates significantly.

Cohen and Bayley[13] came out with the two phase closed thermosyphon as the most suitable system to achieve a better solution of the heat transfer problems of liquid cooled gas turbine blades. With water as the enclosed coolant, they noted that the heat transfer from the heated to the cooled end wad independent of the quantity of fluid inside the thermosyphon, until this quantity was reduced to one percent. The effect of increase in tube diameter and saturation pressure was to raise the heat transfer coefficient.

Schmidt[14] experimented with the fluids near their critical states for the problem of cooling the gas turbine blades. At this state, the coefficient of thermal expansion increase considerably while the viscosity of all liquids has its lowest possible value. For studying the behaviour of such an application he used a vertical tube with critical quantity of fluid, heated below and cooled above, at the critical operating temperature and found that the apparent heat conductivity of the fluid reaches values of 4000 times that of a solid copper rod of the same outer diameter.

Bewilogua and Knoner[15] used two phase system as a nitrogen cryostst for operation in horizontal reaction channel and found it very suitable.

Long[16] used this system to preserve the permafrost under building foundations in Alaska, but did not pay much attention to the effect of various parameters governing the heat transfer characteristics of the system.

2.4 Conclusion from Literature Review

Plate-fin heat exchanger is more preferable than any other type of heat exchangers especially with regard to compactness to save installation space and highly efficient design for heat transfer. Use of finned surface can improve the heat transfer compared to smooth surface. Aluminium is the material used in cryogenic heat exchangers due to good thermal conductivity at cryogenic temperature and low in weight. Manglik and Bergles correlation gives the most comprehensive data for calculating j and f factors. Kutateladze correlation fairly estimates the value of nucleate boiling heat flux on LHe side. Stable film boiling on vertical surface can be described by Labunstov correlation. A two phase thermosyphon has the capability of transferring thermal energy several hundred times greater than a solid conductor of same geometry, so it can be a better option for cooling.

Chapter 3

Design Parameters for SHe to LHe Heat Exchanger

3.1 Thermo-hydraulic Input Data

Table 3.1: Thermo-hydraulic Input Parameters[17]

Parameters	Hot fluid (SHe)	Cold fluid (LHe)
Inlet temperature (K)	5.16	4.2
Inlet pressure (bar)	3.38	0.99
Outlet temperature (K)	4.3	4.2
Mass flow rate (kg/s)	1.36	-
Allowable pressure drop (mbar)	40	20

3.2 General Design Requirements

The general design requirements for the heat exchanger are summarised below.

1. Heat exchanger is going to be placed inside a valve box where space is limited. So,it should be very compact. Total height of heat exchanger should not be more than 1.0 meter.

2. Heat exchanger weight should be as less as possible.

3. Heat exchanger should have economically feasible design and it should be highly reliable .

4. Manufacturing of heat exchanger should be easy.

3.3 Selection of Heat Exchanger Type

The plate-fin heat exchangers are a standout amongst the most positive conceivable outcomes especially with regard to compactness to save installation space and costs. It has such a variety of advantages over alternate sorts of heat exchangers listed below:

Investment cost is low due to :

- Low weight aluminium
- Compactness in design

Ideal execution because of:

- Highly effective heat transfer
- Tailor-made design

3.3.1 Plate Fin Heat Exchanger

Plate fin exchanger is one a sort of compact heat exchanger. In this the heat exchange surface zone is improved by giving extended metal surfaces. These extended metal surfaces are also known as fins and interfaced between the two fluids. Plate fin heat exchangers are exceptional due to their performance and unrivaled construction. They are characterized by compactness, high effectiveness, moderate cost and low weight . As the name suggests, PFHE is a type of compact exchanger that consists of a stack of alternate flat plates called parting sheets. These parting sheets and corrugated fins are brazed together as a block. Flow treams exchange heat by flowing along the passages which are made by the fins between the parting sheets. Separating plates act as the primary heat transfer surfaces and the fins act as the secondary heat transfer surfaces. Fins not only form the extended heat transfer surfaces, but also providing structural supports against internal pressure difference. The side bars act as a wall to prevent the fluid from mixing with the second fluid, leaking to outside or spilling over. The fins and side bars are brazed with the parting sheets to provide good thermal link and mechanical stability.

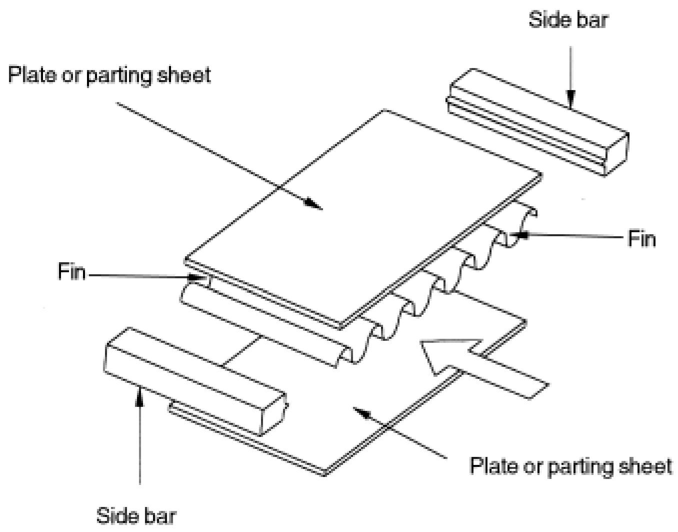

Figure 3.1: Constructional Details of Plate Fin Heat Exchanger[18]

The plate fin heat exchanger is selected with following flow specifications :

- Flow arrangement : Counter flow
- SHe flow : Vertically downwards from top to bottom
- LHe / VHe flow : Supply of LHe from bottom and VHe vent from top

3.4 Selection of Material

Materials used in heat exchangers for helium cryogenic applications are: Aluminium (Al) alloy, OFHC copper (Cu), and stainless steels (SS).

Stainless steels (SS) has very low thermal conductivity compared to Aluminium (Al) alloy and OFHC copper (Cu) .

OFHC copper (Cu) has higher thermal conductivity than Aluminium (Al) alloy, but its density is almost 4 times more than that of Al. So, the weight of copper heat exchanger will be much higher than Al heat exchanger.

Aluminium alloy is advantageous considering strength, fabrication and weight factors compared to OFHC copper (Cu).

Compared to cryogenically compatible material like SS and Copper, Aluminium is light in weight. A typical plate fin heat exchanger have 25 times more surface area per weight compared shell and tube heat exchanger[19]. Also, weight per unit heat transfer is less and very high thermal effectiveness can be achieved[19].

ASTM Al 3003 is widely used for fabrication of plate fin type cryogenic heat exchangers and other equipment in gas-liquefaction industries. ASTM Al 3003 is selected for fins and parting plate of heat exchanger due to the advantages :

- Easily available in the market
- Manufacturing is easy using brazing/welding
- Light weight
- Good corrosion resistance.
- Good machinability
- Good thermal conductivity at cryogenic temperature

The thermal conductivity of Al 3003 is 6.86 W/m-K at 4.5 K, and date has been taken from NIST database[20].

3.5 Selection of Fin Type

Different types of fin geometries play an important role not only on heat transfer density, but also in controlling the pressure drop of the fluid. Each of them having their own advantages and limitations. Different types of fin geometries are shown in figure below.

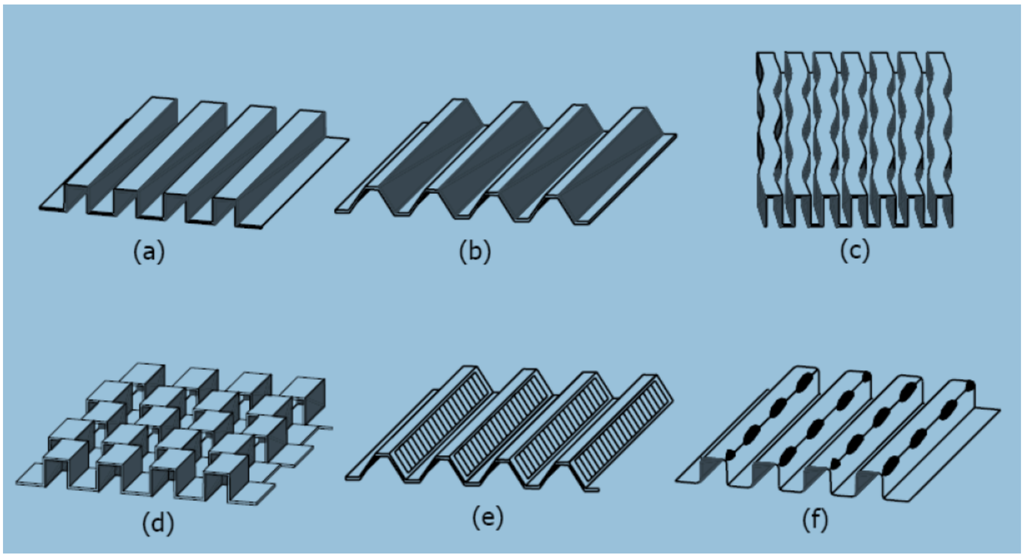

Figure 3.2: (a) Plain Rectangular Fin (b) Plain Triangular Fin (c) Wavy Fin (d) Offset Fin (e) Multi Louver Fin (f) Perforated Fin[21]

Comparison among different types of fins is given below in the table :

Table 3.2: Comparison of Different Fin Types for Plate Fin Heat Exchangers

Fin type	Applications	Relative pressure drop	Relative heat transfer
Plain (rectangular/ triangular)	General Applications	Lowest	Lowest
Offset	Cryogenics, Gas streams in air separation plants	Highest	Highest
Wavy	High pressure streams, Gas streams in hydrocarbon and natural gas applications	High	High
Perforated	Boiling liquid	Low	Low

Due to high heat transfer and high compactness requirement, offset type fin is selected for SHe side as well as LHe side in plate fin heat exchanger. It consists of a type of interrupted surface, which may be visualised as a set of plain fins cut normal to the flow direction at regular intervals. Each segment being offset laterally by half the fin spacing. Surface interruption enhances heat transfer by two independent mechanisms.

- It prevents the continuous growth of thermal boundary layer by periodically interrupting it. The thinner boundary layer offers lower thermal resistance compared to continuous fin types.
- Above a critical Reynolds number, interrupted surfaces offer an additional mechanism of heat transfer enhancement. Oscillations in the flow field in the form of vortices shed from the trailing edges of the interrupted fins enhance local heat transfer by continuously bringing in fresh fluid towards the heat transfer surfaces.

This enhancement is accompanied by an increase in pressure drop. The heat transfer performance of offset strip fin is often as much as 5 times that of a plain fin surface of comparable geometry, but at the expense of higher pressure drop.

Chapter 4

Design of SHe to LHe Plate Fin Heat Exchanger

Offset type fin is selected for SHe side as well as LHe side in plate fin heat exchanger. The geometry of the offset strip fin surface is described by the following parameters:

1. Fin spacing (S_f) (excluding the fin thickness)
2. Fin height (H_f) (excluding the fin thickness)
3. Fin thickness (t_f) and
4. Strip length (L_f) in the flow direction

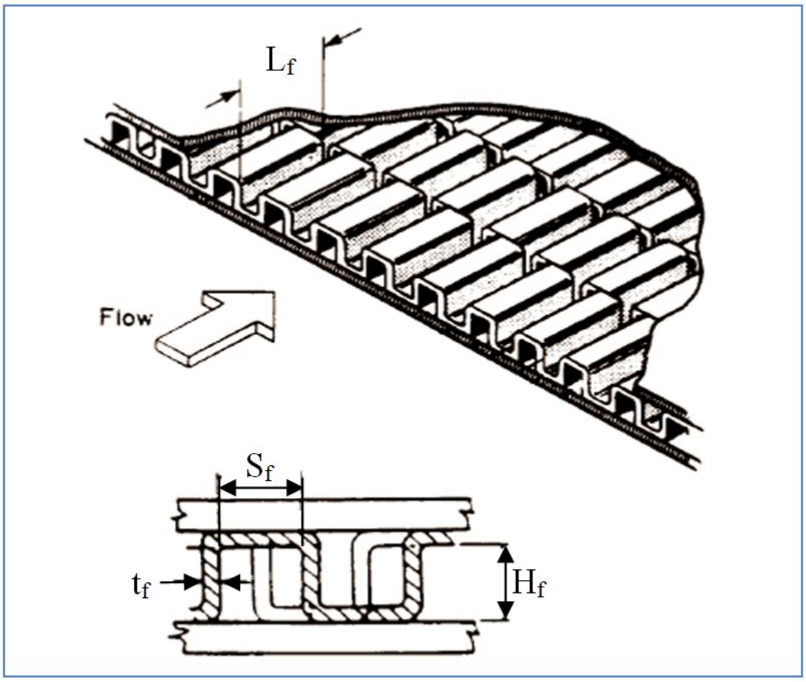

Figure 4.1: Geometrical Parameters of Offset Fin[21]

The dimensional and geometrical parameters of standard offset fins[22] of Al 3003 material, for the design of heat exchanger, are listed in table below.

Table 4.1: Offset Fin Dimensional and Geometrical Parameters

Catalog No.	Fin Length L_f (mm)	Height H_f (mm)	Pitch S_f (mm)	Thickness t_f (mm)
L-3-100-15	12.7	2.5	1.4	0.30
L-3-125-15	12.7	3.2	1.4	0.30
L-6-187-15	25.4	4.7	1.4	0.30
L-9-250-15	38.1	6.4	1.4	0.30
L-9-375-15	38.1	9.5	1.4	0.30

For the compact design of plate fin heat exchanger,smaller fin has been selected as it will result in higher heat transfer are per unit volume which is necessary. However, the smaller fin also causes higher pressure drop. The smallest fin (i.e. L-3-100-15) from table 4.1 has been selected for the of design calculations on both SHe and LHe side.

Other parameters are defined as below for the selected fin.

$\alpha = S_f/H_f$

$\delta = t_f/L_f$

$\gamma = t_f/S_f$

No. of flow passages on LHe side = No. of flow passages on SHe side – 1

Let's take no. of flow passages on SHe side = 50

So, no. of flow passages on LHe side = 49

4.1 Heat Transfer Analysis

Heat transfer analysis is carried out for both SHe side and LHe side separately because both are having different phenomena. LHe enters PFHE from bottom and leaves from top as VHe. Due to phase change of LHe, there is a change in properties along the height of heat exchanger. In order to increase the accuracy of the heat exchanger sizing calculations and to take into account the different phenomenon of phase change, the heat exchanger is divided in to 10 finite number of control volumes in the direction of SHe/LHe flow.

Table 4.2: Heat Exchanger Control Volumes

Control Volume Number	SHe Inlet Temperature (K)	SHe Outlet Temperature (K)	LHe Temperature (K)	LMTD
1	5.16	5.08	4.2	0.92
2	5.08	4.99	4.2	0.83
3	4.99	4.90	4.2	0.75
4	4.90	4.82	4.2	0.66
5	4.82	4.73	4.2	0.57
6	4.73	4.65	4.2	0.49
7	4.65	4.56	4.2	0.4
8	4.56	4.47	4.2	0.31
9	4.47	4.39	4.2	0.23
10	4.39	4.30	4.2	0.14

4.1.1 Heat Transfer on SHe Side

Firstly, Colburn factor J is calculated to analyse the heat transfer on SHe side.

- **j and f factors**

To estimate the Colburn factor and friction factor of fluid flowing through offset fins, different correlations are used by different scientists.The heat transfer coefficient for SHe depends on Colburn factor.Colburn factor is a function of Reynold's number and fin geometrical parameters.Some of the commonly used correlations are described below.

- **Kays Correlation**

Kays[23] initiated the analytical modelling of offset fins for heat transfer and pressure drop estimation. This model is valid for purely laminar boundary layer flow over an offset finned surface.

$$j = 0.665 \times Re^{-0.5}$$

$$f = 0.44 \times \frac{t_f}{L_f} \times 1.328 Re^{-0.5}$$

The limitation of these correlations are that it is not applicable in turbulent boundary layer and the heat transfer is independent of fin geometrical parameters.

- **Wieting Correlation**

Wieting[23] developed an empirical correlations from experimental heat transfer and flow friction data on 22 offset strip fin surfaces of Kays and London, London and Shah, Walters etc over two Reynolds number ranges : Re$<$ 1000 and Re $>$2000.

For $Re < 1000$ (i.e. when flow is laminar)

$$j = 0.483 \times \left(\frac{L_f}{D}\right)^{-0.162} \times \alpha^{-0.184} \times Re^{-0.536}$$

$$f = 7.661 \times \left(\frac{L_f}{D}\right)^{-0.384} \times \alpha^{-0.092} \times Re^{-0.712}$$

For $Re > 2000$ (i.e. when flow is turbulent)

$$j = 0.242 \times \left(\frac{L_f}{D}\right)^{-0.322} \times \left(\frac{t_f}{D}\right)^{0.089} \times Re^{-0.368}$$

$$f = 1.136 \times \left(\frac{L_f}{D}\right)^{-0.781} \times \left(\frac{t_f}{D}\right)^{-0.534} \times Re^{-0.198}$$

$$D = \frac{2 \times S_f \times H_f}{S_f + H_f}$$

The limitation of these correlations is poor definition of hydraulic diameter.

- **Joshi and Webb Correlation**

Joshi and Webb[23] conducted flow visualization experiments to identify the transition from laminar flow. They developed an analytical model in the laminar zone based on the numerical solution done by Sparrow and Liu and a semi empirical method has been used for the turbulent region. The empirical correlations for j and f factors were verified with experimental data on 21 heat exchanger geometries.

For $Re < Re*$(i.e. when flow is laminar)

$$j = 0.53 \times \left(\frac{L_f}{D}\right)^{-0.15} \times \alpha^{-0.14} \times Re^{-0.5}$$

$$f = 8.12 \times \left(\frac{L_f}{D}\right)^{-0.41} \times \alpha^{-0.02} \times Re^{-0.74}$$

For $Re \geqslant Re* + 1000$ (i.e. when flow is turbulent)

$$j = 0.21 \times \left(\frac{L_f}{D}\right)^{-0.24} \times \left(\frac{t_f}{D}\right)^{0.02} \times Re^{-0.4}$$

$$f = 1.12\times\left(\frac{L_f}{D}\right)^{-0.65} \times \left(\frac{t_f}{D}\right)^{0.17} \times Re^{-0.36}$$

Where,

$$Re* = 257 \times \left(\frac{L_f}{S_f}\right)^{1.23} \times \left(\frac{t_f}{L_f}\right)^{0.58} \times D\times \left[t_f + 1.328\left(\frac{Re}{L_f \times D}\right)^{-0.5}\right]^{-1}$$

$$D = \frac{4 \times S_f \times H_f \times L_f}{2\left[(S_f \times L_f) + (H_f \times L_f) + (t_f \times H_f)\right] + [t_f \times S_f]}$$

- **Mochizuki Correlation**

Mochizuki et al[23] correlations are reworking of the Wieting correlations, with modifying its parameters to fit their own experimental data for five scaled up rectangular offset strip fin surfaces. Only fully laminar flow and fully turbulent flow are considered, with an abrupt change of flow regime at Re=2000.

For $Re < 2000$ (i.e. when flow is laminar)

$$j = 1.37\times\left(\frac{L_f}{D}\right)^{-0.25} \times\alpha^{-0.184}\times Re^{-0.67}$$

$$f = 5.55 \times \left(\frac{L_f}{D}\right)^{-0.32} \times \alpha^{-0.092}\times Re^{-0.67}$$

For $Re > 2000$ (i.e. when flow is turbulent)

$$j = 1.17 \times \left(\frac{L_f}{D} + 3.75\right)^{-1} \times\left(\frac{t_f}{D}\right)^{0.089} \times Re^{-0.36}$$

$$f = 0.83 \times \left(\frac{L_f}{D} + 0.33\right)^{-0.5} \times\left(\frac{t_f}{D}\right)^{-0.534} \times Re^{-0.2}$$

$$D = \frac{2 \times S_f \times H_f}{S_f + H_f}$$

- **Manglik and Bergles Correlation**

Manglik and Bergles[23] examined the heat transfer and friction data for 18 offset strip fin surfaces given by Kays and London, Walters & London and Shah, and analyzed the effect of various geometrical attributes of offset strip fins.

For $Re < Re*$(i.e. when flow is laminar)

$$j = 0.6522 \times Re^{-0.5403} \times \alpha^{-0.1541} \times \delta^{0.1499} \times \gamma^{-0.0678}$$

$$f = 9.6243 \times Re^{-0.7422} \times \alpha^{-0.1856} \times \delta^{0.3053} \times \gamma^{-0.2659}$$

For $Re \geqslant Re* + 1000$ (i.e. when flow is turbulent)

$$j = 0.2453 \times Re^{-0.4063} \times \alpha^{-0.1037} \times \delta^{0.1955} \times \gamma^{-0.1733}$$

$$f = 1.8699 \times Re^{-0.2993} \times \alpha^{-0.0936} \times \delta^{0.682} \times \gamma^{-0.2423}$$

Where,

$$Re* = 257 \times \left(\frac{L_f}{S_f}\right)^{1.23} \times \left(\frac{t_f}{L_f}\right)^{0.58} \times D \times \left[t_f + 1.328\left(\frac{Re}{L_f \times D}\right)^{-0.5}\right]^{-1}$$

$$D = \frac{4 \times S_f \times H_f \times L_f}{2\left[(S_f \times L_f) + (H_f \times L_f) + (t_f \times H_f)\right] + [t_f \times S_f]}$$

These are the most comprehensive data for j and f factors of offset fin geometries. These correlations are also experimentally validated for compact plate fin heat exchanger designed and tested by M. Goyal et al[24] for helium liquefaction/refrigeration system.

- **Corberan Correlation**

Corberan et al[25] published new correlations for friction factor for offset fin geometry. These correlations matches with results obtained during their experiment.The correlations are mentioned below.

For $Re \leq 200$,

$$f = \frac{31}{Re}$$

For $Re > 200$,

$$f = -0.02285607 + \frac{44.11786552}{Re} + \frac{3.60647885}{\sqrt{Re}} + \frac{0.6810599}{Re^{0.2}}$$

- **Estimation of Heat Transfer Coefficient**

The convective heat transfer coefficient for SHe is given by,

$$h_s = \frac{j \times Gm \times Cp}{Pr^{\frac{2}{3}}}$$

- j and f factors are calculated based on the Manglik and Bergles correlation. Convective heat transfer coefficient for SHe is shown in table below.

Table 4.3: Heat Transfer Coefficient on SHe Side

Control Volume No.	j - Factor	f - Factor	h (W/m^2K)
1	0.0027	0.0109	776.250
2	0.0027	0.0109	750.995
3	0.0027	0.0110	729.411
4	0.0028	0.0111	710.609
5	0.0028	0.0111	693.842
6	0.0028	0.0112	678.595
7	0.0028	0.0112	664.488
8	0.0028	0.0113	651.227
9	0.0028	0.0113	638.574
10	0.0029	0.0114	626.277

4.1.2 Heat Transfer on LHe Side

Pool boiling heat transfer occurs on LHe side.

The methodology adopted for the estimation of pool boiling heat transfer coefficient on LHe side is summarised below.

1. Estimation of PNBF using correlation described by Zuber et al[9].
2. Estimation of nucleate boiling heat flux using correlation described by Kutanteladze[9].
3. Estimation of nucleate boiling heat transfer coefficient as follows.$h_L = \frac{q_N}{LMTD}$
4. Estimation of film boiling heat transfer coefficient by using correlation described by Labuntsov.

- **Nucleate Boiling Heat Transfer**

Minimum temperature difference between SHe and LHe = 4.34 – 4.2 = 0.14 K

Maximum temperature difference between SHe and LHe = 5.12 – 4.2 = 0.92 K

The temperature of LHe bath is maintained at 4.2 K.

For the range of T between 0.1 K and 1 K, nucleate boiling heat transfer up to PNBF can be best described by the Kutanteladze[9] equation as follows.

$q_N = 1.9 \times 10^{-9} \times \left[g \times \left(\frac{\rho_l}{\eta_l}\right)^2 \times X^3\right]^{0.3125} \times \left(\frac{P \times X}{\tau}\right)^{1.75} \times \left(\frac{\rho_l}{\rho_v}\right)^{1.5} \times \left(\frac{Cp}{\lambda}\right)^{1.5} \times \left(\frac{k_l}{X}\right) \times (T_P - T_L)^{2.5}$

Where,

$$X = \left(\frac{\tau}{\rho_l \times g}\right)^{0.5}$$

- **Peak Nucleate Boiling Flux (PNBF)**

The peak nucleate boiling heat flux for LHe can be described by the correlation developed by Zuber et al[9] as follows.

$$q^* = \left(\frac{\Pi}{24}\right) \times \lambda \times \rho_v \times \left[\frac{g \times \sigma \times (\rho_l - \rho_v)}{\rho_v^2}\right]^{\frac{1}{4}} \times \left(\frac{\rho_l}{\rho_l + \rho_v}\right)^{\frac{1}{2}}$$

- **Film Boiling Heat Transfer**

Film boiling heat transfer coefficient given by Breen and Westwater[9] is as follows :

$$h = \frac{0.37 + \left[0.28 \times \left(\frac{\tau}{g \times D^2 \times (\rho_l - \rho_v)}\right)^{\frac{1}{2}}\right]}{\left(\frac{\tau}{g \times (\rho_l - \rho_v)}\right)^{\frac{1}{8}} \times \left[\frac{(T_P - T_L) \times \eta_v}{k_v^3 \times \rho_v \times (\rho_l - \rho_v) \times g \times \lambda_1}\right]}$$

Where,

$$\lambda_1 = \frac{[\lambda + (0.34 \times Cp \times (T_P - T_L))]^2}{\lambda}$$

With increase in heat flux, the unstable film boiling occurs till vapour layer completely covers the heated surface. Once this is achieved, stable film boiling starts. Labuntsov gave correlation for stable film boiling on vertical surface which was experimentally validated by V. I. Deev et al[26].

$$Nu = 0.25 \times Ra^{0.33}$$

$$h = \frac{Nu \times k_v}{H_x}$$

Where,

$$Ra = \frac{g \times H_x^3 \times Cp \times \rho_v \times (\rho_l - \rho_v)}{\mu_v \times k_v}$$

The applicable heat transfer zone and selected value of heat transfer coefficient are shown in table below.

Table 4.4: Pool Boiling Heat Transfer Zones in LHe

Control Volume No.	Nucleate Boiling q (W/cm^2)	Nucleate Boiling h (W/m^2K)	PNBF q^* (W/cm^2)	Film Boiling q (W/cm^2)	Film Boiling h (W/m^2K)	Applicable Heat Transfer Zone	Selected value of h (W/m^2K)
1	6.121	66500.594	0.578	0.0561	444.650	Film Boiling	444.650
2	4.840	58032.065	0.578	0.0517	447.589	Film Boiling	447.589
3	3.736	49959.484	0.578	0.0473	450.649	Film Boiling	450.649
4	2.798	42303.555	0.578	0.0427	453.857	Film Boiling	453.857
5	2.018	35088.694	0.578	0.0381	457.237	Film Boiling	457.237
6	1.385	28344.347	0.578	0.0334	460.807	Film Boiling	460.807
7	0.889	22107.107	0.578	0.0284	464.578	Film Boiling	464.578
8	0.519	16424.514	0.578	0.0233	477.445	Nucleate Boiling	16424.514
9	0.261	11363.071	0.578	0.0180	481.049	Nucleate Boiling	11363.071
10	0.101	7031.646	0.578	0.0122	484.164	Nucleate Boiling	7031.646

4.2 Sizing of Plate Fin Heat Exchanger

In sizing of plate fin heat exchanger, effective heat transfer area on SHe side, effective heat transfer area on LHe side, and effective heat transfer area of parting plate are to be calculated first. Based on this, thermal resistances are estimated. By using basic steady state heat transfer equation in terms of temperature difference and thermal resistance, the height of the heat exchanger is estimated.

4.2.1 Effective Heat Transfer Area

If we take no. of flow passages on LHe side $= n$

Then, no. of flow passages on SHe side will be $= n + 1$

And, no. of parting plates $=2n$

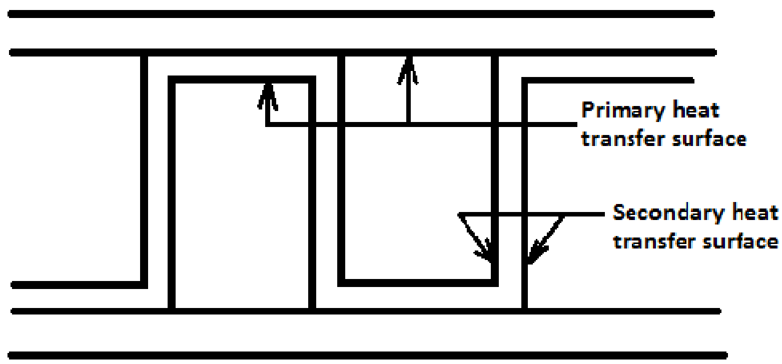

Figure 4.2: Cross-sectional View of Fin and Parting Plate

Primary heat transfer area of finned passage on SHe side of parting plate is given by[27],

$Au_1 = 1 - (n \times t_f)$

Secondary heat transfer area of finned passage on SHe side of parting plate is given by[27],

$Au_2 = n \times (H_f - t_f)$

NOTE: Values of both Au_1 and Au_2 are per unit area of parting plate.

(Similar equations can be used for LHe side also after replacing n by n-1)

The effective heat transfer area[27] of a parting plate is total area of parting plate normal to heat transfer path.

The effective heat transfer area on one side of parting plate per unit area of parting plate is given by,

$Au = Au_1 + (\eta \times \varphi \times Au_2)$

Where, fin efficiency $\eta = \frac{tanh(\beta/2)}{(\beta/2)}$

and

$\beta = H_f \times \left(\frac{2 \times h}{k_f \times t_f}\right)^{0.5}$

φ= un-perforated fraction = 1-(%perforation)/100 = 1 in our case (as selected fin have no perforations)

Let,

Au_S= Effective heat transfer area on SHe side per unit area of parting plate

Au_L= Effective heat transfer area on LHe side per unit area of parting plate

W_P = Width of parting plate = fin pitch * number of fins per passage

Total effective heat transfer area on SHe side = Au_{TS} =$Au_S \times W_X \times H_X$

$\frac{Au_{TS}}{H_X} = Au_S \times W_X = Ah_S$

Where, Ah_S = Effective heat transfer area on SHe side per unit height of heat exchanger

Similarly, we can write $Ah_L = Au_L \times W_X$ = Effective heat transfer area on LHe side per unit height of heat exchanger

Effective heat transfer area of parting plate per unit height of heat exchanger = $Ah_P = W_X$

4.2.2 Height of Heat Exchanger

Steady state heat transfer equation,

$$Q = \frac{\triangle T}{R_T} = \frac{\triangle T}{R_S + R_P + R_L}$$

$$\therefore Q = \frac{\triangle T}{\frac{1}{h_S \times Ah_S \times H_X} + \frac{t_P}{k_P \times Ah_P \times H_X} + \frac{1}{h_L \times Ah_L \times H_X}}$$

$$H_X = \frac{Q}{\triangle T} \times \frac{1}{h_S \times Ah_S} + \frac{t_P}{k_P \times Ah_P} + \frac{1}{h_L \times Ah_L}$$

The estimated values of thermal resistances and height of heat exchanger are shown in table below.

Table 4.5: Thermal Resistances and Height of Heat Exchanger

Control Volume No.	R-SHe Side (K/W)	R-LHe Side (K/W)	R-Parting Plate (K/W)	Total Resistance (K/W)	Area Safety Factor	Height of Heat Exchanger (m)
1	2.12E-06	3.96E-05	2.00E-06	4.37E-05	1.3	0.0503
2	2.18E-06	3.94E-05	2.04E-06	4.36E-05	1.3	0.0509
3	2.23E-06	3.92E-05	2.08E-06	4.35E-05	1.3	0.0525
4	2.28E-06	3.90E-05	2.13E-06	4.34E-05	1.3	0.0555
5	2.33E-06	3.88E-05	2.17E-06	4.33E-05	1.3	0.0600
6	2.38E-06	3.86E-05	2.22E-06	4.32E-05	1.3	0.0668
7	2.42E-06	3.83E-05	2.26E-06	4.30E-05	1.3	0.0771
8	2.46E-06	1.73E-06	2.32E-06	6.51E-06	1.3	0.0143
9	2.51E-06	2.37E-06	2.37E-06	7.25E-06	1.3	0.0210
10	2.55E-06	3.54E-06	2.43E-06	8.52E-06	1.3	0.0388
					Total	0.4874

The area safety factor of 1.3 is considered for the estimation of heat transfer area and height of heat exchanger due to the following reasons.

1. The 20% uncertainty of heat transfer correlations[23].

2. The 10% additional heat transfer area to take in to account effect of flow mal-distribution on process heat transfer[28].

4.3 PRESSURE DROP ANALYSIS

Based on the total heat transfer area requirement for the given heat transfer, the height of heat exchanger is estimated. The pressure drop on SHe as well as LHe side can be estimated by using the height of the heat exchanger. The estimated pressure drop will be within the limit provided as a design input. If it is more than the specified limit, we have to change the fin size to maintain the pressure drop within specified limit.

4.3.1 Pressure Drop on SHe Side

SHe enters the heat exchanger from the top and leaves at the bottom. Pressure drop in SHe side occures due to Pressure drop at entrance of fin passage, pressure drop in the core of heat exchanger, and pressure drop at exit of fin passage.

- **Pressure drop at entrance of fin passage**

Pressure drop at entrance of fin passage is due to two reasons.
(i) Pressure drop due to change in flow area
(ii) Pressure drop due to flow expansion

- Pressure drop at the entrance of core of heat exchanger due to sudden decrease in flow area is given by[28],

$$\triangle P_{i1} = \frac{Gm^2}{2{\times}\rho_{iS}}(1-\sigma^2)$$

Where, = Ratio of minimum free flow area to frontal area of core of heat exchanger

- Pressure drop at the entrance of core of heat exchanger due to expansion of flow is given by[28],

$$\triangle P_{i2} = \frac{Gm^2}{2{\times}\rho_{iS}}Kc$$

Where, Kc = Pressure loss coefficient at the entrance of heat exchanger core

- **Pressure drop in the core of heat exchanger**

Pressure drop in the core of plate fin heat exchanger is due to two reasons.

(i) Pressure drop due to change in density

(ii) Pressure drop due to flow friction

- Pressure drop due to change in density of flow is given by[28],

$$\triangle P_{c1} = \frac{Gm^2}{2\times\rho_{iS}}\times2\times\left(\frac{\rho_{iS}}{\rho_{oS}} - 1\right)$$

- Pressure drop due to flow friction is given by[28],

$$\triangle P_{c2} = \frac{Gm^2}{2\times\rho_{aS}}\times4f\times\frac{H_X}{D}$$

- **Pressure drop at exit of fin passage**

Pressure drop at exit of fin passage is due to two reasons.

(i) Pressure drop due to change in flow area

(ii) Pressure drop due to flow expansion

- Pressure drop at the exit of core of heat exchanger due to change in flow area is given by[28],

$$\triangle P_{o1} = \frac{Gm^2}{2\times\rho_{oS}}(1-\sigma^2)$$

- Pressure drop at the exit of core of heat exchanger due to free expansion of flow area is given by[28],

$$\triangle P_{o2} = \frac{Gm^2}{2\times\rho_{oS}}Ke$$

Where, Ke = Pressure loss coefficient at the exit of heat exchanger core

- **Pressure drop due to static head**

There will be a pressure difference due to static head of fluid column. Due to increase in elevation along the flow passage, there will be a pressure drop; and due to decrease in elevation along the flow passage, there will be a pressure gain. There is net decrease in elevation of SHe since SHe flows from top to bottom. So, the result will be pressure gain and it is given by[28],

$\triangle P_{sh} = -H \times \rho_{aS} \times g$

- **Total pressure drop**

The total pressure drop of heat exchanger is sum of pressure drop at entrance of fin passage, pressure drop in core of heat exchanger, pressure drop at exit of fin passage and static head gain.

$\triangle P_{total} = \triangle P_{i1} + \triangle P_{i2} + \triangle P_{c1} + \triangle P_{c2} + \triangle P_{o1} + \triangle P_{o2} + \triangle P_{sh}$

The estimated pressure drop values for SHe are summarised in table below.

Table 4.6: Pressure Drop on SHe Side along the Height of Heat Exchanger

Control Volume No.	$\triangle P_{i1}$ (Pa)	$\triangle P_{i2}$ (Pa)	$\triangle P_{c1}$ (Pa)	$\triangle P_{c2}$ (Pa)	$\triangle P_{o1}$ (Pa)	$\triangle P_{o2}$ (Pa)	$\triangle P_{sh}$ (Pa)	$\triangle P_{total}$ (mbar)
1	8.064	2.016	-0.194	9.943	-	-	-57.781	-0.380
2	-	-	-0.166	9.892	-	-	-59.643	-0.499
3	-	-	-0.145	10.083	-	-	-62.799	-0.529
4	-	-	-0.129	10.525	-	-	-67.471	-0.571
5	-	-	-0.115	11.270	-	-	-74.144	-0.630
6	-	-	-0.105	12.428	-	-	-83.701	-0.714
7	-	-	-0.096	14.223	-	-	-97.872	-0.837
8	-	-	-0.089	2.608	-	-	-18.303	-0.158
9	-	-	-0.083	3.824	-	-	-27.333	-0.236
10	-	-	-0.079	7.013	6.964	1.741	-50.989	-0.354
							Total	-4.906

Negative sign indicates pressure gain as SHe flows from top to bottom in the core of heat exchanger.

4.3.2 Pressure Drop on LHe Side

LHe is supplied from the bottom side of heat exchanger and the vapour generated due to heat transfer is removed from top of heat exchanger.LHe is supplied from bottom side to maintain the appropriate quantity of LHe inside the heat exchanger. The pressure drop on LHe side is static head loss along the height of the heat exchanger from bottom to top of LHe bath which is estimated as follows.

$\triangle P_{LH} = H \times \rho_{aL} \times g$

The estimated pressure drop values for LHe are summarised in table below.

Table 4.7: Pressure Drop on LHe Side along the Height of Heat Exchanger

Control Volume No.	$\triangle P_{LH}$ (mbar)
1	0.619
2	0.626
3	0.646
4	0.683
5	0.739
6	0.822
7	0.949
8	0.175
9	0.259
10	0.478
Total	5.994

4.4 Selection of Headers

The inlet of SHe, LHe to the PFHE as well as the outlet flow of SHe and evaporated helium gas (GHe) from PFHE is directed through headers.

There are two types of headers : (i) primary headers and (ii)secondary headers.

Primary are also called external headers. They are connected with inlet/outlet pipes of SHe, LHe or GHe on one side and secondary headers on other side.

The secondary are also called internal headers. They are connected with a primary header on one side and finned flow passages on other side.

4.4.1 Primary Headers

A typical primary header is a semi-circular pipe with Tee branch line for inlet or outlet pipe connection.

There are two types of primary headers in the plate fin heat exchanger.

(i)Inlet primary headers and (ii)Outlet primary headers.

An inlet primary header distributes fluid coming from inlet pipe to respective inlet secondary headers.

An outlet primary header collects fluid from each of the outlet secondary headers and directs it to outlet pipe.

Four types of primary headers are present in the plate fin heat exchanger.

(i) Inlet primary header for SHe (1 No.)

(ii) Inlet primary header for LHe (1 No.)

(iii) Outlet primary header for SHe (1 No.)

(iv) Outlet primary header for GHe (1 No.)

4.4.2 Secondary Headers

There are two types of secondary headers in the plate fin heat exchanger .

(i) Inlet secondary headers (ii) Outlet secondary headers

The inlet secondary header distributes fluid coming from inlet primary header to respective finned flow passages in the core of plate fin heat exchanger.

The outlet secondary header collects fluid from each of the finned flow passage and directs it to the outlet primary header.

Four types of secondary headers are there in the present plate fin heat exchanger.

(i) Inlet secondary headers for SHe (50 No.)

(ii) Inlet secondary headers for LHe (49 No.)

(iii) Outlet secondary headers for SHe (50 No.)

(iv) Outlet secondary headers for VHe (49 No.)

In case of bath cooling, evaporated helium gas (GHe) is directly vented through outlet primary header of GHe (1 No.). So, no outlet secondary headers exist for GHe. Whereas outlet secondary headers for VHe (49 No.) are present in case of thermosyphon cooling.

Based on fluid flow directions in inlet and outlet headers the geometrical configuration of secondary oblique headers[29] can be broadly classified as follows.

1. Parallel flow header configuration : In which, direction of fluid flow in outlet header is similar to the direction of fluid flow in inlet header.

2. Counter flow header configuration : In which, direction of fluid flow in outlet header is opposite to the direction of fluid flow in inlet header.

In present, counter flow headers are selected as the direction of fluid flow in outlet header is opposite to the direction of fluid flow in inlet header and pressure drop for secondary headers are taken same as the primary headers to reduce pressure drop complications.

4.4.3 Pressure Drop in Headers

The standard pipe sizes and thicknesses are used for design of headers. The list of pipe sizes with thickness as per ANSI[30] standard is provided in table below.

Table 4.8: Standard Pipe Sizes as per ANSI B36.19

Sr. No.	Pipe size (Inches)	Pipe Outer Dia. (mm)	Schedule 5S Pipe Thickness (mm)	Schedule 10S Pipe Thickness (mm)	Schedule 40S Pipe Thickness (mm)
1	1	33.4	1.65	2.77	3.38
2	1.25	42.2	1.65	2.77	3.56
3	1.5	48.3	1.65	2.77	3.68
4	2	60.3	1.65	2.77	3.91
5	2.5	73	2.11	3.05	5.16
6	3	88.9	2.11	3.05	5.49
7	3.5	101.6	2.11	3.05	5.74
8	4	114.3	2.11	3.05	6.02
9	5	141.3	2.77	3.4	6.55
10	6	168.3	2.77	3.4	7.11

The pressure drop at the entrance of SHe/LHe inlet primary header as well as pressure drop at exit of SHe/GHe primary headers is estimated as follows.

$$P = \frac{\rho f L V^2}{D_i}$$

Where,
P= pressure drop
ρ= fluid density
f= friction factor
L= equivalent hydraulic length of Tee for pressure drop estimation= $60D_i$
V= velocity of fluid
D_i== inner diameter of inlet/outlet pipe
Friction factor has been estimated using Colebrook's equation as follows

$$f = \left[\frac{1}{-2ln\left[\frac{R_h}{3.7D_i} + \frac{2.51}{Re\sqrt{f}}\right]}\right]^2$$

The value of friction factor for the first iteration may be assumed according to the Blasius equation as follows

$$f = \frac{0.079}{Re^{0.25}}$$

The estimated frictional pressure drop in SHe inlet primary header as a function of inlet header size is plotted in Figure 5.1.

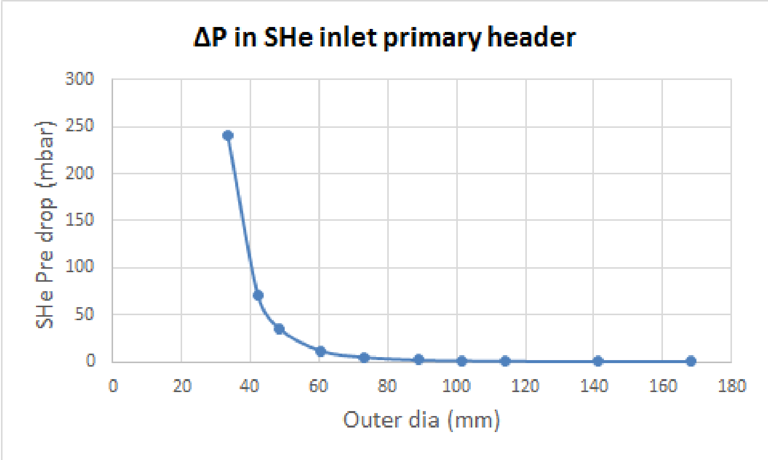

Figure 4.3: Pressure Drop in SHe Inlet Primary Header

The estimated frictional pressure drop in SHe outlet primary header as a function of inlet header size is plotted in Figure 5.2.

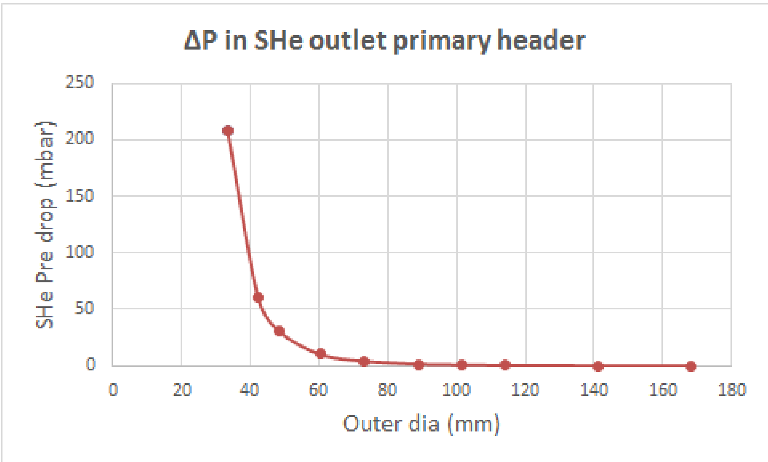

Figure 4.4: Pressure Drop in SHe Outlet Primary Header

The estimated frictional pressure drop in LHe inlet primary header as a function of inlet header size is plotted in Figure 5.3.

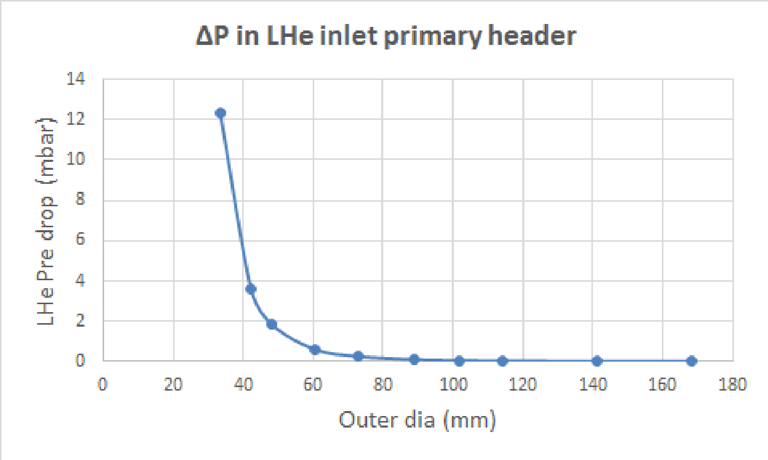

Figure 4.5: Pressure Drop in LHe Primary Header

The estimated frictional pressure drop in LHe inlet primary header as a function of inlet header size is plotted in Figure 5.4.

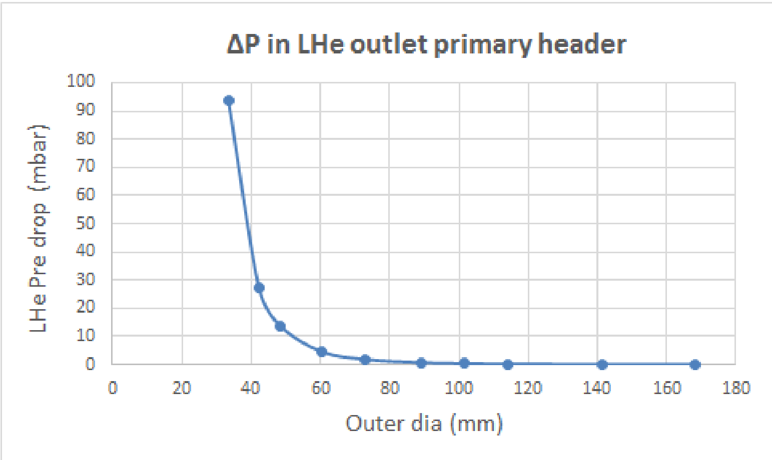

Figure 4.6: Pressure Drop in LHe Outlet Primary Header

Inlet header with outler diameter 73 mm and outlet header with outer diameter 60.3 are selected for SHe flow. The corresponding pressure drop are 4.996 mbar and 10.287 mbar.

Inlet header with outler diameter 48.3 mm and outlet header with outer diameter 73 are selected for LHe flow. The corresponding pressure drop are 1.817 mbar and 1.926 mbar.

Chapter 5

Thermosyphon Cooling Technique

Thermosyphon (or thermosiphon) is a circulating fluid system whose motion is caused by density differences which result from heat transfer. The natural convection is caused by gravity as the colder fluid having high density flowing downwards and the warmer, less dense fluid flowing upwards. Thus, thermosyphon provides cooling by connecting an object to be cooled with a reservoir. There are a variety of designs in thermosyphons. They can be categorized according to

1. The nature of boundaries : open system or closed system.
2. The regime of heat transfer : natural convection or mixed forced and natural convection.
3. The number of phases present : single phase or two phase.
4. The nature of the body force : gravitational or rotational.

The most common industrial thermosyphon applications includes gas turbine blade cooling, electrical machine rotor cooling, transformer cooling, nuclear reactor cooling , cryogenic cool down apparatus, and cooling for internal combustion engines. In cryogenic applications, a two-phase system is widely used in which liquid flows down to the part being cooled and vapor flows back up to the cold sink. Depending on the application, thermosyphons may consist of a single pipe or separate pipes for the cold and the warm fluids. In cryogenic thermosyphons, a variety of working fluids are used, including helium, nitrogen, argon or neon.

Thermosyphons have many advantages. i.e. no external mechanical pumping is required to provide fluid flow for heat transfer. This leads to simpler, more reliable systems. Since the thermal conductivity of most common materials at cryogenic temperatures is quite low, thermosyphons can be a better solution in terms of transferring heat more efficiently than solid conduction. As thermosyphon systems are gravity driven, they are best oriented in vertical or near vertical positions. For taking all the advantages of the thermosyphons, vertical oriented two phase thermosyphon is chosen for SHe to LHe heat exchange. As PFHE is placed outside of the LHe bath in thermosyphon configuration, LHe bath size can be reduced in this configuration. Only the distance between the top and bottom of the thermosyphon must be sufficiently enough to generate the required density gradient in order to set up the natural convection flow needed.

5.1 Thermosyphon Configuration

In present SHe to LHe thermosyphon, heat exchange is taking place in a two phase natural convection in an open system. LHe, which has the responsibility to cool down the SHe to a required temperature level, is the working fluid in this cryogenic thermosyphon cooling.

LHe receives the heat load from SHe in PFHE by absorbing latent heat and converted into vapour. Convective movement of the LHe starts, when there is a density gradient generated due to conversion of liquid helium into vapour helium, causing it to expand and become less dense and thus becomes more buoyant than the cooler liquid in the bottom of the loop. Thermosyphon configuration is shown in figure below.

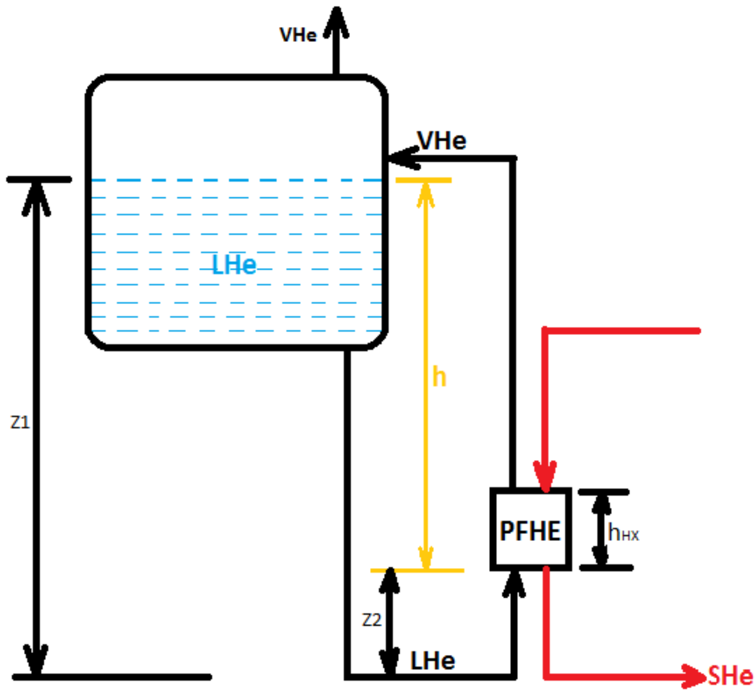

Figure 5.1: Thermosyphon Configuration

Thus due to density difference natural circulation of LHe is possible. There is no requirement of mechanical pump for the circulation of LHe.

5.2 Height of Thermosyphon

It is defined as the diatance between LHe level in the LHe tank and the entrance of the PFHE. Height of thermosyphon "h" shown in figure 5.1 is estimated as follows.

$$\int \frac{P}{\rho g} + \frac{V^2}{2g} + Z = Constant$$

Head loss due to minor losses are also added to determine the height of thermosyphon.

5.3 Minor Losses in Thermosyphon

All the minor losses are calculated separately for different sections as the density varies due to phase change of LHe. Minor losses are due to friction in the flow field within pipe and bends. The frictional pressure drop for flow through the length is given by Darcy -Weisbach equation :

$$\triangle P = f \left[\sum \frac{L}{D_i} \right] \frac{\rho V^2}{2}$$

Where,
$\sum \frac{L}{D_i}$ = equivalent length of straight pipe for fittings like bends, Tees, valves etc.
Friction factor has been estimated using Colebrook's equation as follows

$$f = \left[\frac{1}{-2ln \left[\frac{R_h}{3.7D_i} + \frac{2.51}{Re\sqrt{f}} \right]} \right]^2$$

The value of friction factor for the first iteration may be assumed according to the Blasius equation as follows

$$f = \frac{0.079}{Re^{0.25}}$$

Total minor loss for thermosyphon is coming out to be 1.256 mbar.

Chapter 6

Results and Discussion

SHe to LHe plate fin heat exchanger for fusion grade tokamak machine has been designed. The iterative programming in MS Office Excel has been developed to perform heat transfer analysis, pressure drop analysis and to calculate the height of plate fin heat exchanger. The final specifications of plate fin heat exchanger are provided in the following Tables.

Table 6.1: Specifications of Heat Exchanger

Specification	SHe side	LHe side
Standard fin	L-3-100-15	L-3-100-15
Fin type	OFFSET fins	OFFSET fins
Fin material	Al 3003	Al 3003
Fin pitch (S) (mm)	1.4	1.4
Thickness(t) (mm)	0.3	0.3
Height (h) (mm)	2.5	2.5
Fin length (Lf) (mm)	12.7	12.7
No. of rows	50	49
No. of flow passages in a row	180	180
Core pressure drop (mbar)	-4.906	5.994

Table 6.2: Specifications of heat exchanger

Specification	Value	Unit
Parting plate material	Al 3003	
Parting plate thickness	0.43	mm
Area safety factor	1.3	
Height of HX core	0.487	m
Breadth	0.320	m
Width	0.306	m

Total pressure drop for 6.156 kW heat load for bath heat exchanger and thermosyphon heat exchanger are given in following tables.

Table 6.3: Total Pressure Drop for Bath Heat Exchanger

Pressure Drop	LHe Side (mbar)	SHe Side (mbar)
Inlet Header	3.634	9.992
HX Core	5.994	-4.906
Outlet Header	1.926	20.572
Total	11.554	25.658

Table 6.4: Total Pressure Drop for Thermosyphon Heat Exchanger

Pressure Drop	LHe Side (mbar)	SHe Side (mbar)
Inlet Header	3.634	9.992
HX Core	5.994	-4.906
Outlet Header	3.852	20.572
Minor Loss	1.256	-
Total	14.736	25.658

There are fluctuations in the SHe heat load with time which is a major factor to determine the height of thermosyphon. The SHe heat load profile with different time steps (0-1800sec) is given in figure below.

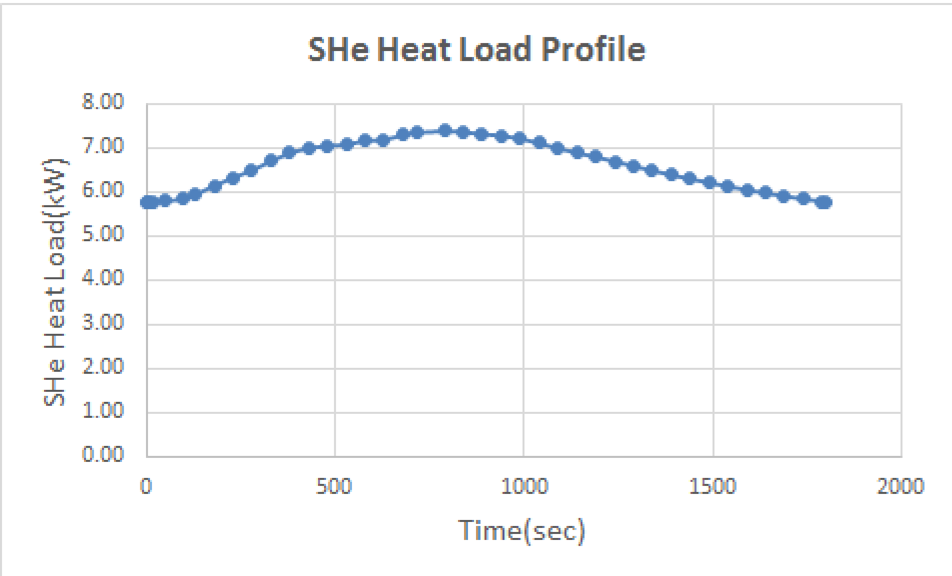

Figure 6.1: SHe Heat Load with Time Steps

LHe mass flow rate varies to meet the requirement of cooling in thermosyphon as SHe heat load varies. Estimated LHe mass flow rate as a function of different cooling loads is plotted below.

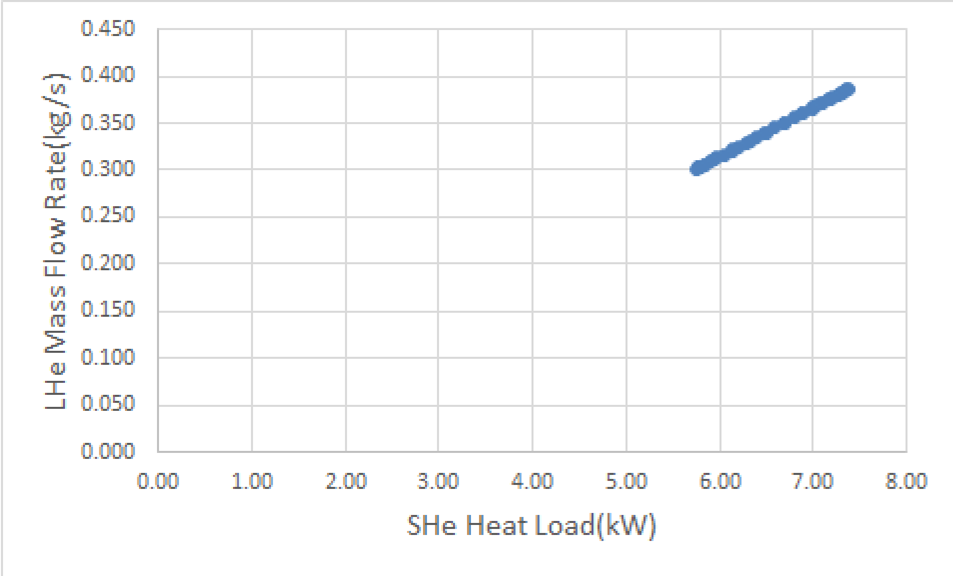

Figure 6.2: LHe Mass Flow Rate vs SHe Heat Load

There are variations in the height of thermosyphon with change in SHe heat load. Estimated height of thermosyphon as a function of different cooling loads is plotted below.

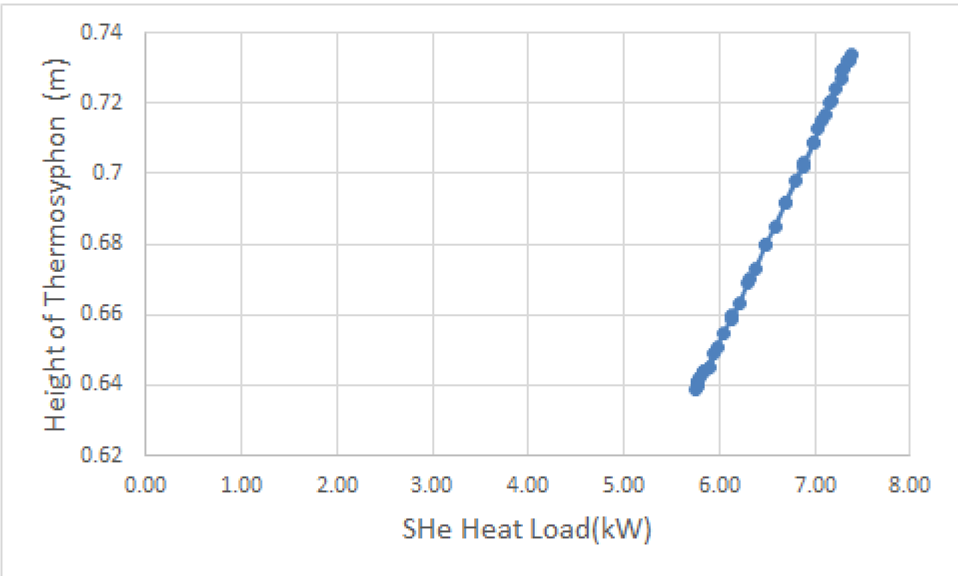

Figure 6.3: Height of Thermosyphon vs SHe Heat Load

As SHe heat load increases, the mass flow rate of LHe also increases as LHe has to absorb the heat load by latent heat only. This is achieved by increasing the height of thermosyphon as it will increase the liquid head in the circuit and leads to increase in mass flow rate.

Chapter 7

Conclusion and Future Scope

Conclusion

The following conclusions are drawn based on the present analytical study.

- The estimated height of the plate fin heat exchanger core is 0.487 m for heat transfer of 6.15kW, which is within a designed constraint (less than 1 m).
- Total pressure drop on SHe side is 25.6 mbar and LHe side is 11.554 mbar for bath cooling.
- Total pressure drop on SHe side is 25.6 mbar and LHe side is 14.736 mbar for thermosyphon cooling. The estimated pressure drop is within the allowable pressure drop that is 40 mbar on SHe side and 20 mbar on LHe side.
- PFHE should be placed below 0.734 m from LHe level in the LHe tank to absorb the fluctuations in SHe heat load.
- For the same heat transfer requirements, thermosyphon cooling can be a better option as size of the LHe bath can be reduced than it is used in bath cooling.

Future Scope

Multi-phase analysis can be made in finite element software like Ansys® and Fluent® .

Bibliography

[1] Iter-INDIA/(https://www.iter.org/mach)

[2] Shukla, V. et al, "Thermo-Mechanical Design Methodology for ITER Cryo-distribution Cold Boxes", Presented at 10th Asia Plasma & Fusion Association Conference (INDIA), 14th-18th December (2015).

[3] Horizon Technologies, n.d., "HePak", 3.4

[4] Kato, T. et al, 1992, "Heat Transfer Characteristics of a Plate Fin Type Super-critical/liquid Helium Heat Exchanger", Cryogenics, 32, pp. 260-263

[5] Jager, B. et al, 1986, "Heat Transfer in He I for Industrially Manufactured Aluminium Plate Heat Exchangers", Cryogenics, 26, pp. 222-225.

[6] Grigoriev, V. A. et al, 1977, "Characteristic Curve of Helium Pool Boiling", Cryogenics, pp. 155-156

[7] Dinaburg, L. B., 1971, "Experimental study of Boiling Heat Transfer in Liquid Helium", Cryogenics, pp. 238-239

[8] Jergel M. and Stevenson, R., 1972, "Heat Transfer to Boiling Helium from Aluminium Surfaces", Cryogenics, pp. 312-313

[9] Sciver, Steven W. Van, 2000, Helium Cryogenics, Springer

[10] E. Schmidt, "General Discussion on Heat Transfer," Institution of Mechanical Engineers, London, Section IV, p.361 (1951).

[11] M. J. Lighthill, "Theoretical Considerations on Free Convection in Tubes," Quarterly Journal of Mechanics ans Applied Mathematics, Vol. 6, Part 4, p.398 (1953).

[12] B. W. Martin and H. Cohen, "Heat Transfer by Free Convection in an Open Thermosyphon Tube," Brirtish Journal of Applied Physics, Vol. 5, p.91, (1954).

[13] H. Cohen and F. J. Bayley, "Heat Transfer Problems of Liquid Cooled Gas Turbine Blades," Proceedings Institution Mechanical Engineers, Vol. 169, p.1063-80 (1995).

[14] E. Schmidt, "Heat Transfer by Narural Convection," Presented at the International Heat Transfer Conference held at the University of Colorado (U.S.A.), August 28-September 1 (1961).

[15] L. Bewilogua and R. Knoner, "The Thermosyphon as a Nitrogen Cryostat for Operation in the Horizontal Reaction Channel," Cryogenics, p.46-47, September (1961).

[16] E. L. Long, "The Long Thermopile," Proc. Permafrost International Conference, p.487-491 (1963).

[17] Serio, L. et al, "The Conceptual Design of the ITER Cryodistribution System", Proceedings of the Twenty-Third International Cryogenic Engineering Conference, 2011.

[18] The Brazed Aluminium Plate Fin Heat Exchanger Manufacturers' Association, 2000, "ALPEMA Standard", 2

[19] Kuppan, T., 2000, Heat Exchanger Design Handbook, Marcel Dekker AG

[20] National Institute of Standards and Technology (NIST), 2010, from http://cryogenics.nist.gov/MPropsMAY/material%20properties.html

[21] Jainender Dewatwal, B.Tech thesis : Design of Compact Plate Fin Heat Exchanger, NIT Rourkela, 2009.

[22] "Lockhart formed fin products for heat transfer applications", Wakefield Thermal Solutions, Inc., USA

[23] Manglik, R. M. and Bergles, A. E., 1995, "Heat Transfer and Pressure Drop Correlations for Rectangular Offset Strip Fins Compact Heat Exchanger", Experimental Thermal and Fluid Science

[24] Goyal, M. et al, 2009, "Development of Plate and Fin Heat Exchangers", Indian Journal of Cryogenics, 34, pp. 1-4

[25] Corberan, J. M. et al, 2008, "Pressure Drop Characterisation of Compact Heat Exchanger Channels", 5th European Thermal Sciences Conference

[26] Deev, V. I. et al, 1977, "Nucleate and Film Pool Boiling Heat Transfer to Saturated Liquid Helium", Cryogenic, pp. 557-562

[27] The Brazed Aluminium Plate Fin Heat Exchanger Manufacturers' Association, 2000, ALPEMA Standard, 2

[28] Shah, R. K. and Sekulic, D. P., 2002, Fundamentals of Heat Exchanger Design, John Wiley and Sons

[29] London, A. L. et al, 1968, "Oblique Flow Headers for Heat Exchangers", Journal of Engineering for Power (Transactions of the ASME), pp. 271-286

[30] American Society of Mechanical Engineers, n.d., "ANSI B36.19"