Design Of 3-Stream(He-He-He) Plate-Fin Heat Exchanger For Helium Plant

Thesis Submitted in Fulfillment of the Requirements for the Award of the Degree of

Master of Technology (M. Tech.)

In

Cryogenic and Vacuum Technology

By

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<u>CERTIFICATE</u>

This is to certify that the dissertation, entitled

"Design of 3-Stream (He-He-He) plate-fin heat exchanger for helium plant"

is a bonafide work done by

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of

Institute for Plasma Research, Gandhinagar, Gujarat

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

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

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This is to certify that the thesis entitled "**Design of 3 – Stream (He-He-He) Plate-fin heat exchanger for Helium",** submitted to the National Institute of Technology, Rourkrla, by **Omshree Mahapatra**, Roll No. **212ME5408** for the award of the Degree of **Master of Technology** in **Mechanical Engineering** with specialization **"Cryogenic and Vacuum Technology"**, is a record of bonafide research work carried out by her under my supervision and guidance. The results presented in this thesis have not been, to the best of my knowledge, submitted to any other University or Institute for the award of any degree or diploma.

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

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ABSTRACT

Plasma Research belongs to a new generation of tokamaks with the major objective being steady state operation of advanced configuration plasma. Liquid helium plant is required for superconducting magnets. The effectiveness of the heat exchangers which is used in the helium plant should be close to 100%. For some cases, heat exchangers with effectiveness less than 90% can be a reason for failure of helium plant to produce liquid helium. To achieve such high effectiveness, it is necessary to use plate fin heat exchangers, which provides very high heat transfer surface area for heat exchange per unit volume (ratio greater than 700 m^2/m^3) of heat exchanger. Such heat exchangers also have benefit of low pressure drop of fluid flowing through it. The helium refrigerator and liquefier (HRL), planned for development at Institute for plasma research (IPR), will have eight main heat exchangers, 3 turbo expanders with Joule- Thomson expansion valves for achieving liquid helium production. The fourth exchanger, whose nominal operating temperature range is ~30 K to ~10 K, is the 3-stream type (He/He/He) plate fin heat exchanger. This 3-stream (He/He/He) heat exchanger will be installed inside a vacuum chamber having vacuum of about 10^{-5} mbar. High effectiveness, compact volume and low pressure drop are main optimizing parameters for the design of this heat exchanger. This design work is a step towards the development of the indigenous liquid helium plant.

Keywords: *Plate-fin ; 3stream;Heat exchanger; Helium*

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NOMENCLATURE

L	Length	m
W	Width	m
H	Height	m
T	Fin thickness	mm
h_{h1}	Hot 1 stream fin height	mm
h_{h2}	Hot 2 stream fin height	mm
h_c	Cold stream fin height	mm
N	No of layers	-
N	Fin density	Fins/m
Ax	Free flow area	m^2
Weff	Effective width	m
B	Fin height + fin thickness	mm
m_{h1}	Mass flow rate of hot 1 stream	kg/s
m_{h2}	Mass flow rate of hot 2 stream	kg/s
m_c	Mass flow rate of cold stream	kg/s
G	Mass velocity	kg/sm^2
J	Colburn factor	-
Re	Reynold number	-
C_P	Specific of heat transfer	J/KgK
H	Heat transfer coefficient	W/m^2K
ΔP_{fr}	Frictional pressure drop	bar
ΔPgap	Gap pressure drop	bar
Tb	Edge bar thickness	mm
L	Flow length	mm
μ	Viscosity	Pa-s
LMTD	Log Mean Temperature Difference	K
ε	Effectiveness	-
Dh	Hydraulic diameter	mm
UA	Overall heat transfer coefficient	W/K
η_f	Fin efficiency	-
V	Volume	M^3
Q	Heat transfer	W
K	Thermal conductivity	W/mK
F	Friction factor	-
B	Banking factor	-
P	Density	Kg/m^3

CHAPTER 1

1. INTRODUCTION

1.1 Institute for Plasma Research (IPR):

It is an autonomous research institute under Department of Atomic Energy, India. It belongs to a new generation of tokamak with the major objective being steady state operation of advanced configuration plasma. A proposal to the Government of India to initiate studies on magnetically confined high temperature plasmas was accepted in 1982 and resulted in establishment of the Plasma Physics Programme (PPP) supported by the Department of Science and Technology. Design and engineering of India's first tokamak ADITYA started at the same time. In 1984 the activities moved into an independent campus at Bhat village in the outskirts of Ahmedabad city in 1984. The PPP evolved into the autonomous Institute for Plasma Research under the Department of Science and Technology in 1986. With the commissioning of ADITYA in 1989, full-fledged tokamak experiments started. With the decision to build the second generation superconducting steady state tokamak SST-1 capable of 1000 second operation in 1995, the institute grew rapidly and came under the administration of the Department of Atomic Energy.

SST-1: SST-1 (Steady State Superconducting Tokamak) is a plasma confinement experimental device in the Institute for Plasma Research (IPR). It belongs to a new generation of tokamaks with the major objective being steady state operation of an advanced configuration ('D' Shaped) plasma. It has been designed as a medium-sized tokamak with superconducting magnets. The SST-1 project will increase India's stronghold in a selected group of countries who are capable of conceptualizing and making a fully functional fusion based reactor device. The SST-1 System is housed in Institute for Plasma Research, Gandhinagar.

SST-2: The next stage of SST mission, the SST-2 fusion reactor, dubbed as 'DEMO' among Indian scientific circles has already been conceived. Many new features like D-T plasma, Test Blanket Module, Biological shielding and an improved diverter will be incorporated in SST-2. SST-2 will also be built in Gujarat. According to the present plan, India will be producing electricity from nuclear fusion.

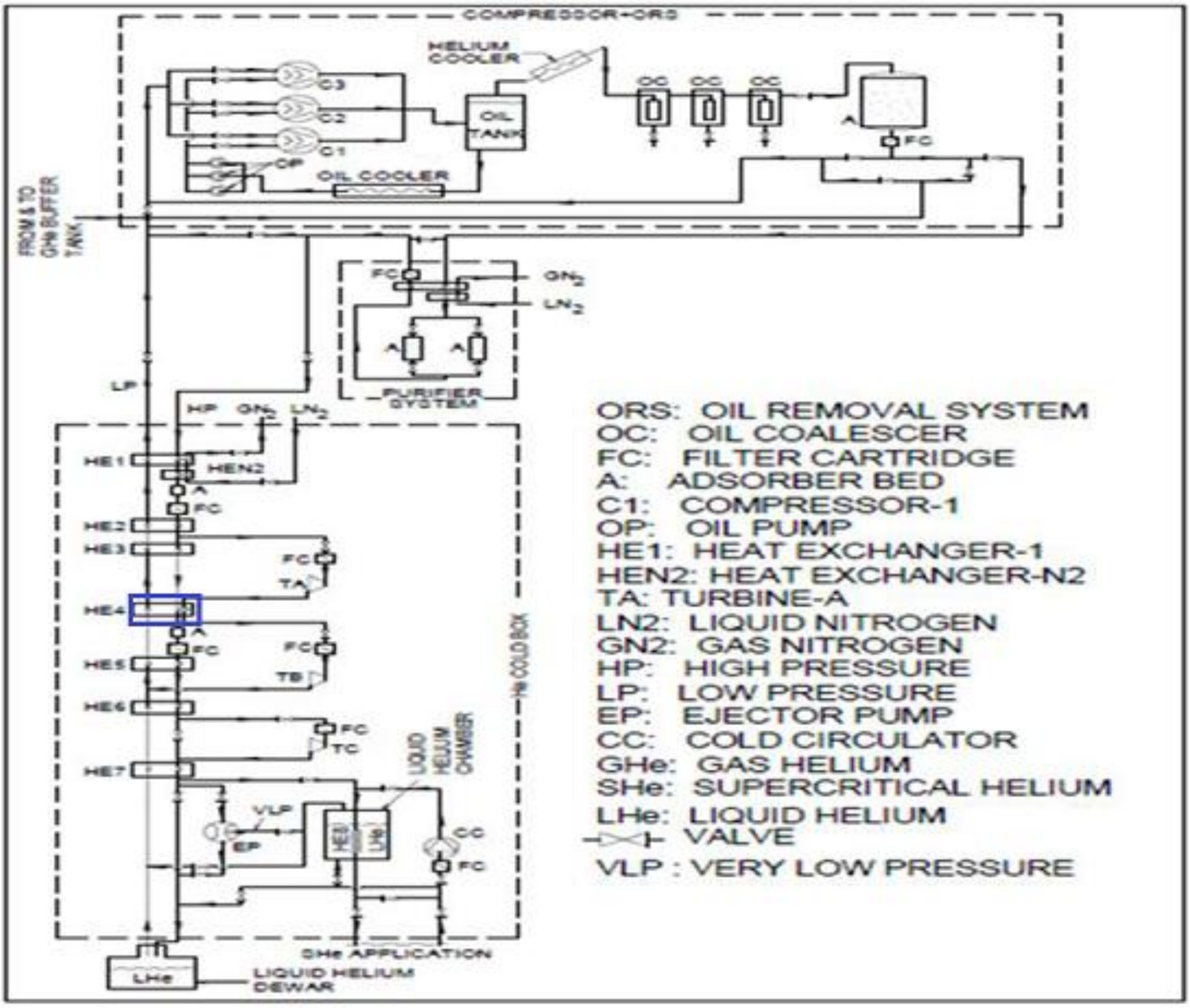


Fig.1.1 Typical Schematic diagram of Helium Refrigeration and Liquefaction plant

I have done the design of 4th heat exchanger which is maintained between temperatures 30K to 10K as shown in the figure 1.1.

1.2 Application of Plate Fin Heat Exchangers (PFHE) in Helium Refrigeration and Liquefaction Plant

Liquid helium is generally used for superconducting magnets. The heat exchangers, which are working at cryogenic temperature, are considered as the prime components of helium liquefier/ refrigerator plant. The heat exchangers having effectiveness less than 90% may cause the failure of the plant, which lead to use of plate fin heat exchanger having large surface area per unit volume and low pressure drop of fluid.

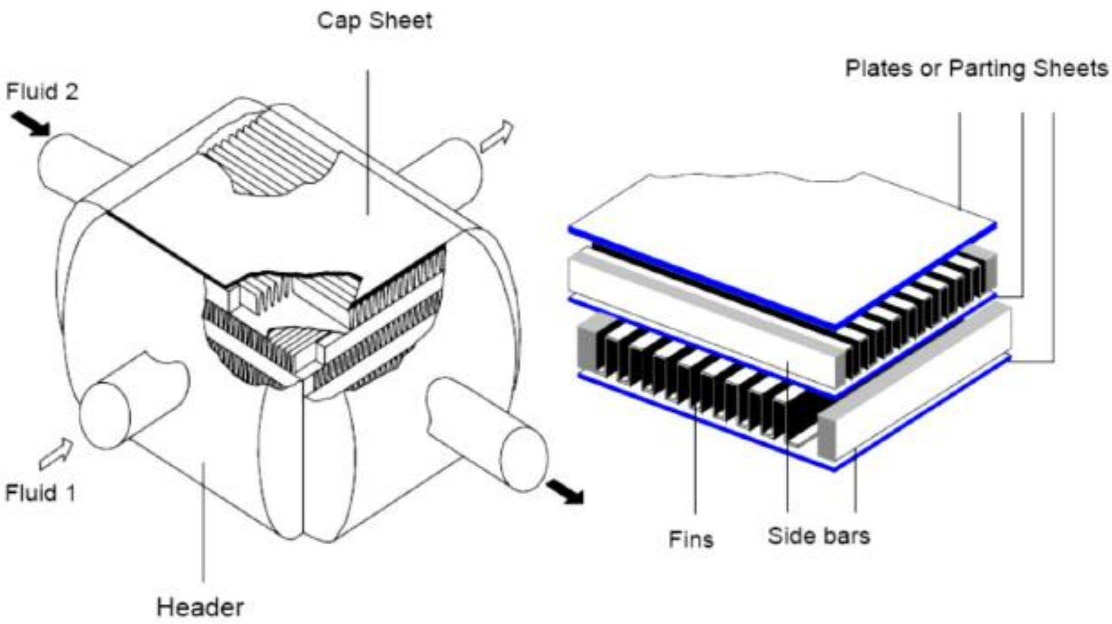
1.3 Objective of the Present Study:

Prime objectives of the present thesis are:

- Thermo-Hydraulic design methodology for 3-stream multi fluid (He/He/He) Plate fin Heat Exchanger for helium plant.
- Computer program in Microsoft excel for design and analytical analysis of 3-stream Plate Fin Heat Exchanger for helium plant.
- Optimization of 3-stream plate fin heat exchanger thermal design with help of graphical presentation method and validation with aspen tech software.

1.4 Plate-fin Heat Exchanger:

Plate fin exchangers have fins of large surface area due to the extended metal surface, placed between the two fluids. The superior construction of plate fin heat exchangers make it unique among compact heat exchangers, which is also responsible for high effectiveness, compactness, lower weight and reasonable cost. Plate fin heat exchanger (PFHE) consists of a stack of parting sheets placed alternatively and corrugated fins brazed collectively as a single block. Heat transfer takes place from the streams due to the flow through the passage made by the fins between the parting sheets. Separating plates and appendages are considered as the primary and secondary heat transfer surfaces respectively and they are bonded together. Fins are responsible for both heat transfer and structural supports against internal pressure difference. The side bars are responsible for prevention of the fluid from dropping over, mixing and leaking. The fins and side bars are joined with the parting sheets for good thermal link and mechanical stability. Figure 1.2 shows the assembly and information of two layers of a plate fin heat exchanger. Such layers are arranged in a monolithic block and form a heat exchanger.



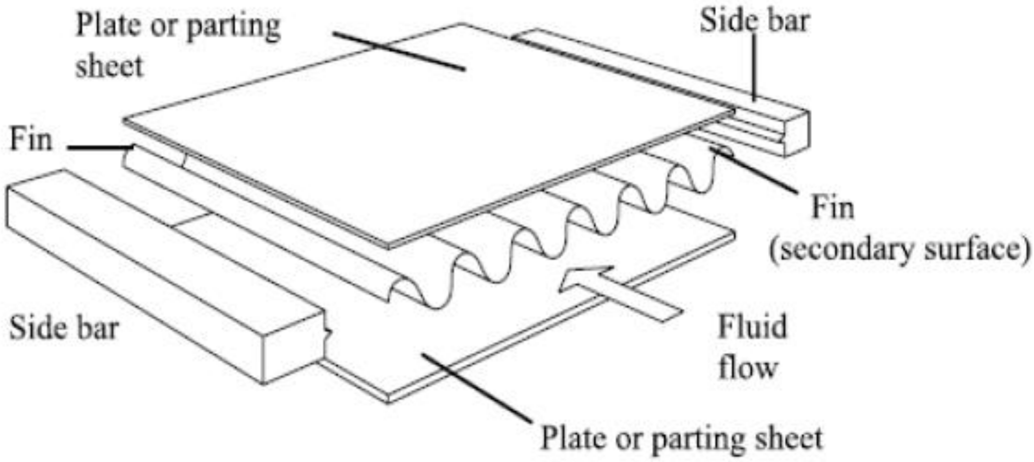


Fig.1.2 Plate-fin heat exchanger assembly and details

Plate fin heat exchangers are preferred over the other types of heat exchanger due to the following reasons.

- Compactness
- Effectiveness
- Temperature control
- Flexibility

Limitations

- The constricted and narrow passage leads to relatively high pressure
- The operating range of temperature and pressure become limited due to rectangular geometry
- It is very difficult to clean the passages, repair the failures and prevent the leakage

1.5 Different Flow Arrangements:

There are several flow arrangements for a plate fin heat exchanger. They are as follows.

1.5.1 Cross flow: In cross flow heat exchangers, the fluids flow normal to each other as shown in figure 1.3. Thermodynamically the effectiveness for cross flow heat exchangers falls in between that for the counter flow and parallel flow arrangements. The largest structural temperature difference exists at the corner of the entering hot and cold fluids. Only two streams are handled in a cross flow type of a heat exchanger which eliminates the need for distributors. Typical applications include automobile radiators and some aircraft heat exchangers.

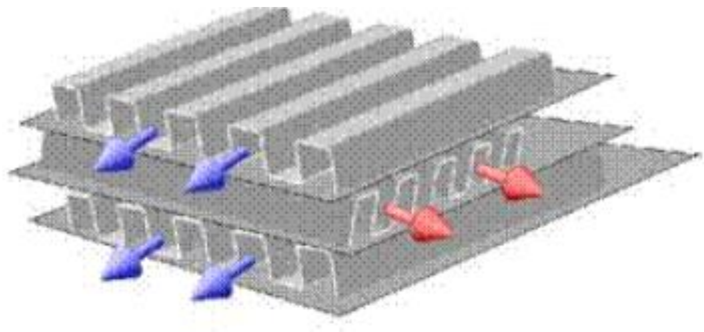


Fig.1.3 Cross flow

1.5.2 Counter flow: In this type of heat exchanger the two streams of fluids flow in a paralelly in opposite direction to each other (Fig1.4). The counter flow heat exchanger has been proved to provide the thermally valuable arrangement in order to recover the temperature from process streams compare to any other arrangement for a given overall thermal conductance (UA), fluid inlet temperatures and fluid flow rates. Application of counter flow arrangement is widely used in Cryogenic refrigeration plants use this geometry. Complex geometry of the headers needs proper design.

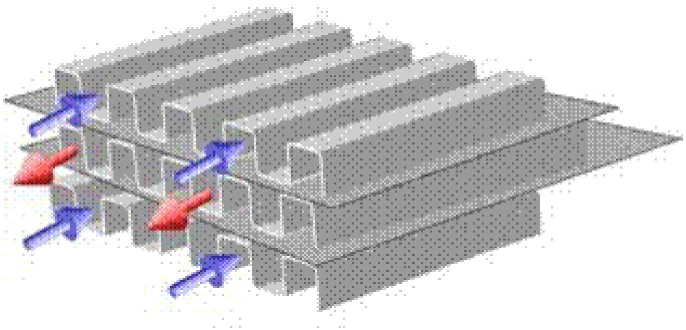


Fig.1.4 Counter flow

1.5.3 Cross-counter flow: This type of arrangement involves the combination of counter flow and cross flow in order to enhance the thermal effectiveness as well as rate of heat transfer simultaneously. In this arrangement, one of the streams flows in a straight path, whereas the second stream follows a zigzag path normal to that of the first stream. While moving along the zigzag path, the second fluid stream covers the length of the heat exchanger in a direction opposite to that of the direct stream. Thus the flow pattern can be assumed to be globally counter flow while remaining locally cross flow. Cross-counter flow PFHEs are used in applications similar to those of simple cross flow exchangers, but they allow more flexibility in design and fabrication. They are particularly suited for the applications where the two streams have considerably different volume flow rates, or permit significantly different pressure drops.

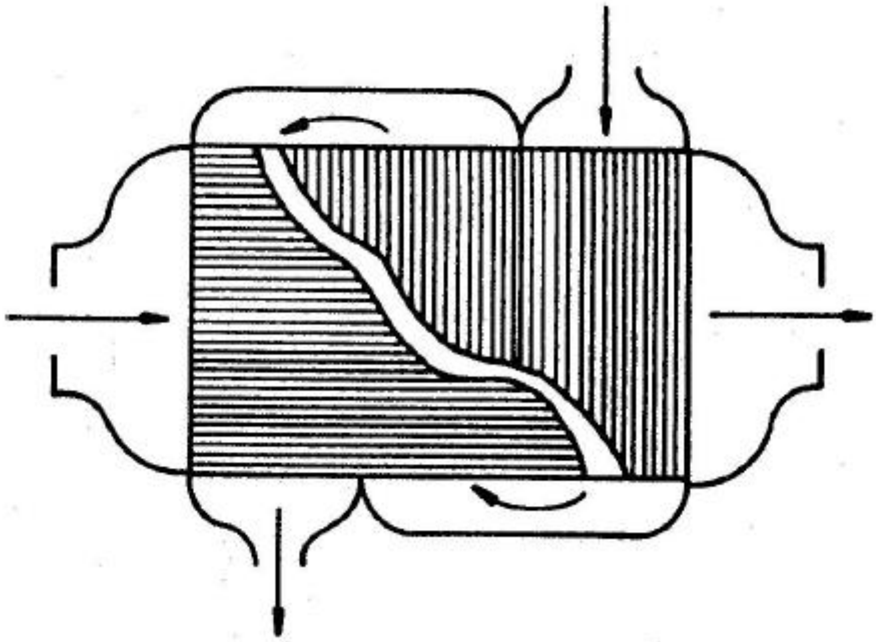


Figure.1.5 Cross-counter flow

The fluid with the larger volume flow rate or that with the smaller value of allowable pressure drop is made to flow through the straight channel, while the other stream follows the zigzag path.

1.6 Different Types of fins: Due to the low heat transfer coefficients in gas flows, extended surfaces are commonly employed in plate fin heat exchangers. By using specially configured extended surfaces, heat transfer coefficients can also be enhanced. While such special surface geometries provide much higher heat transfer coefficients than plain extended surfaces, but at the same time, the pressure drop penalties are also high, though they may not be severe enough to negate the thermal benefits. A variety of extended surfaces like the plain trapezoidal, plain rectangular can perform such function, and we have included the offset strip fin geometry in our present work. In order to improve the gas side coefficients, surface features are needed to be provided on the gas side coefficients.

1.6.1 Plain fins: These are straight fins that are continuous in the fluid flow direction Fig 1.6. Commonly plain fin are obtained in the form of triangular and rectangular cross section, but we can obtain any desired shape according to the ease of manufacturing aspects. Triangular fins are easy to manufacture other than any other cross section. But plain fins have some of limitation like structurally weak, having lower rate of heat transfer during laminar flow. Plain fins are widely used for critical pressure drop. Plain fins require a smaller flow frontal area compare to that with interrupted fins for same thermal and flow property but the larger passage length increases the the overall volume.

Fig.1.6 Plain fin

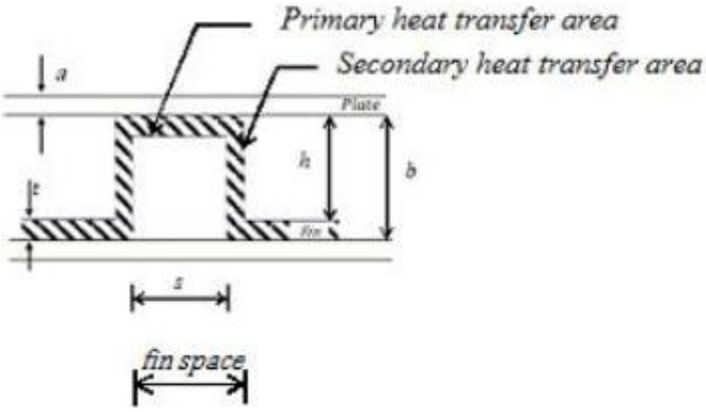


Fig1.7 Plain rectangular fin geometry

Hydraulic diameter (Dh) is given as:

$$D_h = \frac{2*(b-t)*(1-nt)}{1-nt+(n*(b-t))} \qquad (1.1)$$

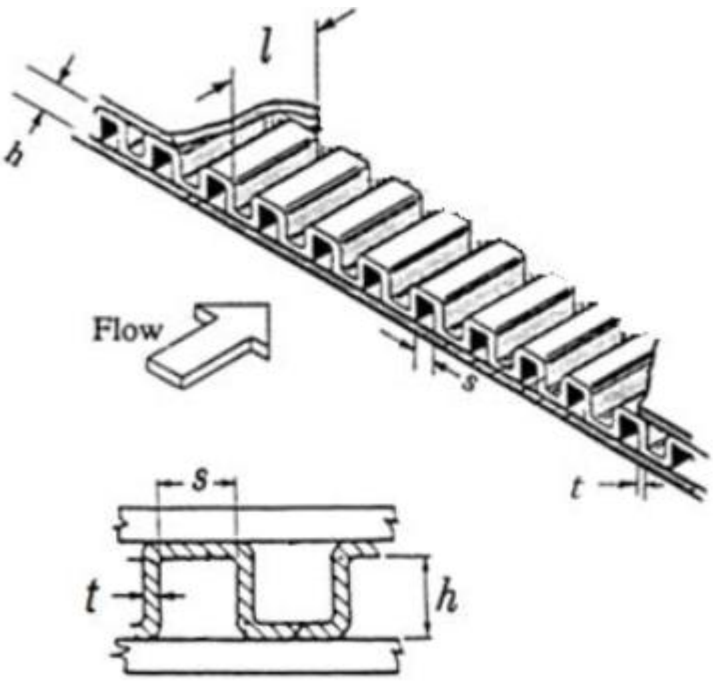


Fig.1.8 Geometrical description of a typical plain rectangular fin

1.6.2 Wavy fins: Wavy fins are uninterrupted fin surfaces with cross-sectional shapes similar to those of plain fins, but with cyclic lateral shifts perpendicular to the flow direction as shown in Figure.1.9 . The resulting wave form provides effective interruptions and induces a complex flow field. Heat transfer is enhanced due to creation of vortices. These counter-rotating vortices form while the fluid passes over the concave wave surfaces, and produce a corkscrew-like flow pattern. The heat transfer and pressure drop characteristics of a wavy fin surface lie between those of plain and offset strip fins. The friction factor continues to fall with increasing Reynolds number. Unlike offset strip fins, the thickness of wavy fins is not limited at high fin densities. Therefore, wavy fins are often used for streams at high pressure, particularly those applications which can tolerate somewhat poor heat transfer coefficient.

Fig.1.9 Wavy fins

1.6.3 Offset Strip (Serrated) Fins: Generally, the offset strip fin is used in the manufacturing of plate fin heat exchanger. As the presence of surface interruption, it enhances the heat transfer by continuously interrupting the thermal boundary layer growth due to which thin boundary layer are produce which is responsible for low resistance for heat conduction and leads to higher rate of heat transfer. Its heat transfer performance is much more than the plain fins. But pressure drop is more for this type arrangement.

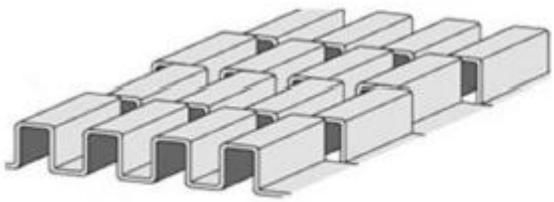


Fig.1.10 Offset Strip (Serrated) Fins

1.6.4 Louver Fins: The louvered fin geometry as shown in Figure 1.11 , bears a similarity to the offset strip fin. Instead of shifting the slit strips laterally, small segments of the fin are slit and rotated 20 to 45 degrees relative to the flow direction. The base surface of the louvered fin geometry can be of triangular or rectangular shape, and louvers can be cut in many different forms. The multi louvered fin has the highest heat transfer enhancement relative to pressure drop in comparison with most other fin types. An important aspect of louvered fin performance is the degree to which the flow follows the louver. At low Reynolds number the flow is nearly parallel to the axial direction (duct flow), whereas at high Reynolds number the flow is in the direction of the louvers (boundary layer flow).

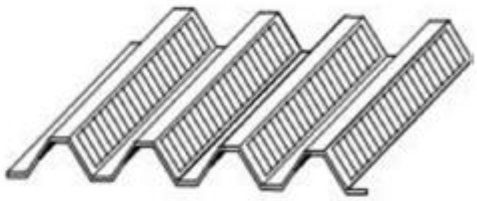


Fig.1.11 Louver fins

1.6.5 Perforated Fins: Perforated fins as shown in Figure 1.12, are made by punching a pattern of spaced holes in the fin material before it is folded to form the flow channels. The channels may be triangular or rectangular in shape with either round or rectangular perforations. While this geometry, with boundary layer interruptions, is a definite improvement over plain fins, its performance is generally poorer than that of a good offset strip fin. Furthermore, the perforated fin represents a wasteful way of making an enhanced surface, since the material removed in creating the perforations is thrown out as scrap.

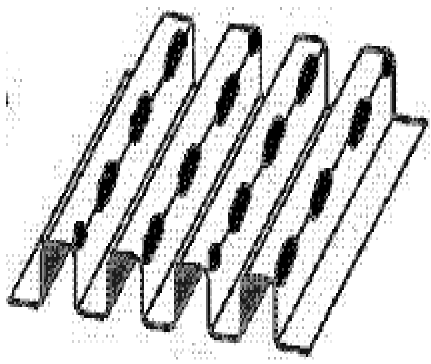


Fig.1.12 Perforated Fins

1.7 Applications of plate fin heat exchanger:

Air separation

- Helium and hydrogen liquefiers
- Petro-chemical processes
- Aero space
- Oil and gas processing
- Automobile radiators
- Air conditioners

1.8 Effects of physical parameters on performance of PFHE

1.8.1 Flow mal-distribution: In heat exchanger's design, generally the fluid is uniformly distributed throughout the heat exchanger cores. In practice, however, it is impossible to distribute the fluid uniformly because of improper inlet configuration, imperfect design and complex heat transfer process. The gross flow mal-distribution and passage to passage flow non-uniformity exist in plate fin heat exchangers.

1.8.2 Longitudinal thermal conduction: The longitudinal heat conduction occurs, when both the fluids flowing through the heat exchanger have same heat capacity rate. The ineffectiveness, in terms of a longitudinal heat conduction parameter is given as

$$\lambda = \frac{K_w L_w}{L C_{min}} = \frac{\delta\varepsilon}{\varepsilon} \quad (1.2)$$

1.8.3 Heat in leakage: The performance of most heat exchangers may be seriously affected by heat exchange with the surroundings. This is particularly true for cryogenic heat exchangers because of the large temperature differences between the ambient and the operating temperatures. This problem can be reduced by using highly effective insulation.

But factors like cost, weight and volume of insulation, difficulties in fabrication and ageing of insulation limit the extent to which heat transfer may be reduced.

1.8.4 Heat transfer and flow friction characteristics: The heat transfer and flow friction characteristics of a heat exchanger surfaces are commonly expressed in non-dimensional form and are simply referred to as the basic characteristic or basic data of the surface. Various correlations are available in literatures which express the Colburn factor, j and friction factor, f as functions of Reynolds number and other geometrical properties**.** The Colburn and friction factors are defined by the relations:

$$h = jC_pGPr^{-\frac{2}{3}} \tag{1.3}$$

Where, h =heat transfer coefficient

Fanning friction factor (*f*) is given by,

$$f = \frac{T}{\frac{1}{2\rho\vartheta^2}} \tag{1.4}$$

Pressure drop,

$$\Delta P = \frac{4fLG^2}{2\rho D_h} \tag{1.5}$$

1.8.5 Selection of Material : As shown in below Figure.1.13, among several types of materials, generally AL is used due to its following characteristics. Such as**,**

- Low density
- High thermal conductivity
- High strength and ductility at low temperature up to 4.3K.
- Low weight

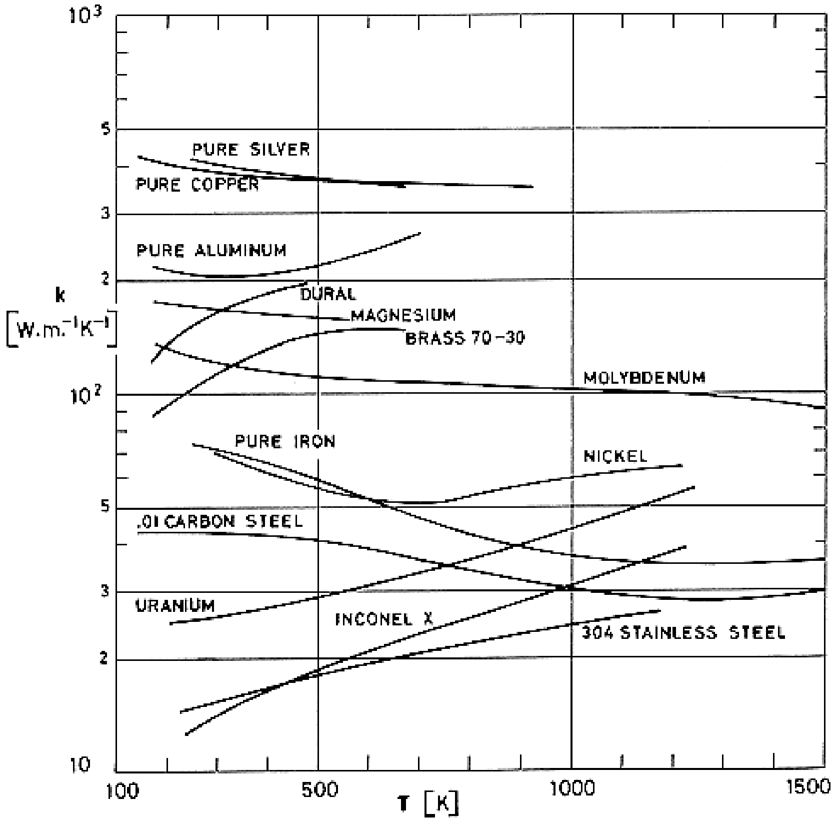


Fig.1.13 Thermal conductivity VS temperature graph

CHAPTER 2

LITERATURE REVIEW

Picon-Nunez et al. (1999) developed a relationship between heat transfer coefficients, pressure drop and exchanger volume by taking in to account the full pressure drop utilization of the plate-fin heatexchanger. More over they represented the surface selection depending upon the volume performance index and selecting the smallest volume of the exchanger within the block dimensions. **Bhutta et al. (2012)** investigated the performance and behaviour of the heat exchangers by taking in to account the Computational Fluid Dynamics (CFD) approach and found pressure drop fouling, thermal analysis in design, fluid flow mal distribution and optimization phase. **Roetzel et al.(2002)** found the networks of the multi-stream (Concurrent and counter current) heat exchanger by analytical process. These networks are solved for streams having constant physical properties. The solution is obtained by matching 3 matrices and is applied to multi-stream heat exchangers like plate-fin heat exchangers. **Picon-Nunez et al. (2002)** employed the design of (Plate and fin type counter current) multi-stream heat exchanger. In this case the Temperature vs. Enthalpy diagrams are used to show the division of the multi stream heat exchanger in to block sections. There after by using the enthalpy intervals, the entry and exit point of the streams are determined for maximum allowable pressure. **Shah et al.(2009)** implemented a comparative study between the offset strip fins and wavy fins by taking the dimension less parameters (fanning friction factor f and colburn factor j vs. Reynolds number Re) which are generated by CFD analysis for the design of compact heat exchanger. **Hu and Herold (1994)** considered pressure drop and heat transfer to develop a comparison liquid cooled and air cooled offset fin compact heat exchanger by using ranges of Prandtl number. It shows that in case of offset fin the effect of Prandtl number is more on Nusselt number for water cooled heat exchanger than air cooled.

Kang and Kim (1999) carried out a comparative study between plane fin, strip fin and their combinations for calculating the pressure drop and heat transfer. Here the existing co-relation and experimental data are compared with the test results. From this study it is known that the hybrid fin is more effective than the other to enhance the heat transfer. **Wang and Sunden (2010)** propose the design and optimization of multi-stream plate-fin heat exchanger by applying pinch technology to its several block sections. Then it is applied practically in industry to find the heat transfer and pressure drop. **Kim et al.(2000)** found a comparative study between the porous fins and the conventional louvered fins for determining the pressure drop and heat transfer in case of plate-fin heat exchanger. The results shows that the thermal performance for both the fins are similar, but in case of louvered the pressure drop performance is little better. **B.S.V Prasad (1995)** developed the differential equations regarding heat transfer when the fin bases are maintained at two different temperatures which are 1) T_A > To >T_B and 2) T_A> T_B>To. The possible mechanisms are based on fin efficiency. The result shows that for all mechanisms the heat transfer equations for fin bases are identical. **Sahin et al.(2005)** found the optimized values of fin parameters such as (fin width, fin height, angle of attack, span wise distance between fins), flow velocity and pressure drop by Taguchi method with considering Nusselt number and friction factor as performance parameters. **Ranganayakulu et al. (1996)** developed the effect of longitudinal heat conduction at different design and operating conditions of heat exchangers by using finite element method. Theresult shows that the thermal performance deterioration of counter flow plate-fin heat exchanger becomes significant when the fluid capacity rate ratio is equal to 1 and at its large value. **Gupta and Atrey (2000)** developed a comparative study between the numerical model that is previously developed and the experimental results by taking in to account the heat in leak and axial conduction parameters for 300K to 80K and 80K to 20K temperature ranges. **Wang et al. (1996)** carried out different tests and comparison of heat exchangers by taking different geometrical parameters. Finally they got the result as follows:

1) Fin spacing is independent of heat transfer coefficient
2) Fin thickness is independent of heat transfer and friction characteristics.
3) The number of tube rows has very little effect on friction factor.

Joshi and Webb (1987) discovered the flow structure at transition flow from laminar to turbulent for predicting the friction factor and heat transfer coefficient for offset strip fin heat exchanger. They used numerical solution method for calculating Nusselt (Nu) Fanning friction factor (f) in laminar regime and semi empirical approach in case of turbulent regime. **Wen and Li (2004)** investigated one header configuration by taking in to account the flow uniformity. A baffle having small holes of three different diameters is installed. Then the ratios of maximum flow velocity to minimum flow velocity are found for different Reynolds number. The results show that the performance of the heat exchanger is improved with a better header configuration. **Jiao and Beak (2006)** implemented optimum distributor configuration by considering the flow mal distribution parameter of the plate-fin heat exchanger. At different Reynolds number the ratio of maximum velocity to minimum velocity in the channels are found out to get the improved design of the heat exchanger. **Manglik and Bergles (1995)** developed the design of rectangular offset strip heat exchanger by using fanning friction factor (f) and colburn factor (j) datas. The asymptotic behaviour of f and j are found as a single continuous form of equation in laminar, turbulent and transition floe regimes.**Lee and Bhowmik (2009)** found a model for studying the pressure drop and heat transfer characteristics of offset strip fin heat exchanger. By using water as medium the variation of j and f factors are observed in relation with Reynold's number Re and more over this factors are used for analysing the heat transfer characteristics and fluid flow in case of laminar, turbulent and transition regions for offset strip fins. **Ozerdem et al. (2005)**

investigated the effects on heat transfer and pressure drop of a plate-fin and tube heat exchanger by changing the fin geometry in Computation Fluid Dynamics (CFD) approach. Different fins are used for the comparison by dividing them in to segments. The results obtained are as follows:

1) The pressure drop depends considerably on the distance between the fins.
2) The heat transfer is affected by placing the fin tube in downstream region.

CHAPTER 3

3. Plate-Fin Heat Exchangers: Design and Optimization

Design of plate fin heat exchangers involves the optimization of many interrelated parameters such as fin density, fin thickness, fin height, no. of layers, pressure drop, length and volume. Due to the presence of large number of influencing parameters, design process of heat exchangers is a challenging task. Hence, the design of heat exchanger was done by carrying out the desired calculations in excel sheet and then plotting the corresponding graphs. Finally optimization of graphs was done to find the optimal parameters which are to be considered for the final design of the heat exchanger. The working temperature range of the heat exchanger (3 stream: He-He-He) is 30 K to 10 K. The mass flow rates corresponding to low pressure, medium pressure and high pressure streams are 125 g/s, 74.72 g/s and 57.79 g/s respectively.

3.1 Design Methodology: The values of parameters which are to be used in further calculation are shown in Table 3.1 as provided by IPR. The design of plate fin heat exchanger was carried out in the following steps.

I: Selection of flow arrangement to be used in the heat exchanger:

Table3.1. Design data provided by IPR:

Properties	**Temperature (K)**	**Pressure (bar)**	**Mass flow rate (kg/s)**
$\boldsymbol{T_{h1-i}}$	27.32	12.99	0.05779
$\boldsymbol{T_{h1-o}}$	15.62	12.99	

T_{h2-i}	27.32	5.3	0.07472
T_{h2-o}	15.62	5.3	
T_{ci}	13.27	1.2	0.125
T_{co}	26.55	1.2	

In comparison to several flow arrangements, counter flow heat exchanger best fulfils the necessary criteria of high thermal effectiveness. So, the current design is based on counter flow arrangement.

II: Selection of material:

The major applications of plate fin heat exchangers demand them to possess the properties of high thermal conductivity so as to enable high heat transfer. Further, the material needs to retain its strength and ductility even at low temperatures. Therefore, amongst all the materials, aluminium alloys (Al 3003 and Al 6061) were found to be most suitable for the construction of heat exchanger. In the current design, Al 3003 was selected as the base material on account of its low cost and high manufacturing ability.

III: Dimensioning of the heat exchanger

The dimensions of the heat exchanger need to decided in such a manner so that the heat exchanger can work efficiently within the operating pressure range. Thus, maximum limit of length, width and height should not exceed 1.5m, 0.7m and 0.7m. The length should be in the flow direction (horizontal). The value of width is assumed within the limit specified before and then other dimensions (length and height) are calculated on the basis of width.

IV: Pattern of layer stacking

The next step in the design process is to decide the pattern of layer stacking. The current design is based on 3 stream plate fin heat exchanger. Hence, the no. of layers for each side are taken as: hot 1 helium side, cold helium side, hot 2 helium side.

$$\text{Mass flow rate for each side} = \frac{\text{Total mass flow rate}}{\text{No. of layers}}$$

The layer stacking pattern which is capable of giving highest thermal effectiveness should be in the following manner:

1. Outermost layer should be the cold helium layer.
2. Warm stream layers should be just adjacent to the cold stream layers.

Let, hot 1 helium be represented as X, cold helium as Y and hot 2 helium as Z.

Then, the layer stacking can be done in the following ways:

YZY YXY YZY YXY

In the above layer stacking pattern, it can be observed that there is one layer of X to two layers of Y and one layer of Z to two layers of Y. Double banking (ratio of hot helium layers to cold helium layers = 1:2)

YZYXYZYXY (1:1)

In this layer stacking pattern, the double banking is equal to 1:1. This layer stacking pattern is balanced except for the presence of a half Y layer on either side. Therefore, it is expected that the outermost Y layer will be able to receive heat from one side only. The adjacent Z layer will reject more amount of heat to the outer layer (Y in this case) as compared to the heat rejected to the inner layer. This will occur from both the side such that the center Y layer will receive equal heat from both the sides.

On account of low density of cold helium stream, the pressure drop encountered is more. Thus, the number of cold streams should judiciously chosen so as to minimize the pressure drop. Higher the height and number of cold layers, lower is the pressure drop. Hence, to lower the pressure drop either no. of cold layers can be increased or height can be increased or both can be done for more reduction in pressure drop.

Selection of layer stacking pattern

In 1:2 arrangement, the plate between two cold streams does not participate in heat transfer, instead it enhances the axial conduction. In order to reduce axial conduction, the length of the heat exchanger needs to be increased which is undesirable. As far as efficiency is concerned, both the patterns are at par with each other. However, cost wise 1:1 arrangement is preferred over 1:2. In 1:1 arrangement, buckling occurs because cold stream (of 1.2 bar) is compressed by two sides of higher pressure stream (i.e. up to 12.99 bar) and medium pressure stream (i.e. up to 5 bar). Therefore, it was decided to go for 1:2 stacking pattern..

V: Calculation of free flow area or frontal area

Effective width:

$$W_{eff} = W - (2 * T_b) \tag{3.1}$$

Free flow area:

$$A_{xh1} = W_{eff} * n_{h1} * (b_{h1} - t) * (1 - (n * t)) \tag{3.2}$$

$$A_{xh2} = W_{eff} * n_{h2} * (b_{h2} - t) * (1 - (n * t)) \tag{3.3}$$

$$A_{xc} = W_{eff} * n_c * (b_c - t) * (1 - (n * t)) \tag{3.4}$$

Where, Ax = The frontal area or the free flow area

Tb = edge bar minimum thickness

VI: Fluid thermo physical properties

By using the following input data, property graphs were plotted.

Inlet temperature of hot 1 and hot 2 helium = 27.32 K.

Outlet temperature of hot 1 and hot 2 helium = 15.62 K.

Pressure for hot 1 helium = 12.99 bar.

Pressure for hot 2 helium = 5 bar.

Cold Helium:

Inlet temperature = 13.27 K.

Outlet temperature = 26.55 K.

Pressure = 1.2 bar.

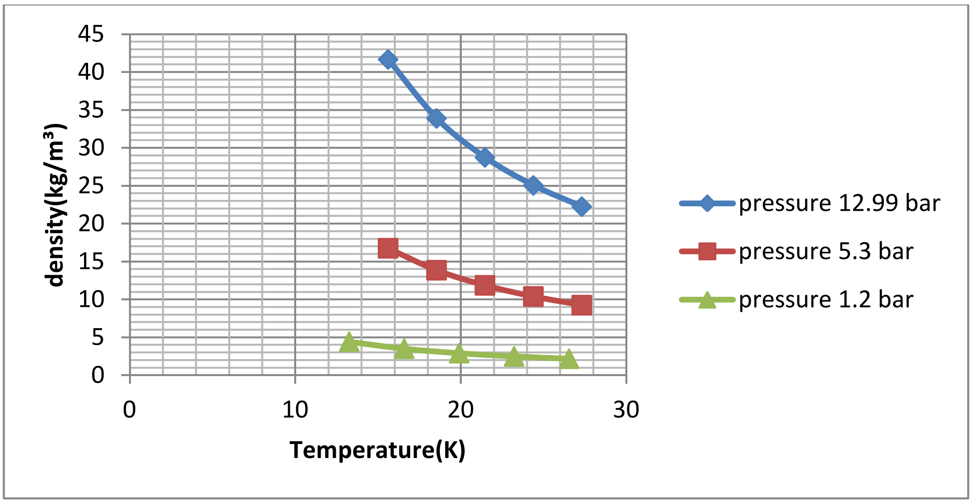


Fig.3.1 Temperature vs. Density

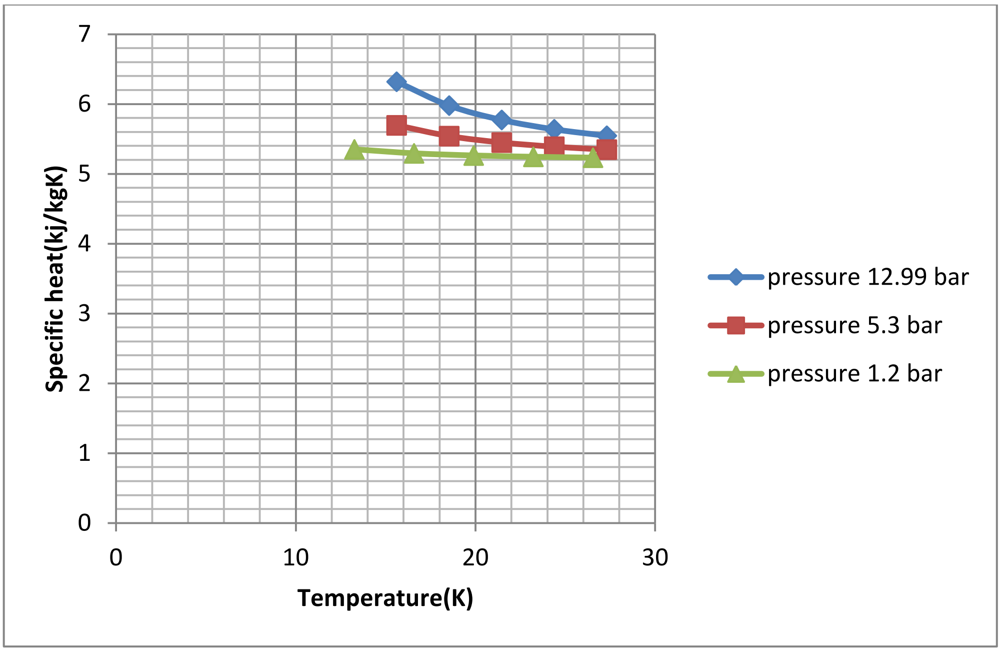


Fig.3.2 Temperature vs. Specific heat

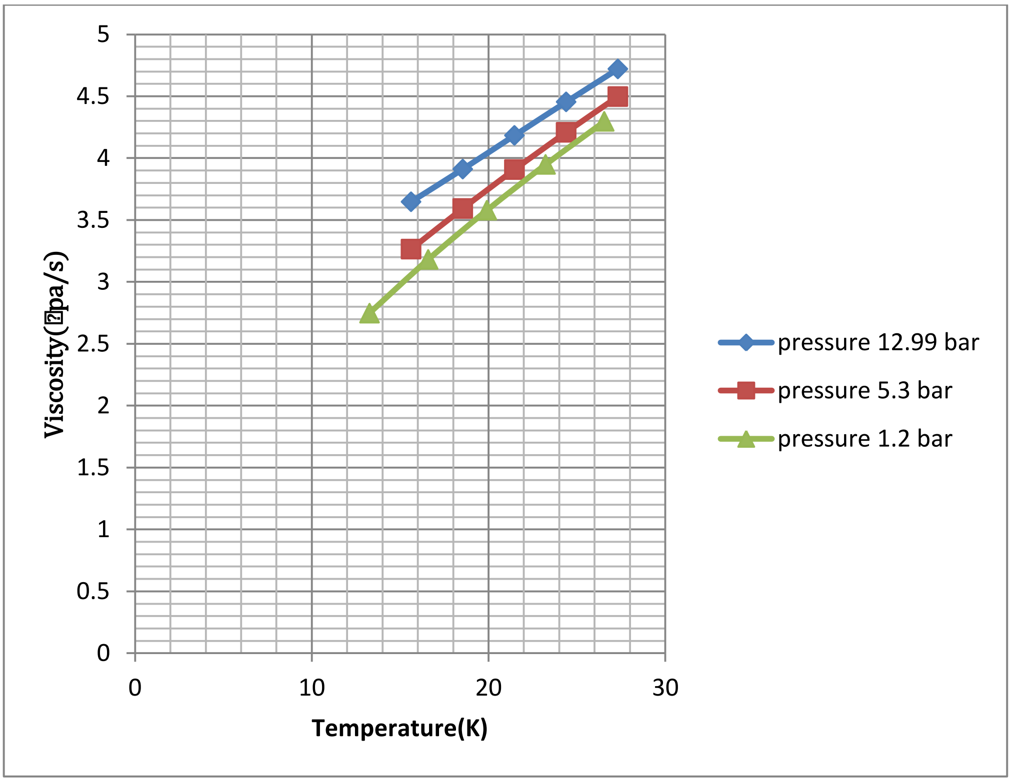


Fig.3.3 Temperature vs. Viscosity

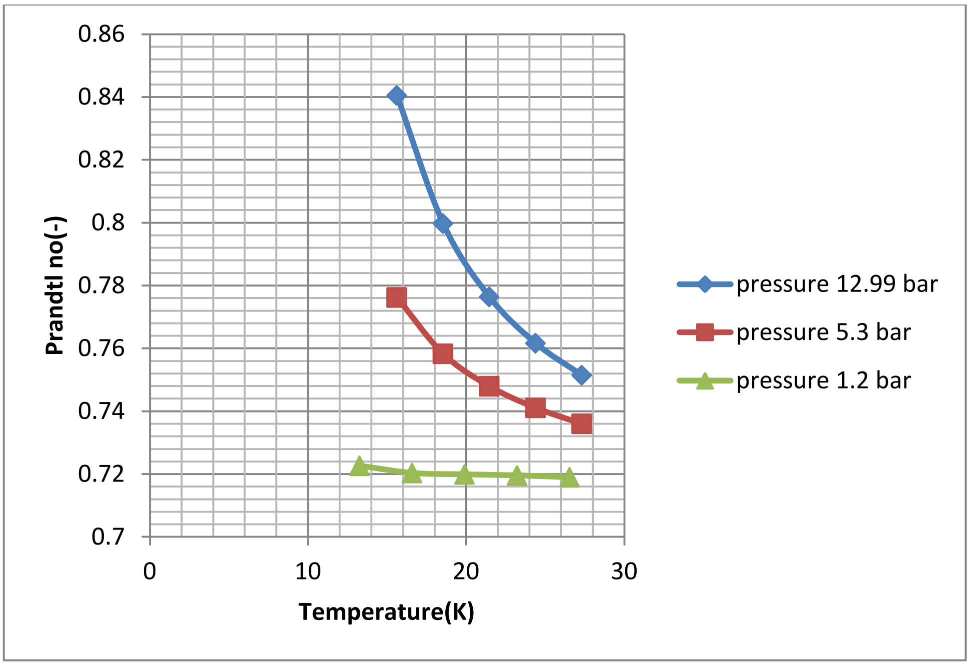


Fig.3.4 Temperature vs. Prandtl no.

Variations in properties for the helium fluid were found to be linear as shown in Fig. 3.1, 3.2, 3.3 and 3.4.Therefore for every property average value is taken corresponding to the inlet and outlet temperatures of the fluid stream.

VI: Log mean temperature difference (LMTD)

LMTD is mostly calculated for counter flow processes due to maximum heat transfer. Here, two hot streams and one cold stream are there. So, we have to divide the cold stream in such a way that, for each hot stream there are different outlet temperatures of cold stream, inlet temperature being same. Because, one is high pressure hot stream and other one is the medium pressure hot stream.

So, actual outlet temperature of cold1(c1) as per heat load from hot1(h1) is given as:

$$h_{c1-o} = h_{ci} + \frac{Q_{h1}}{\left(\frac{m_c * n_{c1}}{n_c}\right)} \tag{3.5}$$

$$T_{c1-o} = T_{co} + \frac{h_{c1-o} - h_{co}}{\left(C_{p_{T_{co}}}\right)} \tag{3.6}$$

Likewise, actual outlet temperature of cold2(c2) as per heat load from hot2(h2) is given as:

$$h_{c2-o} = h_{ci} + \frac{Q_{h2}}{\left(\frac{m_c * n_{c2}}{n_c}\right)} \tag{3.7}$$

$$T_{c2-o} = T_{co} + \frac{h_{c2-o} - h_{co}}{\left(C_{p_{T_{co}}}\right)} \tag{3.8}$$

Hence, log mean temperature difference (LMTD) for h1-c1-h1 and h2-c2-h2 combinations are given as,

$$LMTD_1 = \frac{(T_{h1-i} - T_{c1-o}) - (T_{h1-o} - T_{ci})}{ln\frac{(T_{h1-i} - T_{c1-o})}{(T_{h1-o} - T_{ci})}} \tag{3.9}$$

$$LMTD_2 = \frac{(T_{h2-i} - T_{c2-o}) - (T_{h2-o} - T_{ci})}{ln\frac{(T_{h2-i} - T_{c2-o})}{(T_{h2-o} - T_{ci})}} \tag{3.10}$$

VII: Analysis of heat transfer:

The heat transfer rate for both the hot and cold fluids can be expressed as below:

Hot fluid:

$$Q_h = m_h * C_h * (T_{hi} - T_{ho}) \tag{3.11}$$

Cold fluid:

$$Q_c = m_c * C_c * (T_{co} - T_{ci}) \tag{3.12}$$

As the fluid properties for both the hot and cold streams vary inside the heat exchanger, so the mean value of temperature difference taken for calculation is:

$$Q = UA\Delta T \tag{3.13}$$

VIII: Effectiveness (ε)

Effectiveness can be defined as a measure of the performance of any heat exchanger. It is expressed as follows:

$$\varepsilon = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}} \tag{3.14}$$

$$\varepsilon = \frac{C_h(T_{hi}-T_{ho})}{C_{min}(T_{hi}-T_{ci})} = \frac{C_c(T_{co}-T_{ci})}{C_{min}(T_{hi}-T_{ci})} \quad (3.15)$$

$$NTU = \frac{UA}{C_{min}} \quad (3.16)$$

Where NTU = no of transfer units

IX: Fin geometry

Generally, the offset strip fin is used in the manufacturing of plate fin heat exchanger. As the presence of surface interruption, it enhances the heat transfer by continuously interrupting the thermal boundary layer growth. Its heat transfer performance is much more than the plain fins at the expense pressure drop. Offset strip fins are used inside the cold box of the existing helium plant.

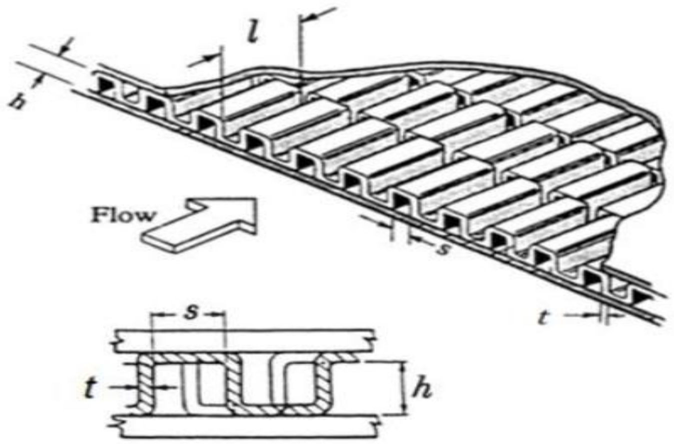


Fig.3.5 Serrated fin geometry

The expression for hydraulic diameter for serrated fin is as follows:

$$D_h = \frac{4*s*h*l}{(2*(sl+lh+ht))+ts} \quad (3.17)$$

Where h = fin height, in place of h we can use:

For hot stream 1 fin height = h_{h1}

For hot stream 2 fin height = h_{h2}

For cold stream fin height = h_c

X: Analysis of convective heat transfer coefficient (h)

The Reynolds number (Re) is a measure of flow characteristic of heat exchanger. It is directly proportional to the flow momentum rate and inversely proportional to the viscous force for a specified geometry.

$$Re = \frac{G * D_h}{\mu} \tag{3.18}$$

$$G = \frac{m}{A_x} \tag{3.19}$$

Where G = core mass velocity

The value of Re is different for all the streams. For the value of G in case of all the streams are, G_{h1}, G_{h2}, G_c for hot 1 , hot 2 and cold stream respectively for different mass flow rates and different values of free flow area such as A_{xh1}, A_{xh2}, A_{xc} respectively. Moreover D_h and μ values are also different for all the three streams.

It can be seen that temperature does not appear directly in the expression of friction factor also. Therefore, the f data determined at one temperature/pressure level are directly usable at other temperature/pressure. But it is seen that j and f are strong functions of fin geometries like fin height, fin spacing, fin thickness etc. Because fins are available in varied shapes, it becomes necessary to test each configuration individually to determine the heat transfer and flow friction characteristics for specific surface. For a given fin geometry, in general increase in heat transfer performance is associated with increase in flow friction and vice versa. Customarily, the ratio of j/f is taken as a measure of the goodness of the fin surface. Though the preferred fin geometry would have high heat transfer coefficient without correspondingly increased pressure penalty, the selection of particular fin geometry mainly depends on the process requirement; one can sacrifice either of heat transfer or pressure loss at the cost of other. For critical applications, direct experimental determination of j and f factors for each fin geometry remains the only choice. In the plate fin heat exchanger the common range of

Reynolds number is 500 to 3000 for most of the applications. The Reynolds number is kept low because the hydraulic diameter of the flow passages is generally small due to closely spaced fins and in such conditions operation with low density gases leads to excessive pressure drop unless the gas velocity in the flow passage is kept low. After several experiments and understandings many correlations are developed. Among them Manglik bergles correlation is used here.

Manglik bergles correlations

For 120<Re<10^4

$$j = 0.6522(Re)^{-0.5403}\left(\frac{s}{h}\right)^{-0.1541}\left(\frac{t}{l}\right)^{0.1499}\left(\frac{t}{s}\right)^{-0.0678}[x]^{0.1} \quad (3.20)$$

Where,

$$x = 1 + 5.269 * 10^{-5}(Re)^{1.34}(\frac{s}{h})^{0.504}(\frac{t}{l})^{0.546}(\frac{t}{s})^{-1.055}$$

$$f = 9.6243(Re)^{-0.7422}(\frac{s}{h})^{-0.1856}(\frac{t}{l})^{0.3053}(\frac{t}{s})^{-0.2659}[x]^{0.1} \quad (3.21)$$

Where,

$$x = 1 + 7.669 * 10^{-8}(Re)^{4.429}(\frac{s}{h})^{0.92}(\frac{t}{l})^{3.767}(\frac{t}{s})^{0.236}$$

Where h = fin height. In place of h, h_{h1}, h_{h2}, h_{hc} are used for hot 1, hot 2, cold stream respectively.

XI: Fin efficiency (*f*)

Assuming no heat transfer through the centre of the fin, treated as adiabatic, the fin efficiency of a fin is:

$$\eta_f = \frac{tanh\,(x)}{x} \tag{3.22}$$

Where $x = h\sqrt{\frac{hB}{2tk}}$

Outside h = fin height

Under root h = heat transfer coefficient

k= Thermal conductivity of Fin material

B= Banking Factor

Normally B value is taken as 2.25 for average condition.

Banking factor:

for hot stream = 4

for cold stream = 1

XII: Overall effective heat transfer surface area per parting sheet surface area

Considering serrated fins S_{eff}:

For cold stream,

$$S_{eff} = \{\left(2\eta lh + ht + sl + (0.5 * ts)\right) * \left(n * W_{eff}/l\right)\} \tag{3.23}$$

For hot stream,

$$S_{eff} = \{\left(\eta lh + ht + sl + (0.5 * ts)\right) * \left(n * W_{eff}/l\right)\} \tag{3.24}$$

Where h = fin height. In place of h, h_{h1}, h_{h2}, h_{hc} are used for hot 1, hot 2, cold stream respectively.

XIII: Actual height:

$$H = \{(n_{h1} * b_{h1}) + (n_{h2} * b_{h2}) + (n_c * b_c) + \left((n_{h1} + n_{h2} + n_c - 1) * t_p\right) + (2 * t_c)\} \tag{3.25}$$

XIV: Heat flux gradient and overall thermal resistance:

$$R_{eff1} = (\frac{1}{h*S_{eff}})_{h1} + (\frac{1}{h*S_{eff}})_c + (\frac{t+t_p}{k*W_{eff}*1}) \tag{3.26}$$

$$R_{eff2} = (\frac{1}{h*S_{eff}})_{h2} + (\frac{1}{h*S_{eff}})_c + (\frac{t+t_p}{k*W_{eff}*1}) \quad (3.27)$$

$$\mathrm{R_{tot1}} = \frac{\mathrm{R_{eff1}}}{\mathrm{n_{c-h1}}} \quad (3.28)$$

$$\mathrm{R_{tot2}} = \frac{\mathrm{R_{eff2}}}{\mathrm{n_{c-h2}}} \quad (3.29)$$

$$UA_1 = \frac{1}{\mathrm{R_{tot1}}} \quad (3.30)$$

$$UA_2 = \frac{1}{\mathrm{R_{tot2}}} \quad (3.31)$$

$$L_1 = \frac{Q_1}{(UA_1*LMTD_1)} \quad (3.32)$$

$$L_2 = \frac{Q_2}{(UA_2*LMTD_2)} \quad (3.33)$$

For L_1 , Q_1 is calculated in two ways. In between that maximum value is taken for optimization. They are given as follows:

$$Q_{h1} = m_{h1} * (h_{diff})_{h1} \quad (3.34)$$

$$Q_C \text{ for } h1 = \left\{\frac{\left(h_{diff_c}*m_c\right)}{n_c}\right\} * n_{c1} \quad (3.35)$$

Where m_{h1}= total mass flow rate of hot 1 stream

m_c= total mass flow rate of cold stream

h_{diff} = total enthalpy difference between the inlet and outlet of a stream

n_{c1} = n_{c-h1} = total no of cold 1 layers (i.e.no of hot 1 layers*2)

For L_2 , Q_2 is calculated in two ways. In between that maximum value is taken for optimization. They are given as follows:

$$Q_{h2} = m_{h2} * (h_{diff})_{h2} \tag{3.36}$$

$$Q_C \text{ for } h2 = \left\{\frac{\left(h_{diff_c} * m_c\right)}{n_c}\right\} * n_{c2} \tag{3.37}$$

Where m_{h2}= total mass flow rate of hot 2 stream

$n_{c2} = n_{c-h2}$ = total no of cold 2 layers (i.e.no of hot 2 layers*2)

Finally, we get two values of the length. As per the optimization we have to decide, which value of length should be taken.

Here A = $W_{eff} * 1$, where Length is considered as 1.

XV: Pressure drop:

Total drop in pressure for He stream (Serrated fins) is expressed as:

$$\Delta P_{total} = \Delta P_{frictional} + \Delta P_{gap} \tag{3.38}$$

Fictional loss is given by,

$$\Delta P_{frictional} = \frac{4fLG^2}{2\rho D_h} \tag{3.39}$$

Where, D_h value is calculated for three different streams. So, that frictional pressure drop is different for different streams.

Gap loss is given by,

$$\Delta P_{gap} = 2 * \in * (1 - \in) * (\frac{G^2}{2\rho}) \tag{3.40}$$

Where, $\in = \left(1 - \frac{t}{b}\right) * (1 - nt)$ (3.41)

No. of fin gaps for Serrated fin is given by,

$$N_{gap} = \frac{L}{l} \tag{3.42}$$

Where, $\Delta P_{gap(total)} = \Delta P_{gap} * N_{gap}$ (3.43)

Here, gravitational pressure drop is neglected, as the heat exchanger is placed horizontally.

3.2 HEADER DESIGN

To keep the pressure loss minimal the distributor fins should be plain or plain perforated, since it is not required to transfer heat in this region. The header is designed for the major purpose of reduction in pressure drop and ideal distribution for liquid or gas flow. For both hot helium flow mitred Side entry distributor with plain fins having angle of deflection = 90°. In this type of distributor, the inlet and outlet should be installed on the opposite sides so as to maintain nearly equal pressure drop and a uniform distribution of flow. For cold helium flow full end distributor is set. The drop in pressure for a full end distributor is less than through any other design. The current header design is based on the average length of flow.

3.3.1 Design of distributor

The same values of mean temperatures and the properties to that of thermal section calculations are taken. The dimension of the header depends geometrically on the inlet radius of the header, since the headers are made out as half cylindrical surfaces. The number of layers in header is taken as that in the thermal section for a particular stream. Fin height is same as used in thermal section. Plain fins are usually used for distributor designs, since, they provide with better flow distribution with lower pressure drop. Pressure drop at the inlet section consists of the reduction in pressure in the header tank is expressed by:

$$\Delta P_{ht} = 1.5\left(\frac{G^2}{2\rho}\right) \tag{3.44}$$

Pressure loss caused by friction is expressed as:

$$\Delta P_{fr} = \frac{4fLG^2}{2\rho D_h} \tag{3.45}$$

Pressure drop in the mitre-section = frictional loss + section head loss

$$\Delta P_{sec} = (k_d + 1)\left(\frac{G^2}{2\rho}\right) - \left(\frac{G^2}{2\rho}\right) \tag{3.46}$$

Pressure drop at the exit of distributor is expressed as:

$$\Delta P_{gap} = 2 * \in * (1 - \in) * \left(\frac{G^2}{2\rho}\right) \tag{3.47}$$

Where,

$$\in = \left(1 - \frac{t}{b}\right) * (1 - nt) \tag{3.48}$$

Using the above methodology, outlet distributor is designed.

CHAPTER 4

4. Results and Optimization

4.1 Controlling Parameters: Generally in the design of heat exchangers, following parameters are to be optimized. They are as follows.

- Size of the heat exchanger
- Pressure Drop
- Weight of the heat exchanger
- Cost of the heat exchanger

The primary objective of the process of optimization is to obtain the required value of pressure drop while maintaining the least possible size of the heat exchanger. The length, breadth and height of the heat exchanger have to within the maximum limit of 1.5m, 0.7m, and 0.7m. It is understood that when the size of the heat exchanger is reduced, its cost and weight also get reduced simultaneously. Optimization is performed to calculate the least possible size while maintaining the needed pressure drop. The dimensions of fins and no. of layers are varied for obtaining the minimum size. The pressure drop allowed for hot stream are 1, hot 2 and cold stream are 2 mbar, 2 mbar, 1 mbar respectively.

4.2 Optimization for Plate Fin Heat Exchanger

4.2.1 Based on the variation of fin thickness and fin density: *F*in thickness is varied from 0.2 mm to 0.9 mm for each of the fin density from 400 fins/m to 900 fins/m by keeping number of hot 1 helium layers = 8, hot 2 helium layers =10, cold helium layers = 36, h_{h1}= 6.5 mm, h_{h1}= 6.5 mm, h_c = 9.5 mm constant. Moreover assuming width = 0.4 m , flow length = 5 mm.

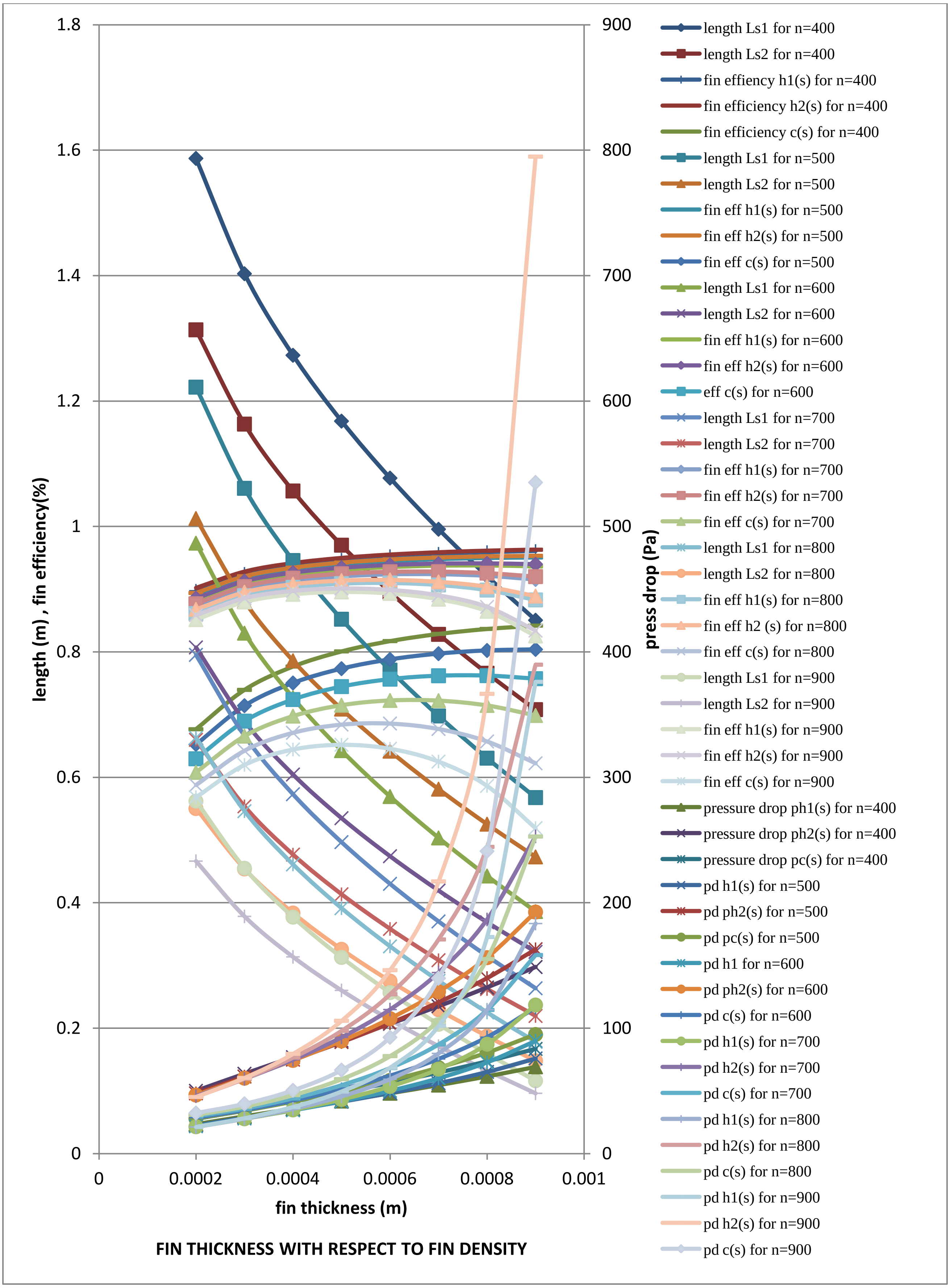
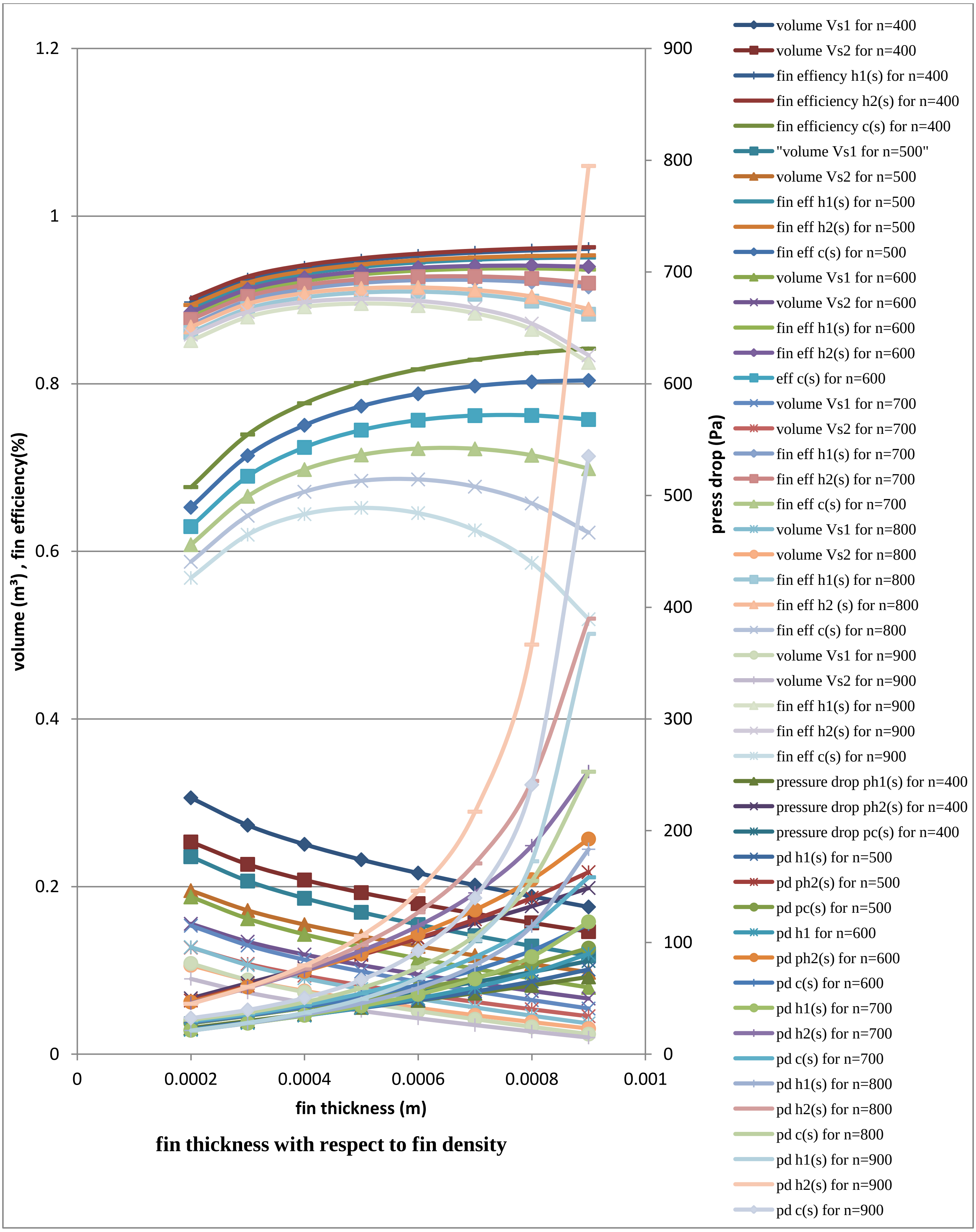


Fig.4.1 Changing Fin thickness w.r.t Fin density

From the above Fig.4.1, we can conclude the following:

- Pressure drop of hot 1, hot 2, cold stream decreases with decreasing the fin density and decreasing fin thickness also.
- Length of all the streams increases with decreasing fin density and decreasing fin thickness.
- Fin efficiency increases with increasing fin thickness and increasing fin density for 400, 500, 600 fins/m. But for increasing fin density 700, 800, 900 fins/m, fin thickness increase up to 0.5 mm and then decreases as the fin efficiency expression is a tan hyperbolic function .Tan hyperbolic function's value is maximum up to 0.5 mm thickness then it decreases, so also the efficiency.



4.2 Changing Fin thickness w.r.t Fin density

From the above Figure.4.2, we can conclude the followings:

- Pressure drop of hot 1, hot 2, cold stream decreases with decreasing the fin density and decreasing fin thickness also.
- Volume of all the streams increases with decreasing fin density and decreasing fin thickness.
- Fin efficiency increases with increasing fin thickness and increasing fin density for 400, 500, 600 fins/m. But for increasing fin density 700, 800, 900 fins/m, fin thickness increase up to 0.5 mm and then decreases as the fin efficiency expression is a tan hyperbolic function. Tan hyperbolic function's value is maximum up to 0.5 mm thickness then it decreases, so also the efficiency.
- Here, from the above graph, minimum volume is taken corresponding to the given pressure drop range. From that value of minimum volume, corresponding fin thickness and fin density values are taken.
- From figure.4. it is observed that the required pressure drop is achieved at 500 fins/m and at fin thickness =0.2 mm.

4.2.2 Based on variation of fin height and no. of layers

Fin height of cold stream is varied from 4 mm to 10 mm for total number of layers 39, 45, 54, 60, 66, 72 in the ratio of 1:2 by tentatively keeping h_{h1}= 6.5 mm, h_{h1}= 6.5 mm, fin density = 500 fins/m constant. Moreover assuming width = 0.4 m, flow length = 5 mm.

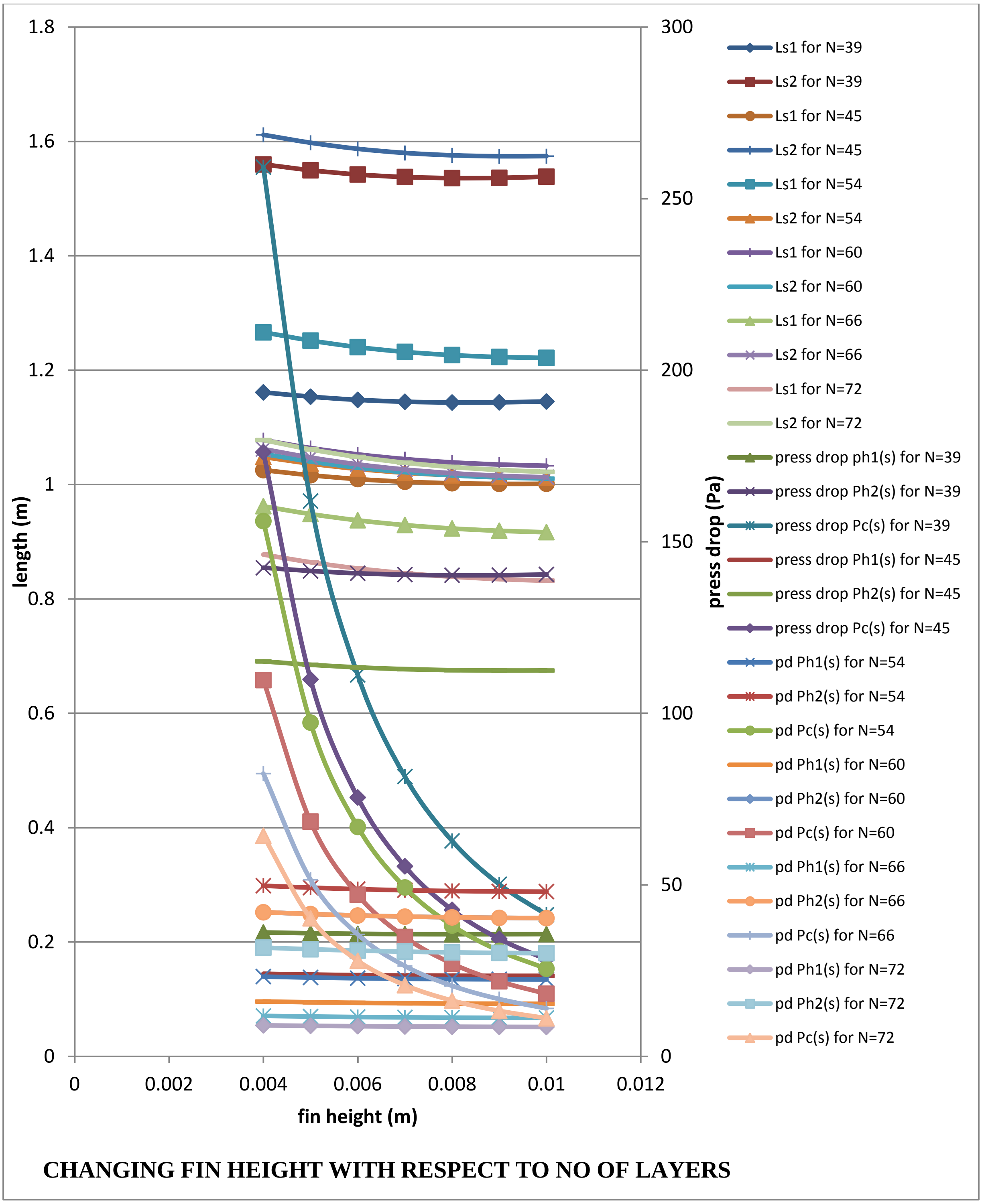


Fig.4.3 Changing fin height w.r.t no of layers

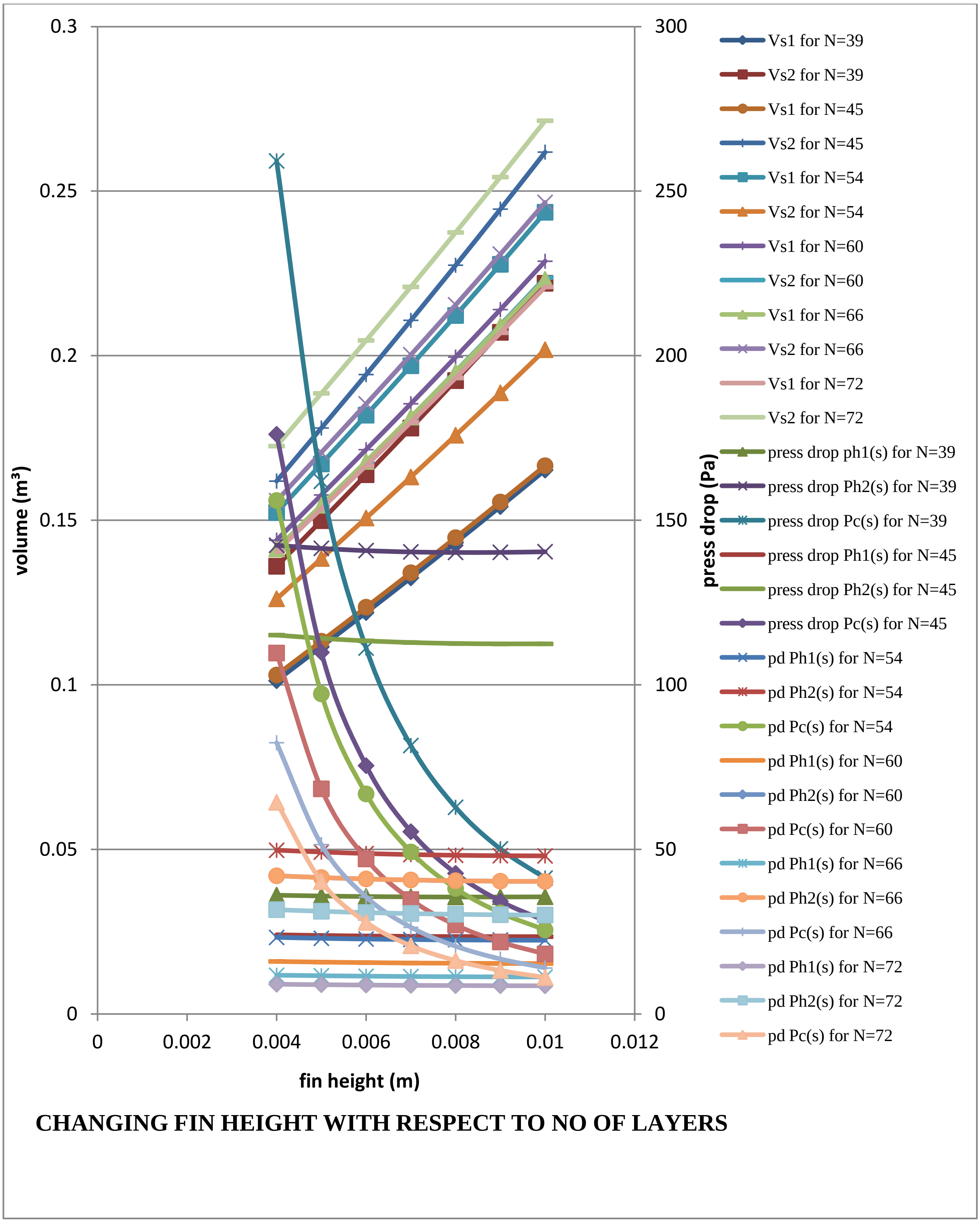


Fig.4.4 Changing fin height w.r.t no of layers

From the above Figs 4.3 , 4.4, we can conclude the followings:

- When the no of layer increases, pressure drop decreases and fin height increases.
- Volume decreases as fin height decreases.
- Here, from the above graph, minimum volume is taken corresponding to the given pressure drop range. From that value of minimum volume, corresponding fin height for cold stream and no of layers values are taken.
- It is observed from the above Figures.4.3 and 4.4 , the required pressure drop is achieved at following parameters:
    - ➢ height of fin = 6.5 mm for both hot streams
    - ➢ fin height = 9.5 mm for cold stream
    - ➢ No of layers = 54
    - ➢ Ratio = 1:2 for layer stacking arrangement.

4.2.3 Header and Distributor:

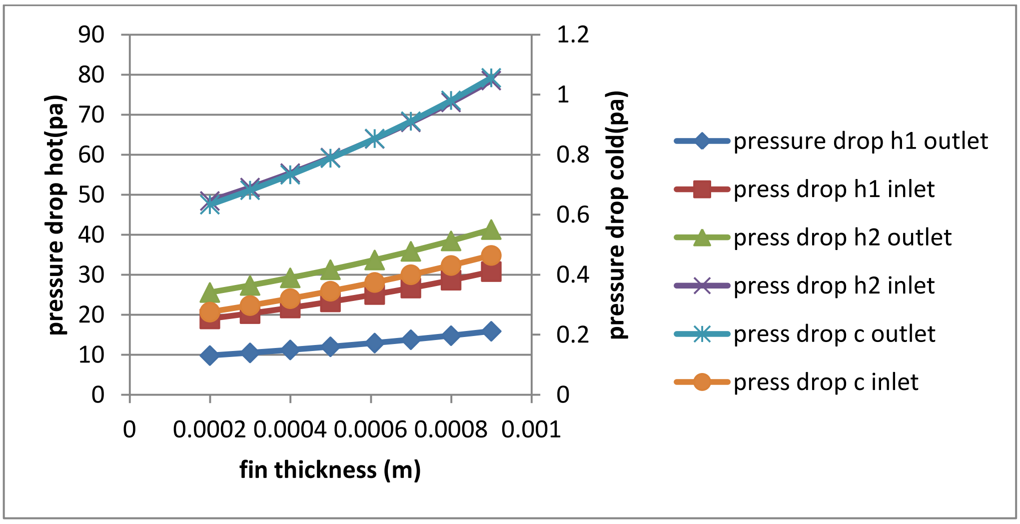


Fig.4.5 Pressure drop Variation vs Fin thickness

From figure.4.5, it is observed that as per the pressure ranges the optimized values of fin thickness is 0.61 mm and 0.2 mm for both hot streams and cold stream respectively.

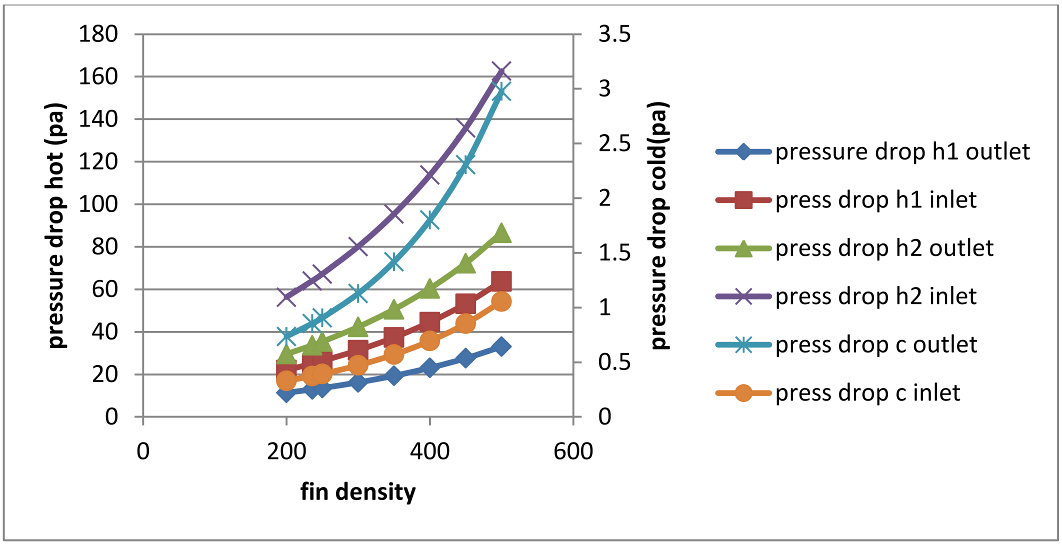


Fig.4.6 Pressure drop vs Fin density

From Figure.4.6, it is observed that the desirable pressure loss is obtained at the fin density for hot stream = 236 fins/m whereas for cold stream, fin density is same as used in thermal section which is further optimized.

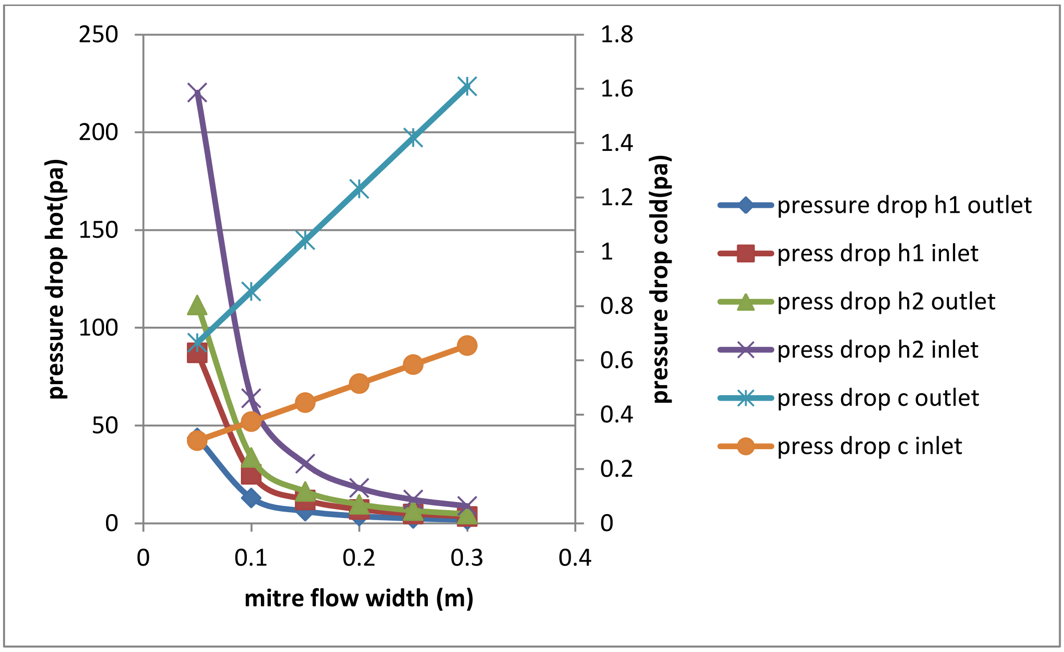


Fig.4.7 Pressure drop w.r.t Mitre flow width

The optimized value of mitre flow width = 100 mm. Because the distributor cross section should be smaller than that of the thermal cross section.

Moreover for fin height, it is observed that the requisite pressure drop is achieved at fin height = 6.5 mm for hot stream and for cold stream fin height = 9.5 mm.

4.3 Output Of Optimization: Optimized parameters are shown in table 4.2 and 4.3:

Table.4.2 Output of optimization without margin

Pressure drop (h1)	0.22 mbar
Pressure drop (h2)	0.48 mbar
Pressure drop (hc)	0.28 mbar
No. of layers (N)	54
No of hot layers	8
No of hot 2ayers	10
No of cold layers	36
Fin height (h1)	6.5 mm
Fin height (h2)	6.5 mm
Fin height (c)	9.5 mm
Fin thickness (t)	0.2 mm
Fin density (n)	500 fins/m
Length of segment	5 mm
Overall Thermal coefficient (UA)	2504.025 W/K
Volume required (V)	0.235 m^3
Heat transfer (Q)	3936.431 W
Length (L)	1.222 m
Width (W)	400 mm
Height (H)	526.8 mm

Here 20% thermal margin accounts for the inefficiencies of other heat exchangers above this 4th heat exchanger. Results obtained from optimization is shown in table 4.3.

Table.4.3 Output of optimization with margin

Pressure drop (h1)	0.27 mbar
Pressure drop (h2)	0.57 mbar
Pressure drop (hc)	0.33 mbar
No. of layers (N)	54
No of hot layers	8
No of hot 2ayers	10
No of cold layers	36
Fin height (h1)	6.5 mm
Fin height (h2)	6.5 mm
Fin height (c)	9.5 mm
Fin thickness (t)	0.2 mm
Fin density (n)	500 fins/m
Length of segment	5 mm
Overall Thermal coefficient (UA)	2504.025 W/K
Volume required (V)	0.283 m^3
Heat transfer (Q)	3936.431 W
Length (L)	1.466 m
Width (W)	400 mm
Height (H)	526.8 mm

Table.4.4 Output of optimization with international standards

Overall Thermal coefficient (UA)	4677.217 W/K
Volume required (V)	0.088 m^3
Heat transfer (Q)	4315.8 W
Pressure drop (h1)	0.74 mbar
Pressure drop (h2)	2.49 mbar
Pressure drop (hc)	1.20 mbar
Length (L)	744 mm
Width (W)	300 mm
Height (H)	446 mm
No. of layers (N)	48
Fin height (h1)	6.3 mm
Fin height (h2)	6.3 mm
Fin height (c)	8.9 mm
No of hot layers	8
No of hot 2ayers	8
No of cold layers	32
Fin thickness (t)	0.2 mm
Fin density (n)	867 fins/m
Length of segment	3.2 mm

Here in table 4.5, optimized parameters of header and distributor are given by:

Table 4.5 Output of header and distributor

Parameters	Hot-1 stream	Cold stream	Hot-2 stream
Distributor type	Mitred side	Full end	Mitred side
Fin type	perforated	serrated	perforated
Fin density (mm)	236	500	236
Fin thickness (mm)	0.61	0.2	0.61
Fin height (mm)	6.5	9.5	6.5
No of layers (-)	8	36	10
Diameter of holes (mm)	2	-	2
Pitch (mm)	4	-	4
Pressure drop (pa)	Outlet 12.952 Inlet 25.076	Outlet 0.8539 Inlet 0.3745	Outlet 33.7123 Inlet 63.955

4.4 OPTIMIZED 3D DRAWING

Here, optimized results are taken into account for the validation, Hence 3D modelling of PFHE is done in catia software.

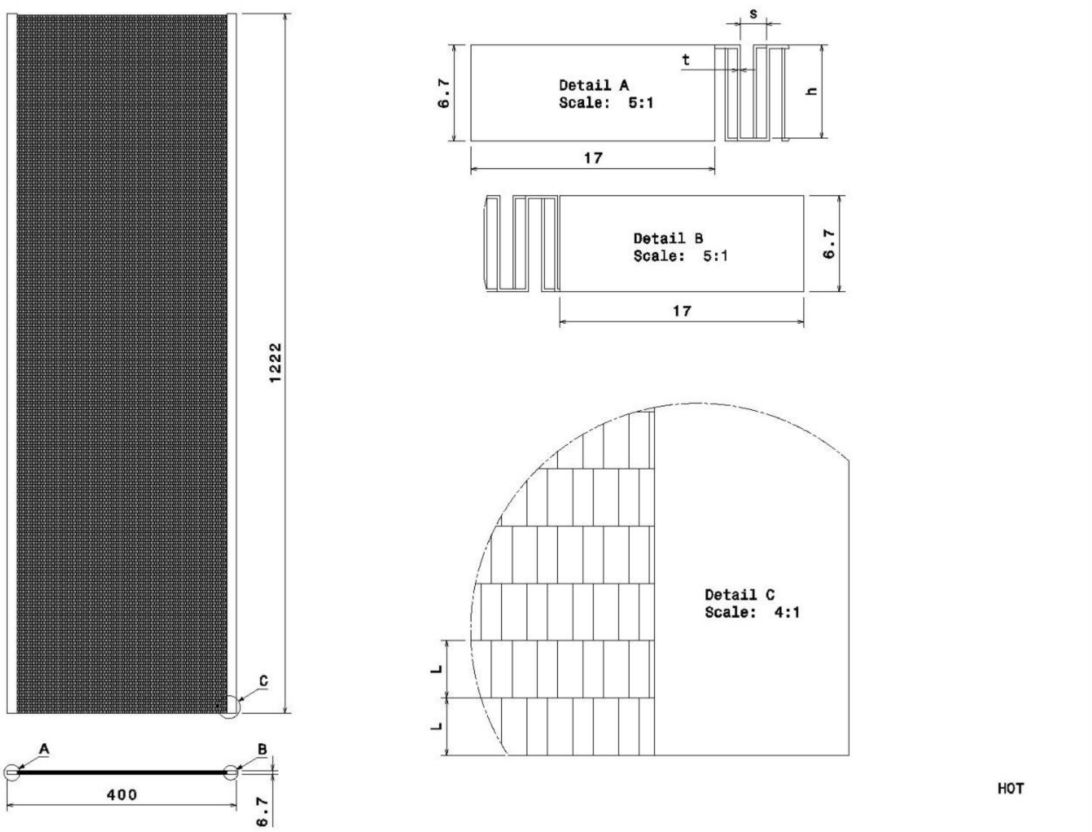


Fig.4.8 Assembly for both hot 1 and hot 2 stream

Figure 4.8 shows the controlling parameters for hot stream.

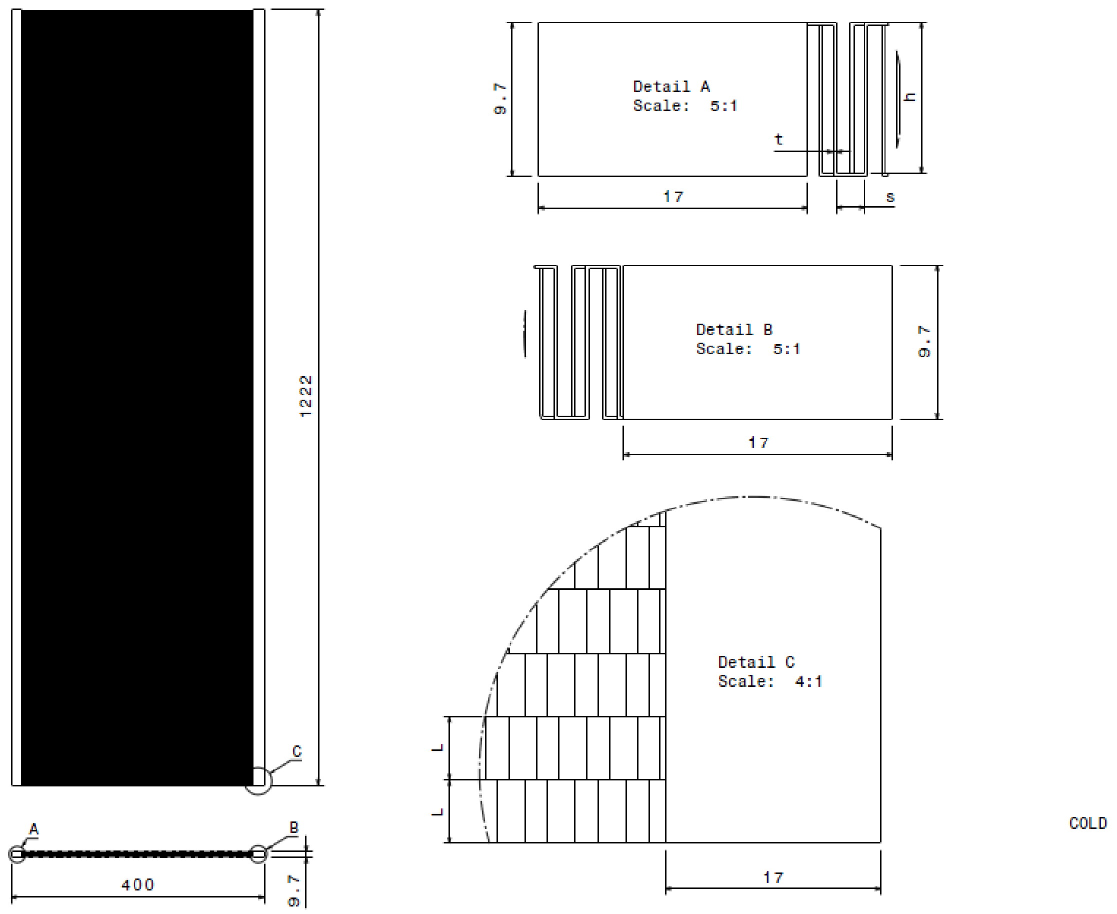


Fig.4.9 Assembly for cold stream

Figure 4.9 shows the controlling parameters for cold stream.

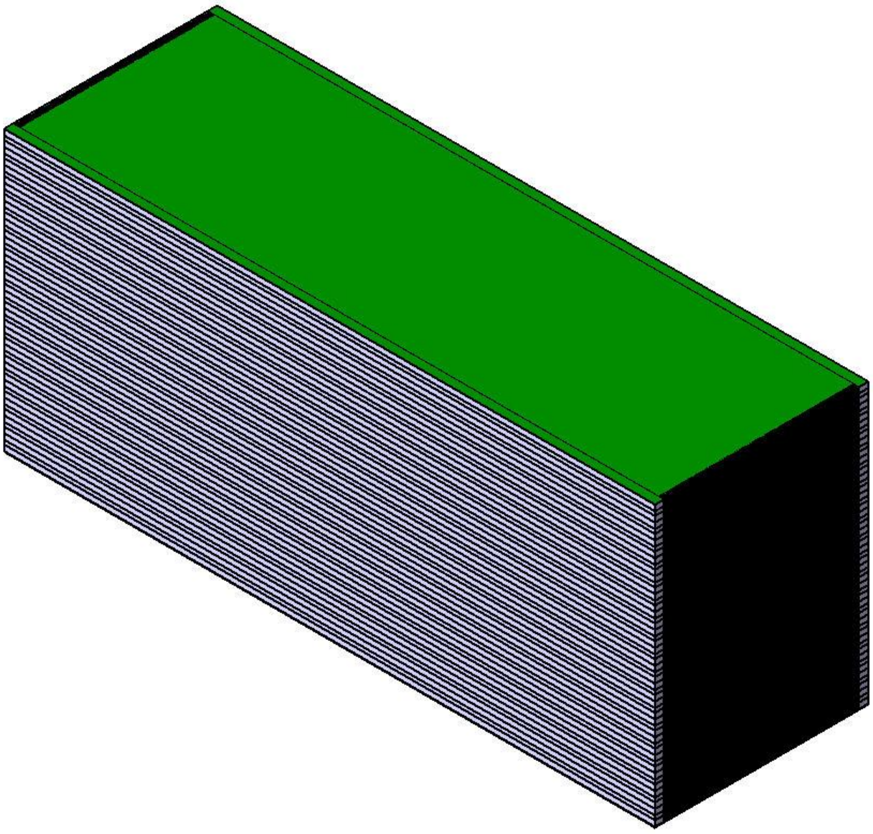


Fig.4.10 3D drawing of Plate fin heat exchanger

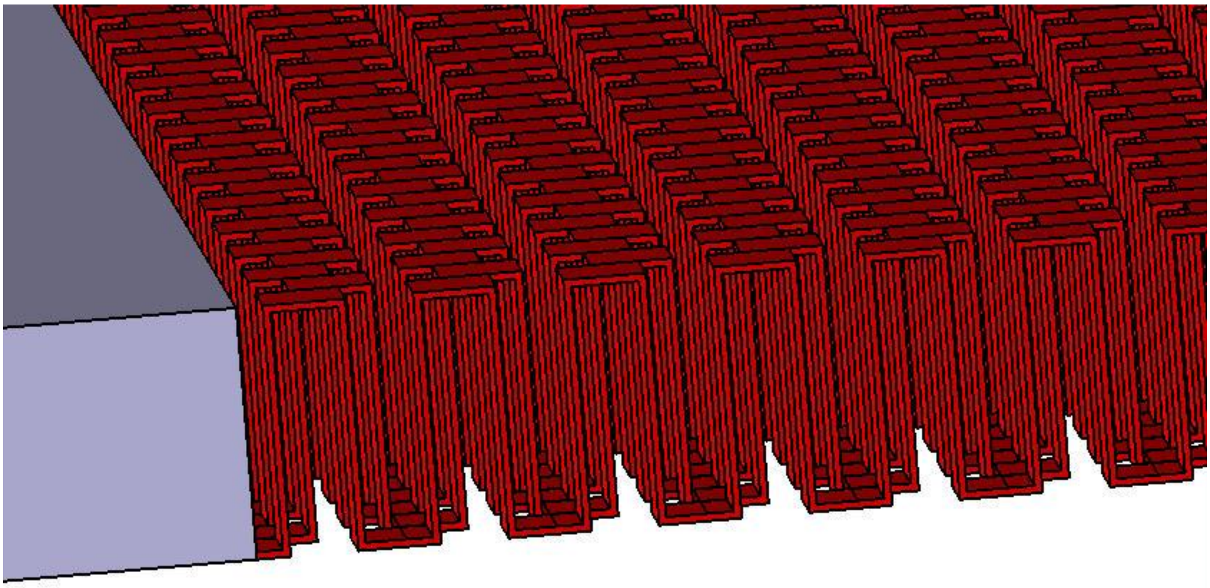


Fig.4.11 3 Dimensional representation of serrated fins

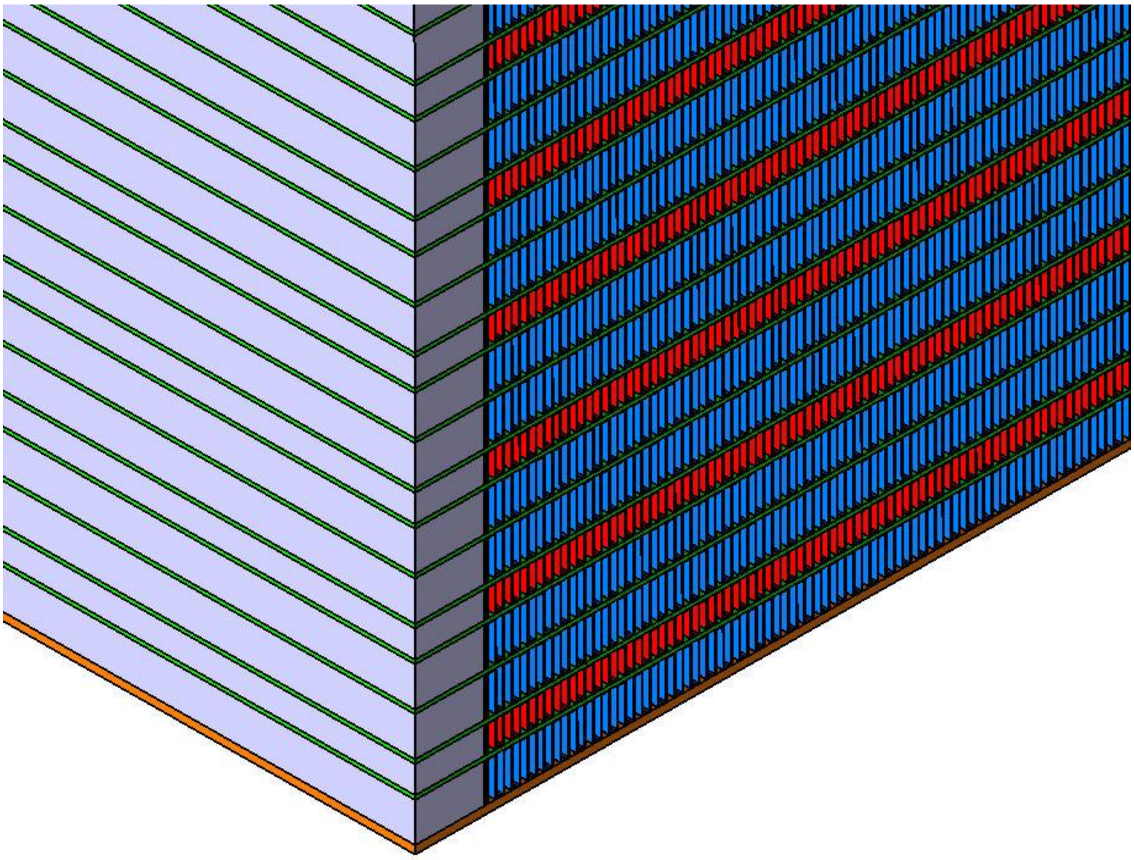


Fig.4.12: 3 Dimensional representation of Plate Fin Heat Exchanger

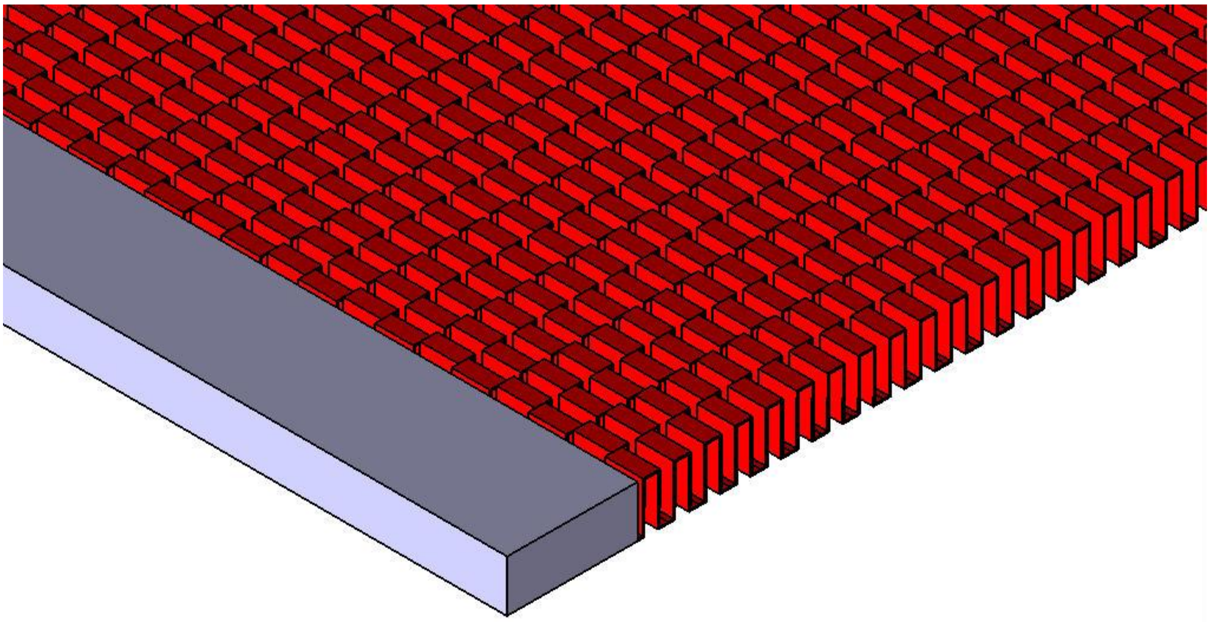


Fig.4.13: 3D drawing of serrated fins

4.5 Validation:

Validation with existing plant (data given by Air Liquide to IPR) is done with analytical procedure and the results are shown in the table 4.6 given below.

Table.4.6 Validation with existing plant

Parameters	Existing plant	Analytical results
Length	880 mm	944 mm
height	440 mm	446 mm
Pressure drop (h1)	-	73.63 pa
Pressure drop (h2)	348.9 pa	249.64 pa
Pressure drop cold	-	120.580 pa
Fin height (h1) = (h2)	6.33 mm	6.33 mm
Fin height cold	8.89 mm	8.89 mm
Fin thickness	0.203 mm	0.203 mm
Fin density	867 fins/m	867 fins/m
No of layers h1	8	8
No of layers h2	8	8
No of layers cold	32	32

Considering similar mass flow rates and fin parameters as per the three stream heat exchanger of the existing plant, the length and height of the heat exchanger are calculated to give 944 mm and 446 m respectively whereas this length and height as per supplied document is 880 mm and 440mm. So, the deviation is within 10%. The pressure drop for hot 2 stream is calculated to be 249 Pa and as per supplied document for the existing plant, it is 348 Pa and hence, deviation is within 30%.

Validation of plant to be developed is done with aspen tech software. The results are shown in table 4.7 given below.

Table.4.7 Validation with Aspentech software

Parameters	Appolo standards	Aspentech software
Length	1222 mm	1222 mm
Height	526.8 mm	526.8 mm
Pressure drop (h1)	22.469 pa	27 pa
Pressure drop (h2)	48.0694 pa	67 pa
Pressure drop cold	27.9383 pa	44 pa
Fin height (h1) = (h2)	6.5 mm	6.5 mm
Fin height cold	9.5 mm	9.5 mm
Fin thickness	0.2 mm	0.2 mm
Fin density	500 fins/m	500 fins/m

No of layers h1	8	8
No of layers h2	10	10
No of layers cold	36	36

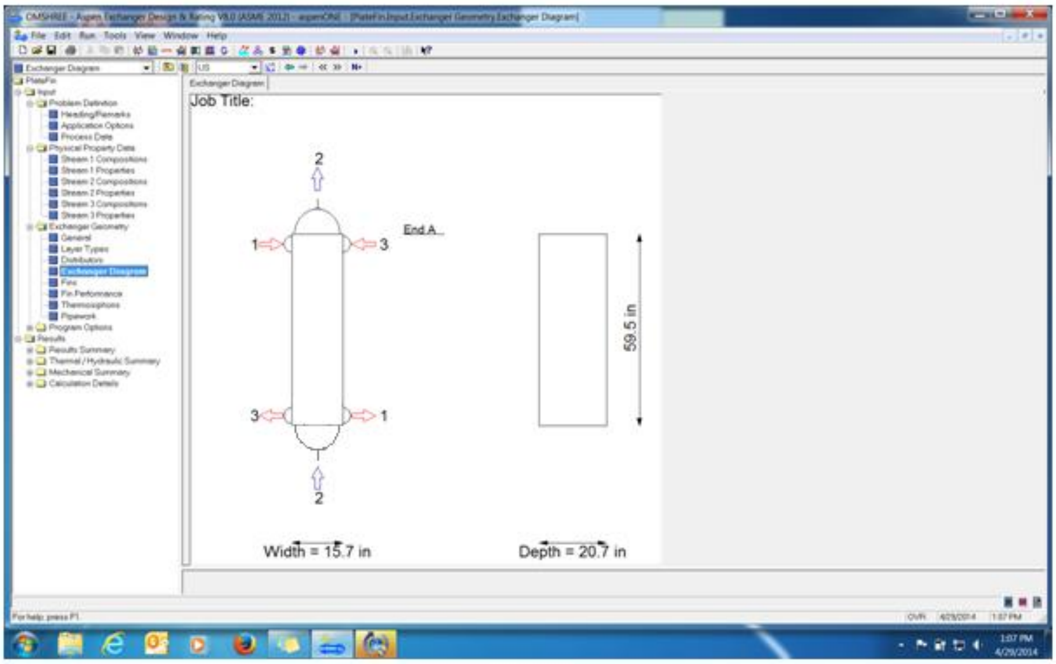


Fig.4.14: 3-stream heat exchanger from Aspentech results

CHAPTER 5

5. CONCLUSIONS

The design of plate fin heat exchanger is done in the excel sheet followed by analysis and optimization by plotting graphs to get lower values of volume, pressure drop and maximum thermal efficiency.

- AL 3003 is used for analysis, because of its low density, high thermal conductivity and high strength, ductility at low temperature.
- Serrated fins are used because pressure drop is estimated within the allowed value and also reduced size is achieved.
- Finally, the overall dimension of the heat exchanger is given as follows:
  - L = 1.222 m
  - H =0.526 m
  - W= 0.4 m
  - V = 0.235 m^3
- Pressure drop of hot 1,hot 2, cold stream are 0.22, 0.48, 0.28 mbar respectively.
- The validation is done by Aspentech software and International standards. The overall deviation between the existing plant and the analytical results are given as follows:
  - L within 10% deviation
  - Height within 10% deviation
  - Pressure within 30% deviation
- Overall deviation between Aspentech software and Appolo standards are given in terms of Pressure drop of hot 1 stream, hot 2 stream and cold stream as 30%.

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