DESIGN OF PLATE FIN HEAT EXCHANGER WITH OFFSET STRIP FINS FOR THE LIQUEFACTION OF NATURAL GASES

A thesis submitted in partial fulfillment of the requirement for the award of degree

of

Master of Technology in Marine Engineering and Management

by

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CERTIFICATE

This is to certify that the thesis entitled **"Design of Plate Fin Heat Exchanger with Offset Strip Fins for the Liquefaction of Natural Gases"** submitted by Mukul Kashiwal to Indian Maritime University Kolkata Campus for the award of the degree in Master of Technology in Marine engineering and Management, is a bonafide record of the project work carried out by him under our supervision. The contents of this thesis, in full or in parts have not been submitted to any other institute or University for the award of any degree or diploma.

The Project has been carried out at Indian Maritime University Kolkata Campus.

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Sincerely,

Mukul Kashiwal

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LIST OF SYMBOLS AND ABBREVIATION

f	Fin density or frequency, fins/meter
δ	Fin thickness
L_f	Fin length
H	Fin Height
λ	Fin offset length
S	Normal fin spacing
b	Distance between plates or plate spacing
P_f	Pitch of fin
δ_w	Plate thickness
N_p	Number of passages
n_f	Total no. of fins at entry with subscript 1 and 2 for hot and cold fluid
A_p	Total primary area with subscript 1 and 2 for hot and cold fluid, m^2
n_{off}	No. of offset strip fins with subscript 1 and 2 for hot and cold fluid
A_f	Total fin area with subscript 1 and 2 for hot and cold fluid, m^2
A_t	Total surface area with subscript 1 and 2 for hot and cold fluid, m^2
A	Heat transfer area of the heat exchanger, m^2
A_c	Free flow area available for the fluid with subscript 1 and 2 for hot and cold fluid, m^2
A_{fr}	Frontal area with subscript 1 and 2 for hot and cold fluid, m^2
D_h	Hydraulic diameter with subscript 1 and 2 for hot and cold fluid, m
σ	Porosity of the heat exchanger with subscript 1 and 2 for hot and cold fluid
V_p	Volume of the heat exchanger with subscript 1 and 2 for hot and cold fluid side, m^3
β	Surface area density with subscript 1 and 2 for hot and cold fluid, m^2/m^3
G	Core mass velocity with subscript 1 and 2 for hot and cold fluid, Kg/m^2-sec
Re	Reynold Number with subscript 1 and 2 for hot and cold fluid
j	Colburn factor with subscript 1 and 2 for hot and cold fluid
f	Friction factor with subscript 1 and 2 for hot and cold fluid
$\eta_{1,2}$	Single fin efficiency for hot and cold fluid side respectively
η_o	Overall fin efficiency with subscript 1 and 2 for hot and cold fluid
A_w	Wall conduction Area, m^2
R_w	Thermal resistance

U Overall heat transfer coefficient, W/K

C Heat capacity rate with subscript h or c for hot and cold fluids, W/K.

c_p Specific heat at constant pressure with subscript h or c for hot and cold fluids, J/kg-K

C_R Heat capacity rate ratio, dimensionless

D_e Equivalent diameter of the flow passage, m

h Convective heat transfer coefficient, W/m^2K

K_c Contraction coefficient

K_e Expansion coefficient

K Conductivity of the fin material, W/m- K

L_2 Fluid flow (core) length of the heat exchanger, m

$\dot{m}$ Mass flow rate with subscript h or c for hot and cold fluids, kg/sec.

NTU Number of heat transfer units

Pr Prandtl number of the fluid

Q Heat load, W

f_d Friction factor used for this circulation

k_d Velocity distribution coefficient

C_c Jet contraction ratio

GREEK SYMBOLS:

μ Fluid dynamic viscosity, pa-sec

ρ Fluid density, kg/m3

ε Effectiveness of heat exchanger, dimensionless

η Fin efficiency, dimensionless.

ΔP Pressure drops of hot and cold fluid with subscripts h or c, KPa.

SUBSCRIPTS:

c,h subscripts c or h for cold and hot fluid

max maimum

min minimum

m mean

i Inlet

o Outlet or Overall

w Wall

ABBREVIATION:

PFHE - Plate Fin Heat Exchanger

No. - Number

ABSTRACT

In the cryogenic systems such as liquefier, cryo-coolers, etc. the heat exchangers are one of the most critical components. The heat exchanger used in cryogenic applications must have high effectiveness to produce a proper refrigerating effect and it should not be less than 85%. It has already experimented that if the value of the effectiveness of the heat exchangers falls below the design value, then the no liquid will be produced. On a general basis, there are so many heat exchangers available in the industry which are used in the cryogenic works but there is a special category of the heat exchangers that are available, which are now used widely because of there because of their compactness, low weight, and high effectiveness. These heat exchangers are known as the compact heat exchangers.

The objective of this work is to design a plate fin heat exchanger, which is when used in the liquefaction cycle must be able to liquefy the natural gases and to produce 1000 Kg liquid in a day, which can be transported easily for the various purposes through various ways. This work has been carried on the plate fin heat exchanger with selecting the offset strip fin, which is the most reliable fin design in the liquefaction applications. The work includes the two cryogenic fluids, one is methane and the other is propane. Propane is used as the coolant to bring down the temperature of methane to this close that when passed through the expander or by precooling in the liquefaction cycle it gets liquefied easily. Catia software was used to model the component and Matlab programming has also been done to design a plate fin heat exchanger. As the geometry were too complex to analyse in the software because of the physical memory limitation, a small symmetric part of the heat exchanger model were imported in the analysis software called Ansys 18.1 for the validation of the results and it can be concluded from the observations that the model is capable to do the desired function. A small variation in the calculation and analysis results has been observed, and the effectiveness of the heat exchanger was found to be above 90 %. Aluminium 3003 has been used as the material for the fins strips and for separating plate.

CHAPTER 1. INTRODUCTION

1.1 INTRODUCTION

A plate fin heat exchanger is a kind of compact heat exchanger which is known for its low weight to volume ratio, high surface compactness, ability to handle multiple streams, and low-temperature properties. Normally these heat exchanger are used in the liquefiers because of their high effectiveness such as 0.85 or above. Heat transfer in this type of heat exchanger is enhanced by using the extended surfaces or fins. The construction of the plate fin heat exchanger used in this work includes the stacks of offset fins which are brazed to thin flat plate (also called parting sheets) and each stack of fins the separated by these thin flat plate. The parting sheet and the fin stacks are brazed together to provide the mechanical stability and a proper channel for heat transfer through the conduction. Beside thin plates and stack of fins, side bars are also used in the construction to prevent the leakage of the fluid. The heat exchange in this heat exchanger takes place primarily by the parting sheets and secondarily by the fins while the fluid flow through the passages created by these. The core of the plate fin heat exchanger has been shown in figure 1 and 2:

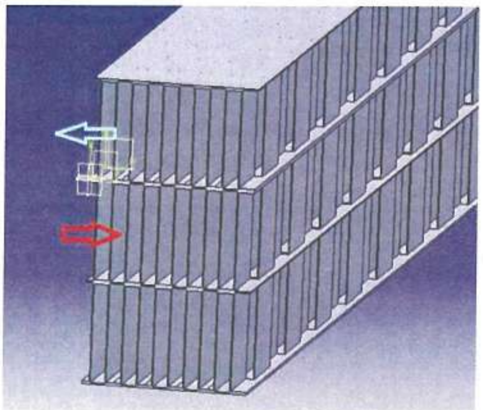

Figure 1. Small Portion of PFHE

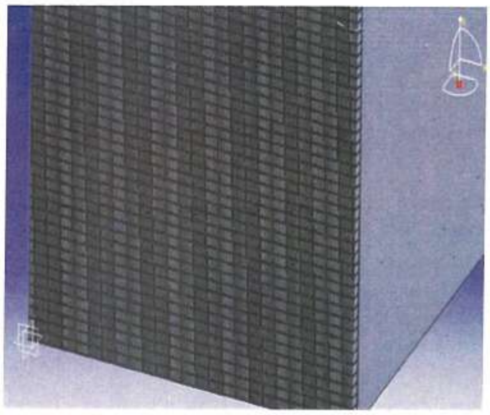

Figure 2. PFHE Core

1.2 ADVANTAGES AND DISADVANTAGES OF PFHE

ADVANTAGES

- The PFHE offers better thermal performance than the other extended surface heat exchanger.
- Through the PFHE a very high thermal effectiveness of up to 95% or may be above can be achieved in cryogenic applications.
- The performance of the plate fin heat exchanger can be enhanced by the use of multi stream and different conditions.

- The PFHE are used in those applications where the precision in temperature approaches as low as 1°C between single-phase fluids and 3°C between multiphase fluids is required.
- The brazed aluminium PFHE are most widely used or primary choice in the cryogenic applications because of their low weight to volume ratio, high surface compactness, ability to handle multiple streams, and with desirable low-temperature properties.

DISADVANTAGES

- The passages are narrow in the plate-fin exchangers which makes them wide open to the fouling and they are difficult to clean by mechanical means, that's why they are only used in applications where the fluid is already clean.
- The use PFHE is limited to a certain pressure and temperature range.
- There is relatively high pressure drop occurs in these heat exchangers due to narrow passage.

1.3 APPLICATIONS

The PFHE are used over a wide range of temperatures and pressures for gas-gas, gas-liquid and multi-phase heat transfer applications. They are used in the following applications:

- Natural gas liquefaction plants
- Petrochemical plants
- Gas treatment plants
- Helium liquefaction plants
- Fuel cells
- Process heat exchangers
- Heat recovery plants
- Pollution control systems
- Fuel processing and conditioning plants
- Ethylene and propylene production plants
- The Brazed aluminium plate fin exchangers are widely used in the aerospace industry.
- The largest kind of plate fin heat exchangers are used for cryogenic air separation and LPG fractionation and a single unit can be of several meters in length.

- They are used in environment control system of the aircraft, avionics cooling, hydraulic oil cooling and fuel heating.
- Because of their compactness these are widely used in the automobile and air conditioning industries as both are space conscious.

1.4 FIN TYPES

The plate fin exchangers are mostly used for liquid-to-gas and gas-to-gas applications. The extended surfaces in the plate fin heat exchanger are used because of the low heat transfer coefficients in gas flows. We uses the unique types of extended surfaces or fins to enhance the heat transfer. These special fins or extended surfaces at the cost of pressure drop provides the high heat transfer coefficient. The following types of fins available for use:

1.4.1 OFFSET STRIP FIN

The offset strip fin (OSF) is mainly used in the plate fin heat exchanger for applications such as liquefaction, air separation etc. The offset strip have rectangular shape; it is cut into small strips of length and are arranged in such a way that each alternate strip of fin is offset by 50% of the fin pitch. The main variables in offset strip fins are fin thickness, fin height, fin offset length etc. The OSF geometry is characterized by high heat transfer coefficients and high heat transfer area per unit volume. Joshi and Webb described the heat transfer mechanism through OSF that the heat transfer enhancement is obtained by periodic growth of laminar boundary layers on the fin length. In the OSF as the friction factor increases the pressure drop also increases and with that the enhancement in the heat transfer takes place. Salient features of the OSF are following:

1. OSFs are used in low Reynolds number up to 1000, also in some higher Reynold number application. If the Reynold no. is high, the j colburn factor will decrease but due to high drag force the friction factor will remain constant.
2. The OSFs provides the heat transfer 1.5 to 4 times the heat transfer by the plain fins but the higher pressure drop takes place in the OSFs.
3. These mainly used in air separation plants where high thermal effectiveness at low mass velocities is required and in liquefaction of natural gases.

4. They are also used for the low Reynolds number applications where the accurate performance prediction is required such as some aerospace applications. Fins has been shown in figure 3.

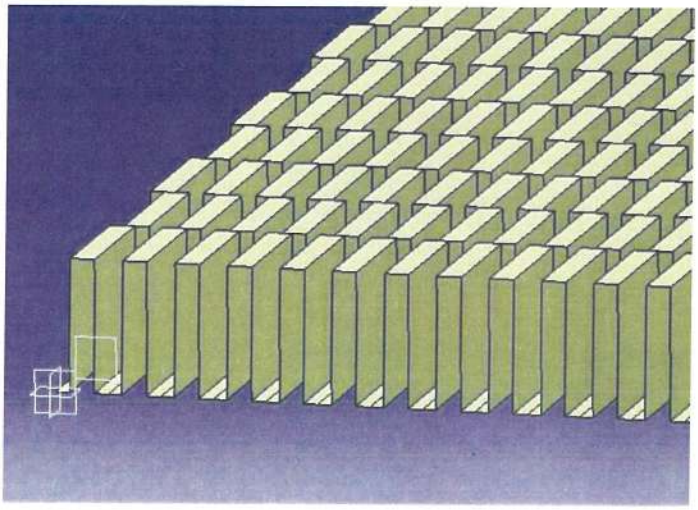

Figure 3. Offset Strip Fin

1.4.2 LOUVER FINS

In Louver fins the entire slit fin is rotated 20°–60° relative to the airflow direction unlike in the OSFs, where we only offset the alternate strip by 50% of its pitch. For the same strip width, the louvered fin geometry provides heat transfer coefficients comparable to the OSF. These are formed by cutting the sheet metal of the fin at intervals and by rotating the strips of metal thus formed out of the plane of the fin. A very high performance can be achieved through the Louver fins by varying the louver angle, louver width, and louver form. The Offset strip fins and parallel louver fin both have small strips aligned parallel to the flow. The heat transfer is enhanced in the louvered fins by providing multiple flat plate leading edges which have high values of heat transfer coefficient. The strip length of the louvered fin is generally more than that of offset strip fins and the louvered fin gage is generally thinner than that of OSFs. The louver fins have a disadvantage over the OSFs which is that they generally form a triangular passage which is not as strong as an OSF structure.

Louvered fin are basically used in those applications where Reynold no. lies in the range from 100 to 5000. Louvered fin surfaces are used widely for heat exchangers (evaporators and condensers), air-conditioning, engine cooling equipment and aircraft oil & air coolers. Because of their compactness, light weight, and low pumping power for a given heat transfer duty, the louver fins are used widely in aerospace applications. Fins has been shown in figure 4.

Figure 4. Louver Fin ; Source : Northern Radiator

1.4.3 SERRATED FINS

Serrated fins is similar to OSF. They differed from the OSFs in the manufacturing process and are formed by simultaneously folding and cutting alternative sections of fins. These fins are also known as lanced fins. Fins has been shown in figure 5. Source [28]

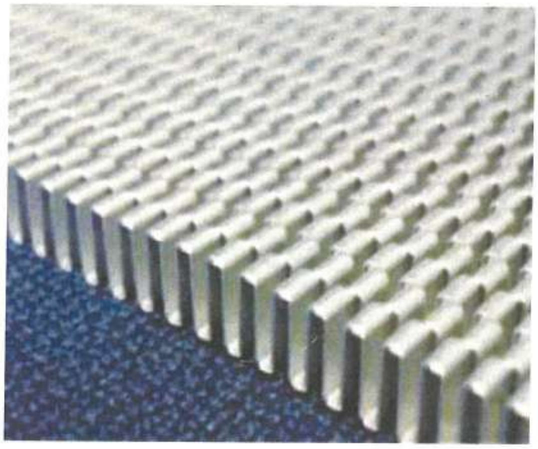

Figure 5. Serrated Fin

1.4.4 HERRINGBONE OR WAVY FIN

Herringbone or Wavy Fin are structurally non- interrupted fins in the flow direction because of this the waveform in the flow direction provides effective interruptions to flow and induces very complex flows also they are mostly likely not to catch the particulates. The augmentation is due to Goertler vortices, which form as the fluid passes over the concave wave surfaces. These fins are comparable in the performance to the offset strip fins. In these fins the wall corrugations increases the heat transfer to 3 times compare to the plain fins. The fins are used in those applications where the Reynold no. ranges from 6000 to 8000. The wavy fins are often a better choice at the higher Reynolds numbers typical of the hydrocarbon industry; the smooth surface allows the friction factor to fall with increasing Reynolds number. Fins has been shown in figure 6. Source [28]

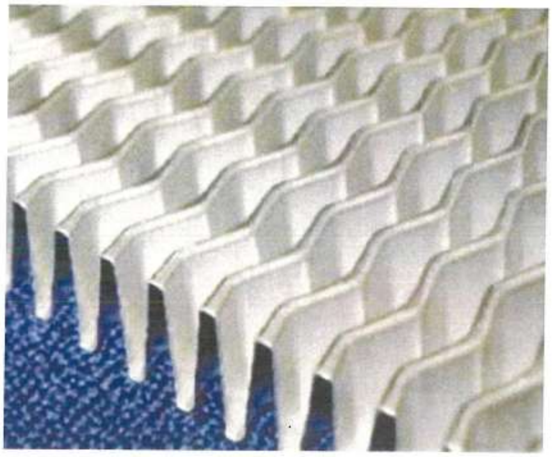

Figure 6. Herringbone Fin

1.4.5 PLAIN FIN

The simplest type of extended surface is plain fin corrugation. These fins are straight and uninterrupted in the fluid flow direction. Two configurations available are - Triangular and rectangular. For the same passage size and fin thickness the triangular fin structurally as strong as the rectangular fin. Plain fin surfaces have relatively low pressure drop but high ratio of heat transfer to pressure drop, these both characteristic of plain fin is similar to the flow through the small bore tubes. In this case of straight fins the value of the heat transfer coefficient is low as the boundary layers tend to be thick. In this type of fin, if the Re based on hydraulic diameter is less than 2000, then the theoretical laminar flow solutions for j and f can be used. The Plain fins or extended surfaces are preferred for very low Reynolds number applications and in applications where the pressure drop is very critical. Fins has been shown in figure 7. Source [28]

1.4.6 PLAIN-PERFORATED FIN

The main purpose of the perforated plain fins use, is the ability of perforated holes to promote turbulence which increases the local heat transfer coefficient compared to plain fins and ability of its surface which is prone to flow-induced noise and vibration. The variables of perforation fins are its size and longitudinal and transverse spacing as major perforation variables. The Perforated fins have two types of perforation- one is round and second is rectangular perforations. A metal strip is first perforated, then corrugated. Perforated areas vary from about 5% to 25% of the sheet area.

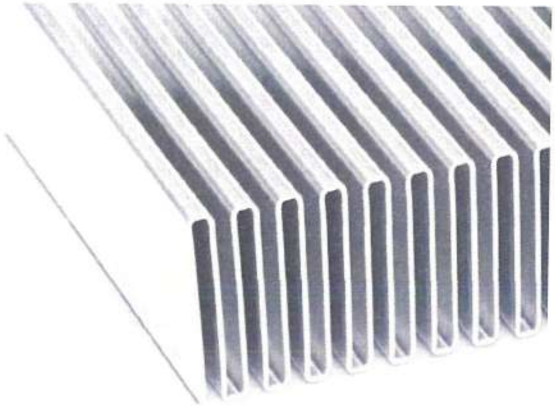

Figure 7. Plain Fin

The perorated fins are used in the oil coolers, in boiling applications and it permits inter-channel fluid migration. Perforated fins were once used in two-phase cryogenic air separation exchangers, but now they have been replaced by OSFs. Fins has been shown in figure 8. source [28]

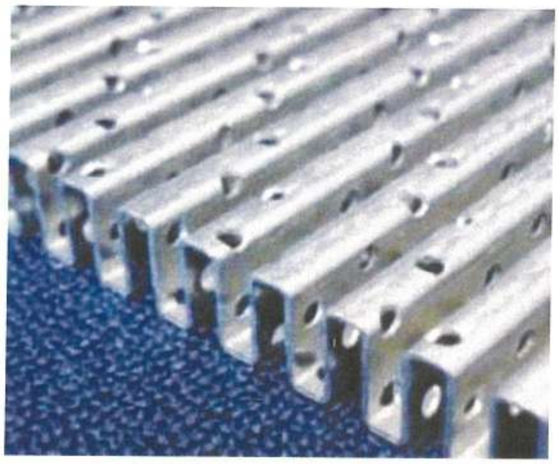

Figure 8. Plain Perforated Fin

1.4.7 PIN FINS

The structure of the Pin Fins includes a cylinder over which an array of the fins are attached to the upper wall. Pins can be of circular or elliptical shapes. These are manufactured at a very high speed and continuously from a thin wire. They are somewhat related to the OSFs in case of heat transfer enhancement mechanism by repeated boundary-layer growth and wake dissipation. The pin fins have some disadvantages over the offset strip fins and louver strip fins that the surface compactness achieved by the pin fin geometry is much lower compared to them and there is chance of vibration occurrence due to vortex shedding behind the circular pins. The pin fins are used in those applications where the very low Reynolds number ($Re < 500$) is achieved and the pressure drop is of no major concern. Presently they are in use in the electronic cooling devices with generally free convective flows over the pin fins and to cool turbine blades. Fins has been shown in figure 9. Source : Global Sources

Figure 9. Pin Fin

1.5 HEAT TRANSFER AND FLOW FRICTION CHARACTERISTICS

The flows in plate fin heat exchanger are generally very complex, having flow separation, reattachment, recirculation, and vortices. Such flows significantly affects Nu and f for the exchanger surfaces such as offset strip fin, pin fin, louver fin etc. Analytically or numerically the relation data between Nu (or j) and f (or Eu) vs. Re cannot be determined, so the data are obtained experimentally. One of the earliest and the most authoritative sources of experimental j and f data on plate and fin surfaces were presented by the Kays and London Kays and London (1998) and Webb (1994) through their experimental setup. Kays and London conducted experiments on different types of plate and fin surfaces and observed from experiments that the heat transfer coefficient and friction factor f of surfaces which have the same diameter are different mutually as per the fin geometrical properties. The expression for j and f data is obtained separately for each surface type. J and f so presented are applicable to surfaces of any hydraulic diameter, provided a complete geometric similarity is maintained.

Offset strip fins:

[13] Kays is the one who made the first attempt at analytical modelling of heat transfer and friction loss in offset strip fins. Then he proposed a modified laminar boundary layer solution that includes form drag contribution of blunt fin edges. [16] Manson made the first attempt at developing predictive equations. However the dissimilar geometries data was employed: scaled up and actual offset strip fins, louvered fins and finned flat tubes. [15] Wieting developed an empirical correlation from experimental heat transfer and flow friction data on 22 offset strip fin surfaces of Kays and London, London and Shah[17] over two Reynolds number ranges: Re< 1000 and Re >2000.

For Re ≤ 1000

$$j = 0.483\ \left(\frac{l}{D_h}\right)^{-0.162}\left(\frac{s}{h}\right)^{-0.184}(Re)^{-0.536}$$

$$f = 7.661\ \left(\frac{l}{D_h}\right)^{-0.384}\left(\frac{s}{h}\right)^{-0.092}(Re)^{-0.712}$$

For Re ≥ 2000

$$j = 0.242\ \left(\frac{l}{D_h}\right)^{-0.322}\left(\frac{s}{h}\right)^{0.08}(Re)^{-0.368}$$

$$f = 7.661\ \left(\frac{l}{D_h}\right)^{-0.384}\left(\frac{s}{h}\right)^{-0.092}(Re)^{-0.712}$$

The prediction of j and f is done by extrapolating the equations to their respective transition zone. Although mostly 85% of all available data were correlated within $\pm$ 15% for friction factor and $\pm$ 10% for heat transfer, a few points showed discrepancy as high as $\pm$ 40%. Wieting's correlation is used effectively for the design of practical heat exchangers, keeping the precaution in extrapolating the data to fins with geometrical parameters outside the recommended range. [14] Joshi and Webb conducted flow visualization experiments to identify the transition from laminar flow and concluded that as the flow rate increases, oscillating velocities develop in the wakes, which lead to vortex shedding and also further increase in Re occurred. The observations were made that the onset of oscillating flow and the consequent change in the wake structure found to correspond approximately to the departure from the laminar log linear behaviour of j and f. A wake width based equation was devolved to determine the critical Reynolds number. Then they developed an analytical model in the laminar zone based on the numerical solution done by Sparrow and Liu [18] and a semi empirical method has been used for the turbulent region.

For Laminar (Re$\leq Re^*$),

$$j = 0.53\ (\frac{l}{D_e})^{-0.15}(\frac{s}{h})^{-0.14}(Re)^{-0.5}$$

$$f = 8.12\ (\frac{l}{D_e})^{-0.41}(\frac{s}{h})^{-0.02}(Re)^{-0.74}$$

For turbulent (Re $\geq Re^* + 1000$),

$$j = 0.21\ (\frac{l}{D_e})^{-0.24}(\frac{t}{D_e})^{0.02}(Re)^{-0.4}$$

$$f = 1.12\ (\frac{l}{D_e})^{-0.65}(\frac{t}{D_e})^{0.17}(Re)^{-0.36}$$

$$Re^* = 257\ (\frac{l}{s})^{1.23}(\frac{t}{l})^{0.58}\ D_e(t + 1.328\ (\frac{Re}{lD_e})^{0.5})^{-1}$$

Where, Re^* = Critical Reynolds number for heat transfer and pressure drop considerations.

The empirical correlation for j and f factors were proposed by the authors with verified experimental data on 21 different heat exchanger geometries and their own observations on scaled up geometries. They were able to correlate 82% of the f data and 91% of the j data within ±15%. [19] Manglik and Bergles examined the data the heat transfer and friction data given by Kays and London [2] for 18 offset strip fin surfaces. The equations have been developed that describe the asymptotic behavior of the data in the deep laminar and fully turbulent zones. These asymptotes have been combined to give the single predictive equation for j and f which are valid for laminar, turbulent and transition zones.

$$j = [0.6522(\frac{t}{s})^{-0.0678}(\frac{t}{l})^{0.1499}(\frac{s}{h})^{-0.1541}(Re)^{-0.5403}] \times$$
$$[1 + 5.629 \times 10^{-5}(\frac{t}{s})^{-1.055}(\frac{t}{l})^{0.546}(\frac{s}{h})^{0.504}(Re)^{1.340}]^{0.1}$$

$$f = [9.6243\ (\frac{t}{s})^{-0.2659}(\frac{t}{l})^{0.3053}(\frac{s}{h})^{-0.1856}(Re)^{-0.7422}] \times$$
$$[1 + 7.669 \times 10^{-8}(\frac{t}{s})^{0.236}(\frac{t}{l})^{3.767}(\frac{s}{h})^{0.920}(Re)^{4.429}]^{0.1}$$

Through the above equations all of the heat transfer data and approximately 90% of the friction data can be caculated within ±20 %.

Maiti and Sarangi calculated velocity, pressure and temperature fields in plate and fin passages using the CFD tool. They obtained correlations for the non-dimensional heat transfer coefficient, j and pressure drop characteristic, f in terms of Reynolds number and other geometrical parameters using the results obtained through the both computation and experiments. They even used the multiple regression to obtain some of the constants in the correlation through the numerically computed results and the rest of the constants from experimental data over the same geometry using the setup in the laboratory.

For laminar (Re≤ Re^*)

$$j = 0.36\left(\frac{t}{s}\right)^{-0.063}\left(\frac{l}{s}\right)^{-0.27}\left(\frac{h}{s}\right)^{0.275}(Re)^{-0.51}$$

$$f = 4.67\left(\frac{t}{s}\right)^{-0.104}\left(\frac{l}{s}\right)^{-0.181}\left(\frac{h}{s}\right)^{0.196}(Re)^{-0.70}$$

For Turbulent (Re≥ Re^*)

$$j = 0.18\left(\frac{t}{s}\right)^{-0.05}\left(\frac{l}{s}\right)^{-0.184}\left(\frac{h}{s}\right)^{0.288}(Re)^{-0.42}$$

$$f = 0.32\left(\frac{t}{s}\right)^{-0.023}\left(\frac{l}{s}\right)^{-0.185}\left(\frac{h}{s}\right)^{0.221}(Re)^{-0.286}$$

The critical Reynolds number for heat transfer coefficient is given by,

$$Re^* = 1568.58\left(\frac{t}{s}\right)^{-0.217}\left(\frac{l}{s}\right)^{-1.433}\left(\frac{h}{s}\right)^{-0.217}$$

For the pressure drop the critical Reynolds number is given by,

$$Re^* = 648.23\left(\frac{t}{s}\right)^{-0.196}\left(\frac{l}{s}\right)^{0.1}\left(\frac{h}{s}\right)^{-0.06}$$

- Hydraulic diameter

$$D_h = \frac{4 \times Free\ flow\ Volume}{Total\ heat\ trasnfer\ area}$$

$$D_h = \frac{4 \times A_c}{P}$$

$$D_h = \frac{4 \times A_c}{A/l}$$

The calculation of A_c, A, are calculated by the different expressions given by the authors, which are as following:

Manglik and Bergles [19]

Free flow area is calculated by A_c= sh. In evaluating the heat transfer area A; the blunt fin edges, both vertical and lateral, have been included in the channel surface area. Heat transfer area is calculated by the expression:

$$A = 2(sl + hl + ht) + ts$$

The hydraulic diameter by manglik and bergles is given by:

$$D_h = \frac{4shl}{2(sl + hl + ht) + ts}$$

Joshi and Webb [14],

Joshi and webb gave the formulas for Free flow area and heat transfer area which are as follows,

Free flow area, A_c = (s-t) h

Heat transfer area, A = 2(sl+ hl+ ht)

Therefore hydraulic diameter is given by the formula:

$$D_h = \frac{4(s-t)h}{2(sl + hl + ht)}$$

Wieting [15] and Kays and London [2],

Wieting and Kays and London gave the formula of hydraulic diameter for a rectangular channel of sh cross section, which is as follows:

$$D_h = \frac{4sh}{2(s + h)}$$

1.6 FLOW ARRANGEMENT

In the heat exchangers, the general flow arrangements of the fluids which occurs are as follows:

- Parallel flow
- Counter flow
- Crossflow

The selection of a particular flow arrangement is dependent upon the required exchanger effectiveness, temperature levels, fluid flow paths, allowable thermal stresses, packaging envelope and other design criteria.

1.6.1 PARALLEL FLOW EXCHANGER

In the parallel flow exchanger, both the fluid streams enter at the same end, flow parallel to each other in the same direction, and leave at the other end. This flow arrangement has the lowest exchanger effectiveness among the single pass exchangers if it is used for the same flow rates, capacity rate ratio, and surface area. The high thermal stresses in the exchanger wall at inlet may occur due to the existence of large temperature differences at the inlet end. Advantages of the Parallel flows:

(a) Rapid heating can be done in the parallel flow. Because of the change in viscosity, the requirements of pumping power gets reduced via the heat exchanger.

(b) It is used where the more moderate mean metal temperatures of the tube walls are required, and

The parallel flow is not used widely, but it is adapted because of the following reasons:

1. The parallel flow is used when there is a possibility that the warmer fluid may reach its freezing point.
2. The desired exchanger effectiveness is low for a balanced exchanger
3. Early initiation of nucleate boiling for boiling applications.
4. The application allows piping only suited to parallel flow.
5. Temperature-sensitive fluids are less likely to be "thermally damaged" in a parallel flow heat exchanger.
6. Certain types of fouling such as chemical reaction fouling, scaling, corrosion fouling, and freezing fouling are sensitive to temperature.

1.6.2 COUNTER FLOW EXCHANGER

In the counter flow exchanger, the two fluids flow parallel to each other but in opposite directions. The counter flow is the most efficient among the all flow arrangements for single-pass arrangements under the same parameters. The minimum thermal stresses are produce by this in the wall for equivalent performance compared to other flow arrangements because the temperature difference across the exchanger wall at a given cross section is the lowest. The counter flow arrangement cannot be achieved easily in many types of heat exchanger because of the manufacturing difficulties associated with the separation of the fluids at each end and at the inlet or outlet header because of the design complexity. If the desired heat exchanger effectiveness is generally more than 80%, then the counter flow unit is preferred in place of the cross and the parallel flow heat exchangers.

1.6.3 CROSSFLOW EXCHANGER

In the cross flow exchanger the two fluids flow normal to each other. Many configuration can be made in the cross flow arrangement:

- Both fluids mixed
- Both fluids unmixed
- First mixed fluid and second unmixed fluid

The thermal effectiveness for the crossflow exchanger lies between those of the parallel flow and counter flow arrangements. This is the most common flow arrangement used for extended surface heat exchangers because it greatly simplifies the header design. If the desired heat exchanger effectiveness is generally more than 80%, then a counter flow unit is preferred. When fluids passes through the individual flow passage without getting mixed then it is said to be unmixed. A tube-fin exchanger with continuous fins and a plate-fin exchanger wherein the two fluids flow in separate passages represent the unmixed–unmixed case. The both fluid mixed case is practically a less important case and represents a limiting case of some multi-pass shell and tube exchangers.

CHAPTER 2. LITERATURE REVIEW

2.1 LITERATURE SURVEY

There are only few research work available on the plate fin heat exchanger, proposing its use in the liquefaction application. [1] Barron R.F study on the cryogenic systems state that the Plate fin heat exchangers used in cryogenic applications must have very high effectiveness (0.85 or more), otherwise the liquefiers won't be able to produce the liquid or one can say if the effectiveness falls below design value the heat exchanger will cease to produce liquid. For the proper functioning of the heat exchanger, the focus is to be made on the design and quality construction of heat exchangers. The two factors are important in the proper designing of the heat exchanger are- heat transfer coefficient and the flow resistance which are expressed in non- dimensional form as Colburn factor (function of Reynold no. and other geometrical parameters) and the friction factor(function of Reynold no. and other geometrical parameters). These factors can be determined by numerical modelling of the flow field through CFD. The numerical solution by the computer is not possible because the models are usually based on certain simplifying assumptions. [2] Hosung Le provided the method for thermal design of heat sinks, thermoelectric, heat pipes, compact heat exchanger and solar cells. [3] E U Schlunder design book of heat exchangers presents the methods to design and numerical calculations for many heat exchanger. [4] Ramesh K Shah and Dušan P. Sekulić provided the most comprehensive data & information on the subject, particularly on compact plate fin heat exchangers. [5] Kays and London in compact heat exchanger book provided an excellent introduction to the analysis of plate fin heat exchangers, and a vast database of the flow friction characteristics and heat transfer of many fin geometries. [6] Thomas M Flynn emphasized on the cryogenic systems, fluids, transportation, liquefaction cycles in his book named as cryogenic engineering.

Journals on heat transfer and thermal engineering devote a sizable portion of their content to research findings on compact fin heat exchangers. [7] Yang Yujie carried out a study on the plate fin heat exchanger with the offset fin strip and evaluated the enhancement in the performance of the heat exchanger by introducing a new parameter called relative entropy generation distribution factor. Two major journals: Heat Transfer Engineering [8] and International Journal of Heat exchangers [9] are almost exclusively dedicated to the subject of heat exchangers.

Kays and London at Stanford University in late of 1940's carried out the, one of the earliest and most comprehensive experimental works on compact heat exchangers published their report in 1948 providing details of the complete methodology and details of the experimental set up. After that so many researchers used the same experimental technique for the experimental determination of non-dimensional heat transfer coefficient and friction factor as the reference. Based on the prior experiments, the kays and London generated the so many afterwards. The heat transfer and flow friction characteristic of a surface are the functions of geometrical parameters such as fin height, fin spacing, fin thickness etc. so the analytical determination of the non-dimensional heat transfer and pressure drop characteristics is difficult because the each fin is needed to be analyzed separately.

2.2 HISTORY OF DEVELOPMENT

The development of the compact heat exchangers were started in the in the automobile and aircraft industries. Second development took place when using the dip soldered copper, the secondary surface plate and corrugation construction became established for aero engine radiators. [11] Mori et al and Cowell at al [12] reviewed the development of compact heat exchangers in the automobile and air conditioning industries.

The introduction of the aluminium dip-brazing process made it possible to manufacture brazed aluminium heat exchangers, fabricated from plate pairs, to employee as aircraft engine intercoolers. The Trane Company in USA employed the dip brazing commercially first time during World War II and the first industrial size exchangers were manufactured in 1949. Because of the properties of aluminium of no loss of strength and ductility at low temperatures made it extremely suitable for cryogenic applications. Then the development of machines capable of producing very precise corrugated fins with varying height and spacing lead to mass production and opened the new areas of researches. Continued reduction in weight, increase of surface area density, enhanced reliability and flexibility that it offers to the manufacturer have made the plate fin heat exchanger indispensable in the industrial applications of gas to gas heat exchange. The plate fin heat exchanger was originally conceived by an Italian mechanic, Paolo Fruncillo. A plate-fin heat exchanger is made of layers of corrugated sheets separated by flat metal plates, typically aluminium, to create a series of stack of fins. In the plate fin heat exchanger, starting from the top, the cold fluid flows in the upper layer and the hot fluid flows through the adjacent layer and these are enclosed by side bars at the edges. Heat is transferred through the fin interface and separator plate from one stream or fluid to the adjacent fluid. More recently the application of plate fin heat exchangers has been extended to boiling and condensation duties. The features of the plate fin heat exchanger such as compact shape, low weight, and design flexibility available with it led the way for their application on a much wider scale and they replaced the tubular heat exchangers used in cryogenic applications. Then development of large aluminium plate fin heat exchangers and that of tonnage air separation plant supported each other for further growth.

Today, brazed aluminium plate fin exchangers are being designed and manufactured by several reputed companies around the globe, among them the name which comes up is Linde. The other companies are the Chart Heat Exchanger USA, Kobe Steel Ltd Japan, Nordon Cryogenie France and Sumitomo Precision Products Co. Ltd., Japan. Several specialized laboratories such as the most notable are the Heat Transfer and Fluid Flow Services in England and Heat Transfer Research Inc. in USA also made significant contribution to the research on plate fin heat exchangers. These organizations, supported by industry and institutions from around the world continue to produce most advanced and authentic information on the subject of plate fin heat exchangers.

2.3 THEORETICAL AND PRACTICAL STUDIES

2.3.1 ANALYTICAL AND NUMERICAL STUDIES

The performance of the plate fin heat exchanger are greatly dependent on the parameter like fin spacing(s), fin height (h), fin thickness (t), offset strip length (l), wavelength (λ), hydraulic diameter (D_h) etc. unlike in the other heat exchanger whose performance depends upon the hydraulic diameter (D_h) mainly. The experimental process of fabrication of heat exchanger cores and conducting experiments with the reasonable ranges of all the geometric variables and Reynold number is very expensive and time consuming. While the parametric study of the heat exchanger through numerical simulation is very easy, takes less time and cost and through these the acceptable correlations for the use of study can be derived. As our computing abilities are increasing and many computational tools are getting developed the numerical prediction of j and f factors are now feasible by solving the continuity, momentum and energy equations.

[20] Shah et al presented a comprehensive review of numerical analysis of some of the important fin geometries employed in compact heat exchangers. [21] Jacobi and Shah did a heat transfer study over the compact heat exchanger to enhance the heat transfer mechanism. They used the so many plate fin heat exchanger geometries in their study and also discussed on the physics of the flow process. [22] Patankar did a research and provided a comprehensive summary of CFD equations relevant to compact heat exchanger passages and also gave the techniques employed for their solution. [23] Levent Bilir et al analysed the effect of three different types of 22 vortex generators on the performance of fin tube heat exchangers using the CFD "FLUENT " module. Their observation were that the three vortex generators when placed suitably will increase the heat transfer with moderate increase in pressure drop.

There have been so many research carried out over the louvered and offset strip fins the reason is the complexity of these to fin and their high heat transfer capacity which attracted the attention of researchers more than other geometries. A brief review of research done over the offset strip fins is given below:

Offset strip fin surfaces:

[24] Patankar, Sparrow and coworkers used the computational fluid dynamics technique to solve or predict the heat transfer coefficient j and f friction factor data in offset strip fin heat exchangers. They were the first to do such thing. The heat transfer and flow friction characteristic results by R.K Shah and London in their experiment over offset strip fins were used by the [25] Patankar and Prakash extending their work further and compared their numerical results for a two dimensional heat transfer matrix having offset strip fins, for the comparison purpose. The conclusion was that the results for the f factors were in the favour, but the predicted j factors were about twice as large as the experimental data.

[26] Suzuki et al carried out a numerical and experimental studies on a two-dimensional model of an offset-strip-fin type compact heat exchanger used at low Reynolds number. He studied the thermal performance of a staggered array of vertical flat plates at low Reynolds number by using a different numerical

approach by solving elliptic differential equations of momentum and energy. He validated his outcomes by conducting the experiments on a two dimensional system, followed by those on a practical offset strip fin heat exchanger and the experimental results were found within the range of Reynolds number Re<800. [27] Zhang et al did a computational study of flow and heat transfer in Parallel-Plate Fin Heat Exchangers on the CM-5 and concluded the effects of flow unsteadiness and three-Dimensionality. He carried out the study by solving the unsteady Navier-Stokes and energy equations and concluded that the inclusion of flow unsteadiness plays an important role in accurate prediction of j and f factors.

2.3.2 EXPERIMENTAL STUDIES

Through the analytical method or numerical method, the calculation of non-dimensional heat transfer coefficient, j and friction factor, f the accurate results is still not achieved because of the limitation of the computing tools and their configuration. For the critical designs complex systems like cryogenic systems, the calculation of heat transfer coefficient and friction factor is only can be done through the experiments. For the less complex designs, the empirical correlations for the j and f factors can be used successfully for calculations. The experimental setups are even used for the verification of the data obtained through the computer setup or the heat exchanger meets the prescribed thermal performance and pressure drop requirements. These are also used in the analysis of the various causes of degradation and malfunctioning.

The experimental methods used are as follows:

1. Steady state method in which heat is transferred from one fluid to another through a separating wall (recuperative heat exchanger).

2. Transient method in which heat is exchanged with a solid matrix (regenerative heat exchanger). This method is further classified as,

- Single blow method, and
- Periodic test methods.

2.3.2.1 STEADY STATE METHODS

A) Kays and London

The initial step or comprehensive work on the compact heat exchangers was carried out by Kays and London at Stanford University in late 1940. The published their first report on compact heat exchanger in 1948, which is considered as the most reliable and authoritative source for heat transfer j and f friction factors. The setup used by them in the Stanford university is known for the accurate calculation of the heat transfer and flow friction characteristic and many correlation generated by the authors between these two are the reworking of their work. In the following lines below, the methodology that kays and London used and the setup the developed for the experiments is described:

In the steady state method, a cross flow type heat exchanger is usually employed as the test exchanger. A channel of the cross flow heat exchanger is made of surface and the fluid flowing over this test surface is the one which is used in the service. Air is used in the testing medium conventionally, as the plate fin heat exchangers are mostly used in the gas to gas application and the gases used in the process are mostly have comparable properties. The fluid which is flowing over the second channel must be able to provide high heat transfer rate and low pressure drop to improve accuracy. The condensing steam, hot water and oil all are have high heat transfer coefficient.

The effectiveness of the heat exchanger in the steady state experiment is calculated through the data of temperatures and mass flow rates in the two sides obtained by the measurement. ε - Ntu is determined by applying an appropriate relation to the cross flow arrangement and hence the overall heat transfer coefficient (U), the inverse of which is related to the resistances of individual sides and that of the separating wall. They assumed that the fouling resistances is negligible; the overall thermal resistance (1/UA) is expressed by the following relation:

$$\frac{1}{UA} = \frac{1}{(\eta_o hA)_{unknown}} + \frac{1}{(\eta_o hA)_{known}} + R_{wall}$$

Where,

η_o = Overall efficiency

η_o= 1- $(\frac{a_f}{a_s}) \times (1 - \eta_f)$

$\frac{a_f}{a_s}$ = the ratio of fins to the total surface area

η_f = Fin efficiency

$$\eta_f = \frac{\tanh(Ml)}{Ml}$$

Where, $$M = \sqrt{\frac{2\times h}{K_f t}}$$

The j factor by its definition is calculated by,

$$j = \frac{h}{G\, c_p}\,(\mathrm{Pr})^{\frac{2}{3}}$$

The plate-fin heat exchangers are mostly used for gas-to gas heat exchange and the important factor in the design of plate fin heat exchanger is pressure drop for each stream. The overall pressure drop through the plate fin heat exchanger include the following:

(1) The pressure drop at the inlet, as the fluid leaves the inlet header and enters the heat exchanger core,

(2) The frictional pressure drop in the heat exchanger core,

(3) The pressure drop or a pressure rise at the outlet, as the fluid leaves the core and enters the outlet header, and

(4) The momentum pressure drop or rise because of the change in the velocity in the heat exchanger core which results from the change in the density of the fluid.

The Fanning friction factor f is given by,

$$f = \frac{\rho_m D_h}{4L}\left[\frac{2\,\Delta P}{G^2} - \frac{1}{\rho_{inlet}}(K_c - 1 - \sigma^2) - \frac{1}{\rho_{outlet}}\left\{K_e + 1 + \sigma^2\left(1 + 4\,\frac{f_d L_d}{D_d}\right)\right\}\right]$$

Where,

K_c = Contraction coefficient

K_e = Expansion coefficient

Following points were be observed from the experimental set up:

- The heat transfer coefficient is very high in condensation heat transfer as the steam is used as the heat transfer medium in the second channel. The magnitude of the wall resistance and the thermal resistance of the second channel are minimized as the separating walls are made thin and thus the measurement accuracy of hA increases. A thermal boundary condition of a uniform wall temperature with zero heat capacity rate (Cr) is given by the flow of steam over other channel. In the lateral direction the longitudinal heat conduction is normally negligible compared to the high rate of heat transfer.

The number of transfer units can be calculated from the below relation:

$$N_{tu} = \ln\frac{1}{(1-\epsilon)} = \ln\frac{T_s - T_{inlet}}{T_s - T_{outlet}}$$

Where,

T_s = Saturation temperature of condensing fluid at its inlet condition

- Less accurate measurement of heat transfer coefficient and j factor is taken with heat exchangers of high N_{tu}, as the thermodynamic limitation restricts the change of outlet fluid temperatures, making them less sensitive to changes in heat transfer coefficient and making the j factor measurement. Therefore N_{tu} of the test core in the steady state experiment is restricted between 1.0 and 3.0.
- The Fanning friction factor, f can be measured simply, as only the measurement of fluid flow rate, inlet temperature, pressure and pressure drop across the core is required. The pressure drop across the core is calculated by deducting the loss of pressure due to flow through elbows and headers from the measured pressure drop.

2.3.2.2 OTHER EXPERIMENTAL WORKS

London and Shah extended the work by Kays and London, London and Shah [38] have reported measurement on eight high performance offset strip fin geometry surfaces. To determine heat transfer in both side of a heat exchanger, Shah has recently suggested a "modified Wilson Plot technique". An experiments were conducted by Davenport to study the effect of louver pitch with all other geometrical properties held constant, on eight louvered fin surfaces. The heat transfer coefficient was determined by the Dittus Boelter equation on this side. The friction factor was derived from the relation:

$$\Delta P = \frac{\rho_{inlet}\, V^2}{2}\,[K_e + K_c + f(\frac{A}{A_{ff}})]$$

5% experimental error in Stanton number and 12% experimental error in friction factor have been found and mainly to flow measurement.

Sunden and Svantesson used the data obtained from the experiment of Davenport [40] but their method differed in the way, when they used a single channel of width 80 mm with fins of height 12.5 mm. Heat was transferred from a constant oil bath (maintained at $600^o c$) to the single channel heat exchanger. The heat transfer coefficient of oil side was calculated by using the Dittus Boelter equation. 15% experimental error in Stanton number and 6% experimental error in friction factor were detected.

To study the performance of fin density on heat transfer and pressure drop Wang et al carried out heat transfer and flow friction experiments. Lozza et al also carried out the steady state experiments on tube heat exchangers having different fin geometries fin and used air and hot water as working fluids.

Ghosh at I.I.T. Kharagpur conducted the experiments on six offsets and 3 wavy fins using the Kays and London to generate the correlation for laminar and turbulent zones by combining the results with the numerical results obtained in the same laboratory by Maiti and Sarangi. He also used the regression method to obtain some of the constants over the numerically computed results and obtained the rest of the constant from the experimental data.

To determine the local heat transfer coefficients for plate heat exchangers S. Freund and S. Kabelac used the Temperature oscillation IR thermography (TOIRT) method. In this method, first the temperature reading were measured on outer surface of a heat transferring wall with an IR camera and then temperature oscillations were generated by radiant heating.

2.3.2.2 TRANSIENT TECHNIQUE

To characterise the heat transfer surfaces, there is an alternative method available which is known as single blow transient method. The method is used for the packed bed regenerator and matrix type high N_{tu} heat exchanger surfaces, to calculate the average heat transfer coefficient. A compact heat exchanger matrix, or a packed bed, is allowed to come in equilibrium with the process fluid temperature first. Then the other cooler fluid flow through the matrix and the heat exchange takes place with the matrix. To switch from one fluid stream to another flowing through the matrix a 3 way valve is used.

Until the core reaches a uniform temperature displayed by a negligible difference between the temperatures of air stream at inlet and exit the heating continuous. Then to generate the step change electricity supply is switched off instantly. The outlet temperature of the fluid is recorded up to new equilibrated temperature during the cooling period. Then the comparison is made between the measured temperature response and exit fluid temperature history, which was derived from a mathematical model of the system. It is possible to determine the heat transfer coefficient from the parameters of the mathematical model and the operating condition.

Many computational methods are available to analyse the measured data for the determination of N_{tu}, which are as following:

(1) The zero intercept method

(2) The maximum slope method,

(3) The direct curve matching method, and

(4) The first moment of area method.

The data reduction procedures are more complex in the transient methods compared to those in the steady state but they are easy to use. For large N_{tu}, heat exchangers the transient methods are used as they cannot be used for those have small N_{tu}, because of the complexity in the data reduction. The steady state methods provide more accurate results for high performance surfaces.

CHAPTER 3. THEORETICAL DESIGN OF HEAT EXCHANGER

3.1 DESIGN OF PLATE FIN HEAT EXCHANGER

The sizing of heat exchanger or design includes the determination or calculation of the dimensions of heat exchanger. The problem can be reduced to the rating problem by assuming or specifying the dimensions of the heat exchanger and then approaching to the final dimensions by doing some calculations. The pressure drop is a function of flow velocity and as the Reynold number increases the heat transfer coefficient increases, so because of the increase in the Reynold number the pressure drop increases too. For an effective and correct design of a plate fin heat exchanger, the non-dimensional heat transfer coefficient (j) and friction factor (f) must be predicted accurately and wisely. Since the development of the heat exchanger so many correlation for the heat transfer coefficient and friction factor has already been derived and it became very difficult for engineer or a designer to select the best and effective correlation for the calculation. In this work the plate fin heat exchanger is designed by using the reference of [2] Hosung Leeauth's book, then programmed and simulated in Matlab and Ansys respectively. The results of the performance of heat exchanger design are validated by the effectiveness and the results obtained from the analysis.

This chapter describes the design procedure for the plate fin heat exchanger with offset strip fins. Irreversibility's in the heat exchanger performance may occur due to various factors such as flow maldistribution at the headers, longitudinal heat conduction, and heat loss to the surroundings, manufacturing irregularities etc. But in this case we assumed that the heat is getting transferred only through the conduction by the fins in contact to the separating plate and through the convection from fluid to fin and vice versa. Assumption has been made that there is no heat is getting lost to the surroundings. The entrance of the plate fin heat exchanger consist narrow passages. The pressure drop in the gas to gas heat exchanger is high as the gases passes through the heat exchanger. The header of the heat exchanger is responsible for the uniform distribution of the fluid through the entrance of the channels, so attention should be made that the cross sectional area should be more of the header passages than the tubes diameter. At inlet and outlet, both the fluids should have high enough pressure so that the pressure drop throughout the heat exchanger should be less. The heat transfer coefficient, j and flow friction, are the main function of the surface geometry and the heat transfer is dependent on the increase in the flow friction.

Many study has revealed that the ratio of $j/f \leq 0.25$ for strip fin surface. The justification can be given as along with each interruption the flow develops in such surface. In the absence of a form drag, on the basis of the Reynolds analogy for fully developed turbulent flow over a flat plate, prandtl number $Pr \approx 1$. The skin friction in developing laminar flow for such an interrupted surfaces and the form drag contributes same in terms of magnitude, j/f will be 0.25 approximately. If the $j/f > 0.3$ then the calculations are questionable.

3.2 HYDRAULIC DESIGN

3.2.1 GEOMETRIC CHARACTERISTICS

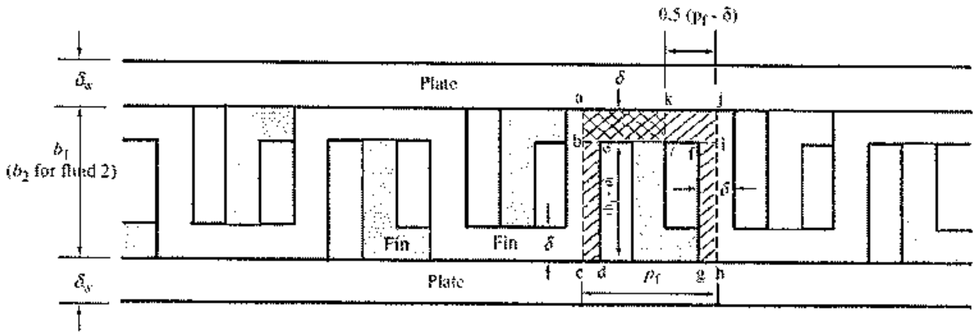

Figure 10. Offset Strip Fin ; source [2]

The offset strip fin used in Plate Fin heat exchanger for a single pass counter flow are shown in the figure 10. The offset rectangular fins of plate fin heat exchanger are made of Aluminium alloy Al 3003 because of its high thermal conductivity and other properties. While analysing the heat exchanger the total heat transfer area for each fluid have to be defined. The total area is the combination of the primary area and the fin area. The primary area consists of the plate area except the fin base area, multi-passage side walls, and multi-passage front and back walls. The following dimensions of the fins are used in table 1:

Table I. Basic Fin geometry used in the PFHE

Fin Geometry	Value
Fin Density, f	700 fins per meter
Fin offset length, λ	9 mm
Fin Height, h	7 mm
Fin Thickness, δ	0.1 mm

Other geometrical characteristics of the fin:

1) Pitch of the Fin, (p_{f1}) $= \frac{1}{\text{Fin density}}$

$= \frac{1}{700}$

$=$ 1.4286 mm

2) Normal Fin Spacing, (s) $= \frac{1-(P_f * t)}{P_f} = p_f - \delta = \frac{1}{\text{Fin density}} - t$

$=$ 1.4286 – 0.1

$=$ 1.3286 mm

3) Fin Length, (L_f) $= \frac{b}{2} - \delta$

$= \frac{7.1}{2} - 0.1$

$=$ 3.45 mm

4) Distance Between plates or plate spacing, (b) = h + t

$=$ 7 + 0.1

$=$ 7.1 mm

5) The numbers of passages for the hot fluid (Methane) side and the cold fluid (Propane) side are Np and Np +1 to minimize the heat loss to the ambient. While designing the heat exchanger the top and bottom passages are designated to be cold fluid. The number of passages are calculated from the expression for L3 as,

$$N_P = \frac{L_3 - b_2 - 2\delta_w}{b_1 + b_2 + 2\delta_w}$$

Where, N_P = the number of passages counts; the number based on one flow passage between two plates, not all the individual channels between the plates.

$N_P = 47$ ...for hot fluid (Methane)

$N_P + 1 = 48$ ...for Cold fluid (Propane)

6) Total number of fins (n_f),

$n_{f1} = N_P \frac{L_1}{p_{f1}}$ ...for hot fluid (Methane)

$n_{f1} = 9812$

$$n_{f2} = (N_p + 1)\frac{L_1}{p_{f1}} \quad \text{...for Cold fluid (Propane)}$$

n_{f2}= 10022

7) Primary Area (A_p),

$$A_{p1} = 2L_1L_2N_p - 2\delta L_2 n_{f1} + 2b_1L_2N_p + 2(b_2 + 2\delta_w)L_1(N_p + 1)$$

Where, Total Plate Area = $2L_1L_2N_p$ = 45.97747 m^2

Fin base Area = $2\delta L_2 n_{f1}$ = 3.2184229 m^2

Passage side Wall Area = $2b_1L_2N_p$ = 1.088133457 m^2

Passage front and back wall area = $2(b_2 + 2\delta_w)L_1(N_p + 1)$ = 0.209023333 m^2

So, A_{P1} = 44.05620389 m^2 ...for hot fluid side

$$A_{p2} = 2L_1L_2(N_p + 1) - 2\delta L_2 n_{f2} + 2b_2L_2(N_p + 1) + 2(b_1 + 2\delta_w)L_1N_p$$

Where, Total Plate Area = $2L_1L_2(N_p + 1)$ = 46.96153 m^2

Fin base Area = $2\delta L_2 n_{f2}$ = 3.2873071 m^2

Passage side Wall Area = $2b_2L_2(N_p + 1)$ = 1.111422877 m^2

Passage front and back wall area = $2(b_1 + 2\delta_w)L_1N_p$ = 0.204643333 m^2

So, A_{P2} = 44.99028911 m^2 ...for cold fluid side

8) Number of offset strip fins (n_{off}),

$$n_{off1} \ or \ n_{off2} = \frac{L_2}{\lambda_1}$$

= 182

9) Total fin area(A_f),

$$A_{f1} = 2(b_1 - \delta)L_2 n_{f1} + 2(b_1 - \delta)\delta n_{off1} n_{f1} + (p_{f1} - \delta)\delta(n_{off1} - 1)n_{f1} + 2\delta p_{f1} n_{f1}$$

Where, Fin Surface Area = $2(b_1 - \delta)L_2 n_{f1}$ = 225.289603 m^2

Offset strip edge Area = $2(b_1 - \delta)\delta n_{off1} n_{f1}$ = 0.236246712 m^2

Internal offset strip edge Area = $(p_{f1} - \delta)\delta(n_{off1} - 1)n_{f1}$ = 0.002803333 m^2

Offset strip edge Area = $2\delta p_{f1} n_{f1}$ = 2.503217811 m^2

So, A_{f1} = 228.0318709 m^2 ...for hot fluid side

$$A_{f2} = 2(b_2 - \delta)L_2 n_{f2} + 2(b_2 - \delta)\delta n_{off2} n_{f2} + (p_{f2} - \delta)\delta(n_{off2} - 1)n_{f2} + 2\delta p_{f2} n_{f2}$$

A_{f2} = 232.9124579 m^2 ...for cold fluid side

10) Total heat transfer area (A_t),

$$A_{t1} = A_{p1} + A_{f1}$$

A_{t1}= 44.05620389 + 228.0318709

= 272.0880747 m^2 ...for hot fluid side

A_{t1}= 44.99028911 + 232.9124579

= 277.902747 m^2 ...for cold fluid side

11) Free-flow area (A_c),

$$A_{c1} = (b_1 - \delta)(p_{f1} - \delta)n_{f1}$$

= 0.0912485 m^2 ...for hot fluid side

$$A_{c2} = (b_2 - \delta)(p_{f2} - \delta)n_{f2}$$

= 0.0932015 m^2 ...for cold fluid side

12) Frontal area (A_{fr}),

$$A_{fr1} = L_1 L_3$$

= 0.20403 m^2 ...for hot fluid side

$$A_{fr2} = L_1 L_3$$

= 0.20403 m^2 ...for cold fluid side

13) Hydraulic diameter (D_h),

$$D_{h1} = \frac{4A_{c1}L_2}{A_{t1}}$$

= 0.002200121 m ...for hot fluid side

$$D_{h2} = \frac{4A_{c2}L_2}{A_{t2}}$$

= 0.002200191 m ...for cold fluid side

We preferred the general hydraulic diameter over the Manglik and Bergles's expression for the correlations, the reason is to reduce loss of any information of the dimensions (L1, L2 and L3) with the approximation of Equation for modelling purposes. The result of the general hydraulic diameter expression and Manglik and Bergles's expression are found to be almost similar.

$$D_{h\_MB} = \frac{4(p_{f1} - \delta)b_1\lambda_1}{2[(p_{f1} - \delta)\lambda_1 + b_1\lambda_1 + b_1\delta] + (p_{f1} - \delta)\delta}$$

D_{h_MB} = 0.00221562 m

14) Porosity (σ),

$$\sigma_1 = \frac{A_{c1}}{A_{fr1}}$$

σ_1 = 0.447230799 ...for hot fluid side

$$\sigma_2 = \frac{A_{c2}}{A_{fr2}}$$

σ_2 = 0.456802921 ...for cold fluid side

15) Volume of the exchanger (V_p),

$$V_{p1} = b_1 L_1 L_2 N_p$$

V_{p1} = 0.163220019 m^3 ...for hot fluid side

$$V_{p2} = b_1 L_1 L_2 (N_p + 1)$$

V_{p2} = 0.166713432 m^3 ...for cold fluid side

16) Surface area density (β),

$$\beta_1 = \frac{A_{t1}}{V_{p1}}$$

β_1 = 1667.001862 ...for hot fluid side

$$\beta_2 = \frac{A_{t2}}{V_{p2}}$$

β_2 = 1666.948754 ...for hot fluid side

17) Mass velocities (G),

$$G_1 = \frac{\dot{m}_1}{A_{c1}}$$

G_1 = 0.126840441 kg/m^2-sec ...for hot fluid

$$G_2 = \frac{\dot{m}_2}{A_{c2}}$$

G_2 = 0.111765369 kg/m^2-sec ...for cold fluid

18) Reynold Number (Re),

$$Re_1 = \frac{G_1 D_{h1}}{\mu_1}$$

Re_1 = 31.52983973 ...for hot fluid

$$Re_2 = \frac{G_2 D_{h2}}{\mu_2}$$

Re_2 = 1.240629419 ...for cold fluid

19) Colburn Factor (j_1),

$$j_{1a} = 0.6522 Re_1^{-0.5403} \left(\frac{p_{f1} - \delta}{b_1 - \delta}\right)^{-0.1541} \left(\frac{\delta}{\lambda_1}\right)^{0.1499} \left(\frac{\delta}{p_{f1} - \delta}\right)^{-0.0678}$$

j_{1a} = 0.079261441

$$j_{1b} = \left[1 + 5.269\,(10)^{-5}\, Re_1^{1.34} \left(\frac{p_{f1} - \delta}{b_1 - \delta}\right)^{0.504} \left(\frac{\delta}{\lambda_1}\right)^{0.456} \left(\frac{\delta}{p_{f1} - \delta}\right)^{-1.055}\right]^{0.1}$$

j_{1b} = 1.000456474

$j_1 = j_{1a}\, j_{1b}$

j_1 = 0.079297622 ...for hot fluid

$$j_{2a} = 0.6522 Re_2^{-0.5403} \left(\frac{p_{f2} - \delta}{b_2 - \delta}\right)^{-0.1541} \left(\frac{\delta}{\lambda_2}\right)^{0.1499} \left(\frac{\delta}{p_{f2} - \delta}\right)^{-0.0678}$$

j_{2a} = 0.455225435

$$j_{2b} = \left[1 + 5.269\,(10)^{-5}\, Re_2^{1.34} \left(\frac{p_{f2} - \delta}{b_2 - \delta}\right)^{0.504} \left(\frac{\delta}{\lambda_2}\right)^{0.456} \left(\frac{\delta}{p_{f2} - \delta}\right)^{-1.055}\right]^{0.1}$$

j_{2b} = 1.000005991

$j_2 = j_{2a}\, j_{2b}$

j_2 = 0.455228162 ...for cold fluid

20) Friction Factor,

$$f_{1a} = 9.6243\, Re_1^{-0.7422} \left(\frac{p_{f1} - \delta}{b_1 - \delta}\right)^{-0.1856} \left(\frac{\delta}{\lambda_1}\right)^{0.3053} \left(\frac{\delta}{p_{f1} - \delta}\right)^{-0.2659}$$

f_{1a} = 0.509384383

$$f_{1b} = \left[1 + 7.669\,(10)^{-8}\, Re_1^{4.429} \left(\frac{p_{f1} - \delta}{b_1 - \delta}\right)^{0.92} \left(\frac{\delta}{\lambda_1}\right)^{3.767} \left(\frac{\delta}{p_{f1} - \delta}\right)^{0.236}\right]^{0.1}$$

f_{1b} = 1

$$f_1 = f_{1a}\, f_{1b}$$

f_1 = 0.509384383 ...for hot fluid

$$f_{2a} = 9.6243\ Re_2^{-0.7422} \left(\frac{p_{f2}-\delta}{b_2-\delta}\right)^{-0.1856} \left(\frac{\delta}{\lambda_2}\right)^{0.3053} \left(\frac{\delta}{p_{f2}-\delta}\right)^{-0.2659}$$

f_{2a} = 5.622063801

$$f_{2b} = \left[1 + 7.669\ (10)^{-8}\ Re_2^{4.429} \left(\frac{p_{f2}-\delta}{b_2-\delta}\right)^{0.92} \left(\frac{\delta}{\lambda_2}\right)^{3.767} \left(\frac{\delta}{p_{f2}-\delta}\right)^{0.236}\right]^{0.1}$$

f_{2b} = 1

$$f_2 = f_{2a}\, f_{2b}$$

f_2 = 5.622063801 ...for cold fluid

21) Heat Transfer Coefficient (h),

$$h_1 = \frac{j_1 G_1 c_{p1}}{(Pr_1)^{\frac{2}{3}}}$$

h_1 = 26.30621912 w/m^2k ...for hot fluid

$$h_2 = \frac{j_2 G_2 c_{p2}}{(Pr_2)^{\frac{2}{3}}}$$

h_1 = 50.2040073 w/m^2k ...for cold fluid

22) Single Fin efficiency (η_f),

$$L_{f1} = \frac{b_1}{2} - \delta$$

L_{f1} = 0.00345 m

$$m_1 = \left[\frac{2h_1}{k_{f1}\delta}\left(1+\frac{\delta}{\lambda_1}\right)^1\right]^{0.5}$$

m_1 = 54.36349924

$$\eta_{f1} = \frac{\tanh(m_1 L_{f1})}{m_1 L_{f1}}$$

η_{f1} = 0.988437159 ...for hot fluid side

$$L_{f2} = \frac{b_2}{2} - \delta$$

L_{f2} = 0.00345 m

$$m_2 = \left[\frac{2h_2}{k_{f2}\delta}\left(1+\frac{\delta}{\lambda_2}\right)^1\right]^{0.5}$$

m_2 = 75.10128686

$$\eta_{f2} = \frac{\tanh(m_2 L_{f2})}{m_2 L_{f2}}$$

η_{f2} = 0.978207501 ...for cold fluid side

23) Overall Fin Surface efficiency (η_{o1}),

$$\eta_{o1} = 1 - (1 - \eta_{f1})\,\frac{A_{f1}}{A_{t1}}$$

η_{o1} = 0.990309402 ...for hot fluid side

$$\eta_{o2} = 1 - (1 - \eta_{f2})\,\frac{A_{f2}}{A_{t2}}$$

η_{o2} = 0.981735537 ...for cold fluid side

24) Overall Heat Transfer Coefficient (UA),

$A_w = 2\, L_1 L_2 (N_p + 1)$

A_w = 46.96153 m^2

$$R_w = \frac{\delta_w}{K_w A_w}$$

R_w = 1.183E-08

$$UA_1 = \frac{1}{\eta_{o1} h_1 A_{t1}}$$

UA_1 = 0.000141079 W/K ...for hot fluid

$$UA_2 = \frac{1}{\eta_{o1} h_1 A_{t1}}$$

UA_2 = 7.30086E-05 W/K ...for cold fluid

$$UA = \frac{1}{UA_1 + R_w + UA_2}$$

UA = 4670.734775 W/K

3.3 THERMAL DESIGN

The two basic approaches are available which are used mostly and widely in the thermal design of a heat exchanger:

a) NTU-effectiveness approach and,

b) Log mean temperature difference (LMTD) approach

These two approaches have the same principles but the design variables are arranged differently in the two approaches. The heat exchanger design problems are basically divided into two categories- sizing problems and rating problems. In the designing or sizing problem, where the heat transfer rate and temperature of the all terminal are known and we have to determine the heat transfer surface area (the heat exchanger size), both methods are of the same utility level. While for the rating problem or design situation, where the heat exchanger surface area and the inlet fluid temperatures are known and we have to determine the heat transfer rate, the NTU-effectiveness method has some specific advantages over the Log mean temperature difference (LMTD) approach. The NTU-effectiveness approach is superior to LMTD approach in specific cases because of the following reasons:

1) NTU-effectiveness method is direct calculation method, while in the LMTD approach in the rating problem always involves iteration.

2) In the NTU-effectiveness, the heat exchanger effectiveness gives a direct measure of how closely the heat exchanger approaches the best possible performance in transferring energy, while the value of the LMTD does not give an explicit indication of how good the heat exchanger is or how effective the heat exchanger is in performing its task of energy exchange.

The basic value of the parameters which are used in the design calculation:

For Hot Fluid (Methane):

Hot fluid inlet temperature (T_{h1})	=	300 K
Hot fluid outlet temperature (T_{h2})	=	155 K
Hot fluid mass flow rate ($\dot{m}_h$)	=	0.011574 Kg/sec
Hot fluid inlet pressure (P_{h1})	=	400 Kpa
Inlet density (ρ_{hi})	=	2.5901 Kg/m^3
Outlet density (ρ_{ho})	=	5.2718 Kg/m^3

Mean density ($\rho_{hm} = \frac{2\,\rho_{hi}\,\rho_{ho}}{\rho_{hi} + \rho_{ho}}$) = 3.473585057 Kg/$m^3$

For Cold Fluid (Propane):

Cold fluid (Propane) inlet temperature (T_{c1}) = 90 K

Cold fluid mass flow rate ($\dot{m}_c$) = 0.0104167 Kg/sec

Cold fluid in the pressure (P_{c1}) = 400 KPa

Specific heat of cold fluid (At Average temperature, 230.9 K) (c_{pc})=2250.7 J/Kg-K

Inlet density (ρ_{ci}) = 728.51 Kg/m^3

Outlet density (ρ_{co}) = 565.41 Kg/m^3

Mean density ($\rho_{cm} = \frac{2\,\rho_{ci}\,\rho_{co}}{\rho_{ci} + \rho_{co}}$) = 636.6805353 Kg/$m^3$

1) The average temperature for hot methane stream,

$$T_{h,avg} = \frac{1}{2}(T_{h1} + T_{h2})$$

$$= 227.5 \text{ K}$$

At this temperature, Specific heat (c_{ph}) = 2158 J/Kg-K

Density (ρ) = 3.4518 Kg/m^3

Thermal conductivity (k) = 0.025484 W/m-K

Viscosity (μ) = 8.8508E-06 Pa-sec

Prandtl Number (Pr) = 0.749490912

2) The average temperature of the cold propane fluid is estimated to be approximately 230.9 K. At this temperature,

Specific heat (c_{pc}) = 2250.7 J/Kg-K

Density (ρ) = 581.43 Kg/m^3

Thermal conductivity (k) = 0.1295 W/m-K

Viscosity (μ) = 0.00019821 Pa-sec

Prandtl Number (Pr) = 3.444874494

3) The cold fluid exit temperature (T_{c2}),

$$T_{c2} = T_{c1} + \frac{\dot{m}_h c_{ph}(T_{h1} - T_{h2})}{\dot{m}_c c_{ph}}$$

$$T_{c2} = 244.4739148 \text{ K}$$

4) Capacity rate,

Hot fluid capacity (C_h) = $\dot{m}_h c_{ph}$ = 24.976692 W/K

Cold fluid capacity $(C_c) = \dot{m}_c c_{pc}$ = 23.44486669 W/K

5) Capacity rate ratio (C_R),

$$C_R = \frac{C_{min}}{C_{max}}$$

$$C_R = 0.938669808$$

6) Number of Transfer Unit (NTU),

$$\text{NTU} = \frac{UA}{C_{min}}$$

$$\text{NTU} = 199.2220658$$

7) Effectiveness (ϵ),

$$\epsilon = 1\text{-exp}\ [\frac{1}{C_R} * NTU^{0.22} * \{\exp(-CR * NTU^{0.78}) - 1\}]$$

$$\epsilon = 0.967110841$$

8) Maximum Heat Transfer Rate (Q),

$$Q = \epsilon\ C_{min}\ (T_{h1} - T_{c1})$$

$$Q = 4761.494797\ \text{W}$$

9) Pressure Drop (ΔP),

a) Here for this case the Reynolds number is assumed to be fully turbulent (Re $=10^7$) because of the frequent boundary layer interruptions due to the offset strip fins.

Friction factor used for circulation $(f_d) = 0.049\ Re^{-0.2}$

$$f_d = 0.001950725$$

b) Velocity distribution coefficient $(K_{d\_square}) = 1 + 1.17\ (K_{d\_tube} - 1)$

$$K_{d\_square} = 1.016038409$$

Where, $K_{d\_tube} = 1.09068\ (4f_d) + 0.05884\ ((4f_d)^{0.5}) + 1$

$$K_{d\_tube} = 1.013708042$$

For Hot Fluid:

Expansion Coefficient $(K_{e_{sqaure\ 1}}) = 1 - 2\sigma_1\ K_{d\_square} + {\sigma_1}^2$

$$K_{e_{sqaure\ 1}} = 0.291208048$$

Jet Contraction Ratio $(C_{c_{tube}\ 1}) = (4.374\ (10^{-4}) * (\exp(6.737({\sigma_1}^{0.5}))) + 0.621$

$$C_{c_{tube\,1}} = 0.660586293$$

Contraction coefficient $(K_{c\_square1})$ = $\frac{1-2\,C_{c_{tube\,1}}+C_{c_{tube\,1}}{}^{2}(2\,K_{d_{square}}-1)}{(C_{c\_tube}1)^{2}}$

$$K_{c_{square\,1}} = 0.296074211$$

So, $\Delta P_h = \frac{G_1{}^2}{2\rho_{hi}}[(1-\sigma_1{}^2+K_{c_{square\,1}}) + 2\,(\frac{\rho_{hi}}{\rho_{ho}}-1) + \frac{4f_1L_2}{D_{h1}}(\frac{\rho_{hi}}{\rho_{hm}}) - (1-\sigma_1{}^2-K_{e_{sqaure\,1}})\frac{\rho_{hi}}{\rho_{ho}}]$

$$\Delta P_h = 3.516991095 \text{ KPa}$$

For Cold Fluid:

Expansion Coefficient $(K_{e_{sqaure\,2}})$ = $1\text{-}2\sigma_2\, K_{d\_square} + \sigma_2{}^2$

$$K_{e_{sqaure\,2}} = 0.280410282$$

Jet Contraction Ratio $(C_{c_{tube}\,2})$ = $(4.374\,(10^{-4})*(\exp\,(6.737(\sigma_2{}^{0.5}))) + 0.621$

$$C_{c_{tube}\,2} = 0.66253109$$

Contraction coefficient $(K_{c\_square2})$ = $\frac{1-2\,C_{c_{tube}\,2}+C_{c_{tube}\,2}{}^{2}(2\,K_{d_{tube}}-1)}{(C_{c_{tube}}2)^{2}}$

$$K_{c_{square\,2}} = 0.291527617$$

So, $\Delta P_c = \frac{G_2{}^2}{2\rho_{ci}}[(1-\sigma_2{}^2+K_{c_{square\,2}}) + 2\,(\frac{\rho_{ci}}{\rho_{co}}-1) + \frac{4f_2L_2}{D_{h2}}(\frac{\rho_{ci}}{\rho_{cm}}) - (1-\sigma_2{}^2-K_{e_{sqaure\,2}})\frac{\rho_{ci}}{\rho_{co}}]$

$$\Delta P_c = 0.164456606 \text{ KPa}$$

CHAPTER 4. ANALYTICAL DESIGN OF HEAT EXCHANGER

4.1 METHODOLOGY

4.1.1 MODELLING

In this counter flow heat exchanger, the layers of fins are arranged in an alternating way, first cold then hot and it continuous like in this arrangement. The geometry has been modelled in the Catia software. To predict or validate the calculations, only a small part of the whole geometry has been designed for the analysis purpose. The model is so designed that a symmetric body can be analysed to predict the result for the whole geometry. The middle passage is used for the hot fluid (methane) and the side passage for the cold fluid. The model of 90 mm length, 2.8572 mm width and 14.4 mm height was created in the Catia as shown in the figure 11.

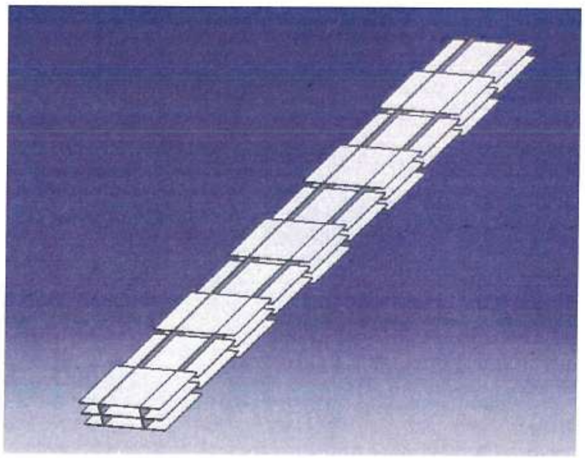

Figure 11. Small symmetric portion of core for Analysis

First the model was imported in the Ansys sub-module of geometry. Four zones has been created there three for fluids: one for hot fluid and two for cold fluid, and one as solid which represents the separating plate and fin itself. The figure 12 shows the zoned geometry.

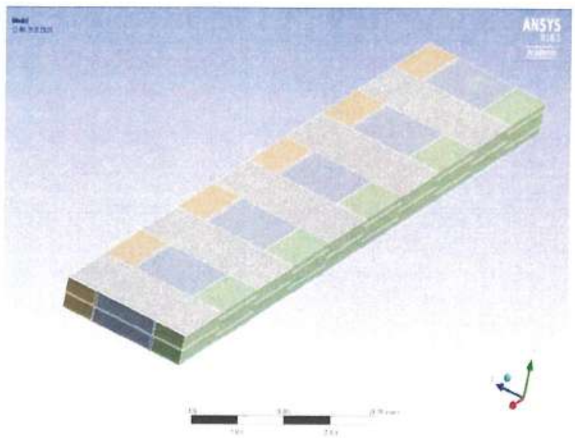

Figure 12. Zoned model for fluids and separating plate

4.1.2 DISCRETIZATION OR MESHING

Discretization is a process which is considered as the first step of the numerical solution. In discretization or meshing, a model is discretized into the finite sized small elements to solve the problem by applying the mathematical equations on them and the approximately accurate results are generated by combining the results of each element as one. The process is used to solve complex geometry easily, with the help of a computer. Many iterations can be performed in this method which is very time consuming and expensive with the human mind. The discretized elements are connected at the nodes. In this work, the model of the plate fin heat exchanger core is discretized into the 7140089 small elements and 1686366 nodes were created. The separating plate and fin zone was subdivided into the 0.1 mm sized elements. Both hot and cold fluids zones were subdivided into the 0.2 mm sized elements. Figure 13 shows the meshed geometry below.

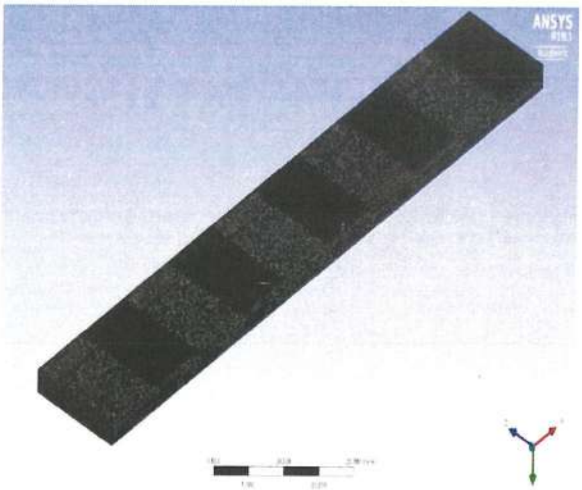

Figure 13. Meshed Model

4.1.3 BOUNDARY CONDITION

The boundary conditions are conditions required at the end or part of the region in which the set of a differential equation are to be solved. Mass flow inlet was used as the inlet boundary condition for both the hot and cold fluids. The figures from 14 to 21 shows the boundary conditions and named sections of the model. Figure 14 shows the hot fluid inlet. The inlet temperature of the methane (hot fluid) is 300 K, pressure has been taken as 400 KPa which is constant throughout the heat exchanger to solve the problem and the mass flow rate for a single fin for the model is 2.34529e-06 Kg/sec. The properties of the hot fluid at the inlet are given in the table II.

Table II. Hot fluid properties at inlet

Input properties	**Value**
Density, ρ (in Kg/m^3)	3.4518
Thermal Conductivity, k (in W/m-K)	0.025484
Viscosity,μ (in Pa-s)	8.8508e-06

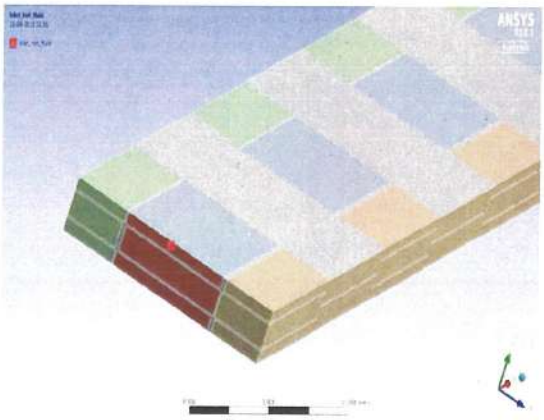

Figure 14. Inlet for hot fluid

Figure 15 shows the cold fluid inlet. Cold fluid properties. The inlet temperature of the propane (cold fluid) is 90 K, pressure has been taken as 400 KPa which is constant throughout the heat exchanger to solve the problem and the mass flow rate for a single fin for a model is 2.06681e-06 Kg/sec. The properties of the cold fluid at the inlet are given in the Table III.

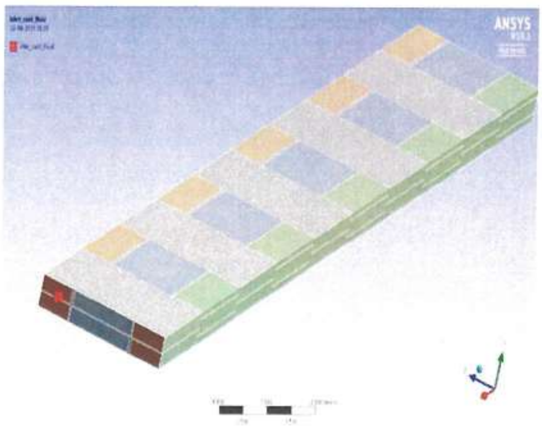

Figure 15. Inlet for cold fluid

Table III. Cold fluid properties at inlet

Input properties	**Value**
Density, ρ (in Kg/m^3)	581.43
Thermal Conductivity, k (in W/m-K)	0.1295
Viscosity,μ (in Pa-s)	0.00019821

Figure 16 and 17 shows the outlet for the cold and hot fluid. For both of the fluids the boundary condition is pressure flow, as the flow velocity and pressure are not known at the outlet.

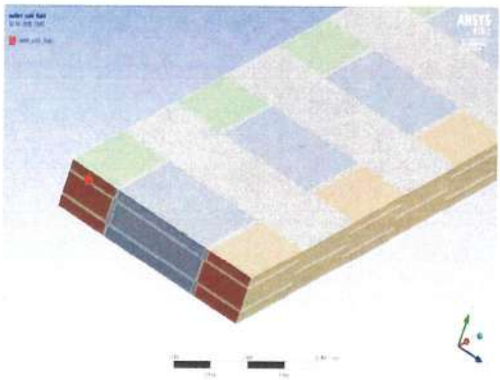

Figure 16. Outlet for hot fluid

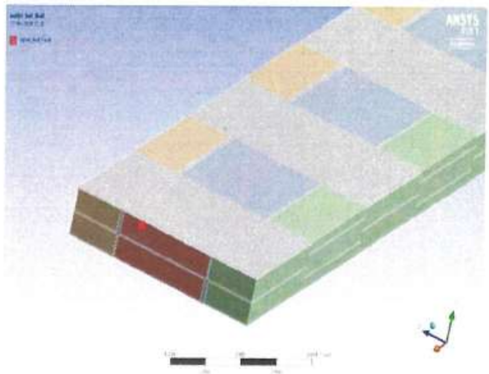

Figure 17. Outlet for cold fluid

Figure 18 to 21 shows the symmetric portion of the fins and separating plate, which are considered for the prediction of the result for the whole geometry.

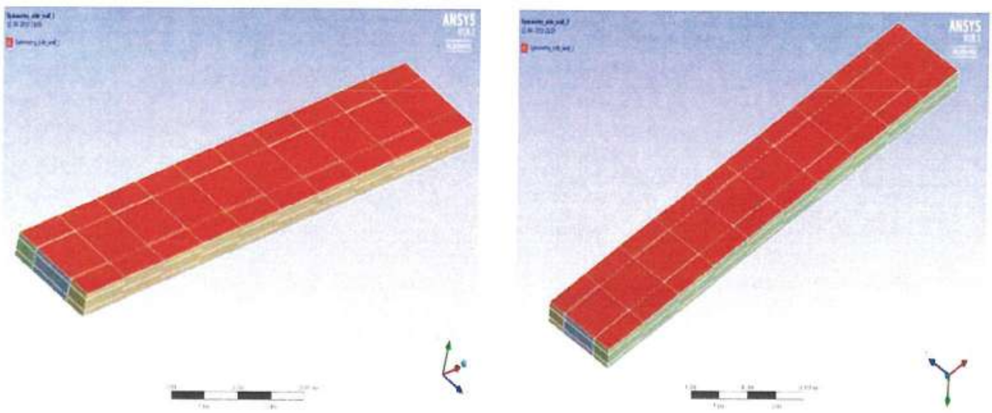

Figure 18. Symmetric Side Wall 1 & 2

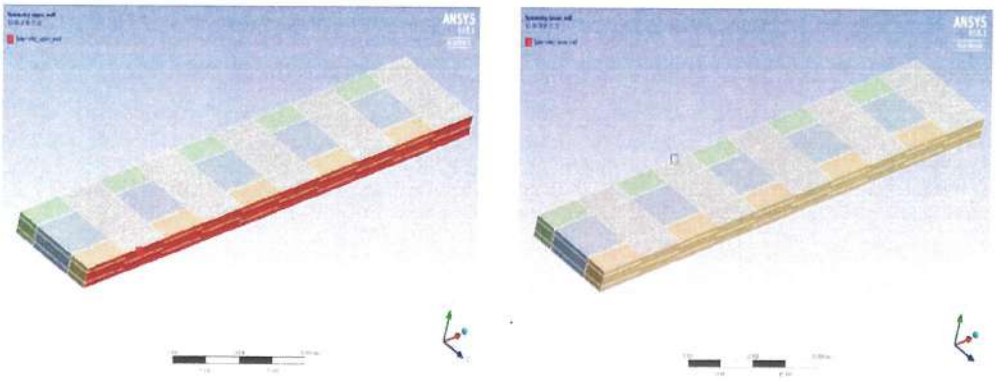

Figure 19. Symmetric upper & lower wall

As per the requirement of the plate fin heat exchanger, the material should be able to hold up against the low temperature as the gas or liquid flows through the fins and must have effective thermal conductivity. The reason for the use of the Aluminium 3003 is its extensive use in such applications as per the industry scenario and the other required material properties. The separating plate and fins are made of the aluminium 3003 which have better thermal conductivity, light weighted, etc. The properties of the aluminium used are shown in Table IV.

Table IV. Aluminium 3003 properties

Properties	**Value**
Density, ρ (in Kg/m^3)	2730
Specific heat, C_p (in j/Kg-K)	893
Thermal conductivity, k (in W/m-K)	180

4.2 RESULT AND DISCUSSION

Figure 20 and 21 shows the reduction in the temperature of the hot fluid and increase in the temperature of the cold fluid, as it is flowing through the alternate fins stacks and, the heat transfer is taking place by the convection from fluid to fin and by the conduction from the fins to fluids and vice versa. The work is carried out for the methane liquefaction, initially the methane is at 300 K temperature which we have to liquefy with the help of propane (coolant). The propane is initially at 90 K temperature and will be taking heat from the fins by conduction and convection. A variation has been seen in the result, calculated analytically and theoretically. On the theoretical basis, it can be said that there will be 8 K reduction approximately in the hot fluid temperature per 90 mm length of the heat exchanger core but through analytical results it has been observed that the hot fluid temperature is getting reduced by 10 K approximately. Also there is approx. 13 K rise in temperature of cold fluid. From the figure, it can be observed that the temperature is getting changed in a different pattern, as the flow was interrupted at many places because of the alternate offset strip in the flow channel.

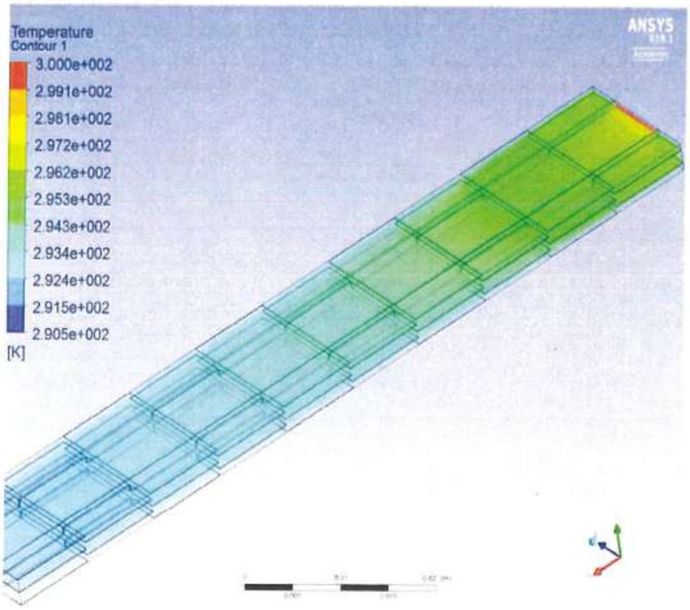

Figure 20. Temperature reduction in hot fluid

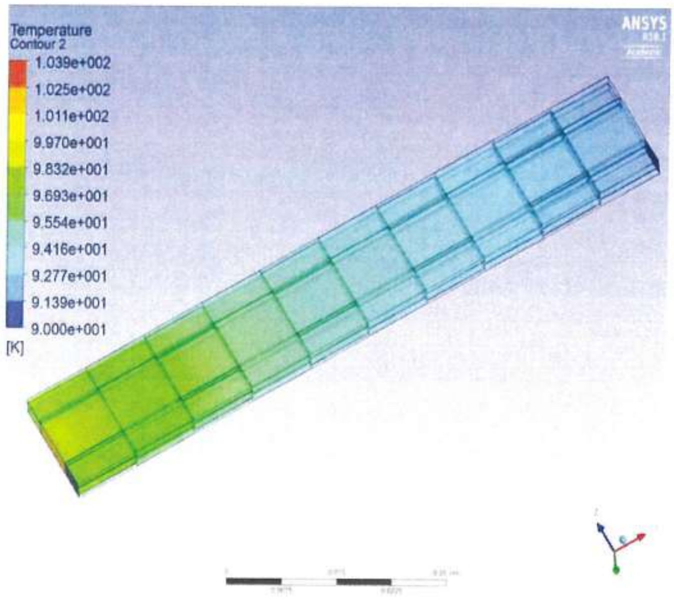

Figure 21. Temperature reduction contour for cold fluid

CHAPTER 5. CONCLUSION

5.1 CONCLUDING STATEMENT

The plate fin heat exchanger design has been done by both theoretical and analytical processes. Through the theoretical modelling, it was found that the heat exchanger is able to perform its desired function and with the effectiveness above 90 % provide a large heat transfer area compared to other heat exchangers. The analytical design included the use of Catia software for design, Ansys software for the analysis and Matlab software to make a program for the PFHE design. From the observation of analytical results, it has been found that the results are slightly better than the theoretical calculations. A concept of novelty has been tried in this work by flipping the alternate fin stack. The following points have been observed in this study:

- The effectiveness of the plate fin heat exchanger calculated is above 90%, which is reliable for the liquefaction work.
- The temperature reduction found through the analysis results are slight better than the theoretical results in hot fluid and temperature rise in cold fluid.
- The material used Aluminium 3003 for fins and separating plates withstood its desired function against the temperature.
- Pressure drop found by the theoretical study for both methane (hot fluid) and propane (cold fluid) is within the limits.

5.2 FUTURE SCOPE

Various types of fins geometries are available for the work in the plate fin heat exchanger and many correlations that are available can be used. The correlation used in this work for the hydraulic diameter is already validated with the correlation provided by the Manglik and Bergles. The following activities are proposed for future work:

- An experimental setup can be made and testified of this work.
- Slight changes can be made in the geometry of the fins
- Pressure drop can be controlled further although it's within the limit
- An analysis of the whole geometry at once can be done with better physical memory.

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7. MATLAB PROGRAM

7.1 MATLAB PROGRAMMING FOR PLATE FIN HEAT EXCHANGER DESIGN FOR LIQUEFACTION OF NATURAL GASES

```
% Program to Design a Plate Fin Heat Exchanger with offset strip fin
clc
clear
disp('Program to Design a Plate Fin Heat Exchanger with offset strip fin')

disp('Insert input Data for Calculation :')
format long
f = input('Fin Density/m : ')
h = input('Fin Height(in m): ')
Lamda_1 = input('Fin offset length(in m): ')
Lamda_2 = Lamda_1
delta = input('Fin thickness(in m): ')
disp('Distance between plates or plate spacing(in m): ')
b1 = h + delta
b2 = b1
disp('Fin Length(in m): ')
Lf = (b1/2)-delta
disp('Pitch of fin(in m): ')
Pf = 1/f
disp('Normal fins spacing(in m): ')
s = (Pf)- delta
delta_wall = input('Plate thickness(in m): ')
Kw = input('Thermal Conductivity of fin material(in J/Kg-K): ')
L1 = input('Width of Heat Exchanger Core(in m): ')
L2 = input('Length of Heat Exchanger Core(in m): ')
L3 = input('Height of Heat Exchanger Core(in m): ')

disp('Thermal Input')

Th1 = input('Hot fluid inlet temperature(in K): ')
Th2 = input('Hot fluid outlet temperature(in K): ')
Tc1 = input('Cold fluid inlet temperature(in K): ')
m_dot_hot = input('Hot fluid mass flow rate(in Kg/sec): ')
```

```
m_dot_cold = input('Cold fluid flow rate(in Kg/sec): ')
p1 = input('Hot fluid pressure(in Kpa): ')
p2 = input('Cold fluid pressure(in Kpa): ')
cph = input('Specific heat of hot fluid(in J/Kg-K): ')
cpc = input('Specific heat of cold fluid(in J/Kg-K): ')
rho_hot_i = input('Density of hot fluid at inlet(in Kg/m^3): ')
rho_hot_o = input('Density of hot fluid at outlet(in Kg/m^3): ')
rho_cold_i = input('Density of cold fluid at inlet(in Kg/m^3): ')
rho_cold_o = input('Density of cold fluid at outlet(in Kg/m^3): ')
disp('Heat capacity of hot fluid(in W/K)')
Ch= m_dot_hot*cph
disp('Heat capacity of cold fluid(in W/K)')
Cc= m_dot_cold*cpc
if(Ch>Cc)

   Cmax = Ch

   Cmin = Cc
else
   Cmax = Cc

   Cmin = Ch
end
disp('Cold fluid temperature at outlet(in K)')
Tc2 = Tc1+((Ch/Cc)*(Th1-Th2))
disp('Average temperature of hot fluid(in K)')
Th_avg = (Th1+Th2)/2
disp('Average temperature of cold fluid(in K)')
Tc_avg = (Tc1+Tc2)/2

disp('Thermophysical Properties of hot fluid at avergae temperature')

pho_1 = input('Density of hot fluid(in Kg/m^3): ')
cp1 = input('Specific heat of hot fluid(in J/Kg-K): ')
k1 = input('Thermal conductivity of hot fluid(in W/m-K): ')
```

```
mu_1 = input('Viscosity of hot fluid(in Pa-s): ')
Pr1 = input('Prandtl No. of hot fluid: ')

disp('Thermophysical Properties of hot fluid at avergae temperature')
pho_2 = input('Density of cold fluid(in Kg/m^3): ')
cp2 = input('Specific heat of cold fluid(in J/Kg-K): ')
k2 = input('Thermal conductivity of cold fluid(in W/m-K): ')
mu_2 = input('Viscosity of cold fluid(in Pa-s): ')
Pr2 = input('Prandtl No. of cold fluid: ')

disp('Solution : ')

disp('No. of Passages for hot fluid: ')
Np =((L3)-(b2)-(2*(delta_wall)))/((b1)+(b2)+(2*(delta_wall)))
disp('Note: No. of passage for cold fluid is one more than the hot fluid: ')
disp('Total no. of fins for hot fluid side: ')
nf1= ((L1)*(Np))/(Pf)
disp('Total no. of fins for cold fluid side: ')
nf2= (L1*(Np+1))/Pf
disp('Total Primary Area of hot fluid side = Total Plate Area + Fin base Area +
Passage side Wall Area + Passage front and back wall area(in m^2): ')

Ap1 = (2*(L1)*(L2)*(Np)) - (2*(delta)*(L2)*(nf1)) + (2*(b1)*(L2)*(Np)) +
(2*((b2)+(2*(delta_wall)))*(L1)*((Np)+1))

disp('Total Primary Area of cold fluid side = Total Plate Area + Fin base Area +
Passage side Wall Area + Passage front and back wall area(in m^2): ')

Ap2 = 2*L1*L2*(Np+1)- 2*delta*L2*nf2 + 2*b2*L2*(Np+1) +
2*(b1+2*delta_wall)*L1*Np

disp('No. of offset strip fins for hot fluid side: ')
noff1 = L2/Lamda_1
disp('No. of offset strip fins for cold fluid side: ')
noff2 = L2/Lamda_2
```

```
disp('The total fin areaof hot fluid side =  Fin Surface Area + Internal offset strip
edge Area + First and last offset strip edge Area + Offset strip edge Area(in m^2):
')
Af1 = 2*(b1-delta)*L2*nf1 + (Pf-delta)*delta*(noff1-1)*nf1 + 2*Pf*delta*nf1 + 2*(b1-
delta)*delta*noff1*nf1

disp('The total fin areaof cold fluid side =  Fin Surface Area + Internal offset strip
edge Area + First and last offset strip edge Area + Offset strip edge Area(in m^2):
')
Af2 = 2*(b2-delta)*L2*nf2 + (Pf-delta)*delta*(noff2-1)*nf2 + 2*Pf*delta*nf2 + 2*(b2-
delta)*delta*noff2*nf2

disp('Total Surface Area of hot fluid side(in m^2): ')
At1 = Af1+Ap1
disp('Total Surface Area of cold fluid side(in m^2): ')
At2 = Af2+Ap2
disp('Free Flow Area of hot fluid side(in m^2): ')
Ac1=(b1-delta)*(Pf-delta)*nf1
disp('Free Flow Area of cold fluid side(in m^2): ')
Ac2=(b2-delta)*(Pf-delta)*nf2
disp('Frontal Area of hot fluid side(in m^2): ')
Afr1 = L1*L3
disp('Frontal Area of cold fluid side(in m^2): ')
Afr2 = L1*L3
disp('Hydraulic diameter of hot fluid side(in m): ')
Dh1 = 4*Ac1*L2/At1
disp('Hydraulic diameter of cold fluid side(in m): ')
Dh2 = 4*Ac2*L2/At2
disp('Hydraulic diameter by Manglik and Bergles(in m): ')
Dh = (4*(Pf-delta)*b1*Lamda_1)/(2*((Pf-
delta)*Lamda_1+b1*Lamda_1+b1*delta)+(Pf-delta)*delta)
disp('Porosity of hot fluid side: ')
Sigma_1 = Ac1/Afr1
disp('Porosity of cold fluid side: ')
```

```
Sigma_2 = Ac2/Afr2
disp('Volume of the exchanger for hot fluid side(in m^3): ')
Vp1 = b1*L1*L2*Np
disp('Volume of the exchanger for cold fluid side(in m^3): ')
Vp2 = b2*L1*L2*(Np+1)
disp('Surface area density of hot fluid side(in m^2/m^3): ')
Beta_1 = At1/Vp1
disp('Surface area density of cold fluid side(in m^2/m^3): ')
Beta_2 = At2/Vp2
disp('Mass Velocity of hot fluid(Kg/m^2-sec): ')
G1 = m_dot_hot/Ac1
disp('Mass Velocity of cold fluid(Kg/m^2-sec): ')
G2 = m_dot_cold/Ac2
disp('Reynold Number of hot fluid: ')
Re1 = (G1*Dh1)/mu_1
disp('Reynold Number of cold fluid: ')
Re2 = (G2*Dh2)/mu_2

disp('Colburn factor for hot fluid: ')

j1a = 0.6522*(Re1^(-0.5403))*(((Pf-delta)/(b1-delta))^(
0.1541))*((delta/Lamda_1)^(0.1499))*((delta/(Pf-delta))^(-0.0678))

j1b = (1+5.269*(10^(-5))*(Re1^(1.34))*(((Pf-delta)/(b1-
delta))^(0.504))*((delta/Lamda_1)^(0.456))*((delta/(Pf-delta))^(-1.055)))^0.1

j1 = j1a*j1b

disp('Colburn factor for cold fluid: ')

j2a = 0.6522*(Re2^(-0.5403))*(((Pf-delta)/(b2-delta))^(-
0.1541))*((delta/Lamda_2)^(0.1499))*((delta/(Pf-delta))^(-0.0678))

j2b = (1+5.269*(10^(-5))*(Re2^(1.34))*(((Pf-delta)/(b2-
delta))^(0.504))*((delta/Lamda_2)^(0.456))*((delta/(Pf-delta))^(-1.055)))^0.1
```

```
j2 = j2a*j2b

disp('Friction factor for hot fluid: ')

f1a = 9.6243*(Re1^(-0.7422))*(((Pf-delta)/(b1-delta))^(-
0.1856))*((delta/Lamda_1)^(0.3053))*((delta/(Pf-delta))^(-0.2659))

f1b = (1+7.669*(10^(-8))*(Re1^(4.429))*(((Pf-delta)/(b1-
delta))^(0.92))*((delta/Lamda_1)^(3.767))*((delta/(Pf-delta))^(0.236)))^0.1

f1 = f1a*f1b

disp('Friction factor for hot fluid: ')

f2a = 9.6243*(Re2^(-0.7422))*(((Pf-delta)/(b2-delta))^(-
0.1856))*((delta/Lamda_2)^(0.3053))*((delta/(Pf-delta))^(-0.2659))

f2b = (1+7.669*(10^(-8))*(Re2^(4.429))*(((Pf-delta)/(b2-
delta))^(0.92))*((delta/Lamda_2)^(3.767))*((delta/(Pf-delta))^(0.236)))^0.1

f2 = f2a*f2b

disp('Heat Transfer Coefficient of hot fluid(W/m^2-K): ')
h1 = (j1*G1*cp1)/(Pr1^(2/3))
disp('Heat Transfer Coefficient of cold fluid(W/m^2-K): ')
h2 = (j2*G2*cp2)/(Pr2^(2/3))
m1 = (2*h1*(1+(delta/Lamda_1))/(Kw*delta))^0.5
disp('Fin length(in m): ')
Lf1 = (b1/2)-delta
disp('Single Fin efficiency of hot fluid: ')
etaf_1 = (tanh(m1*Lf1))/(m1*Lf1)
m2 = (2*h2*(1+(delta/Lamda_2))/(Kw*delta))^0.5
disp('Fin length(in m): ')
Lf2 = (b2/2)-delta
disp('Single Fin efficiency of cold fluid: ')
```

```
etaf_2 = tanh(m2*Lf2)/(m2*Lf2)
disp('Overall Fin Surface efficiency of hot fluid side: ')
etao1 = 1-((1-etaf_1)*(Af1/At1))
disp('Overall Fin Surface efficiency of cold fluid side: ')
etao2 = 1-((1-etaf_2)*(Af2/At2))
disp('Wall Conduction Area(in m^2): ')
Aw = 2*L1*L2*(Np+1)
disp('Thermal Resistance: ')
Rw = delta_wall/(Kw*Aw)
disp('Overall Heat Transfer Coefficient: ')
UA1 = 1/(etao1*h1*At1)
UA2 = 1/(etao2*h2*At2)
UA = 1/(UA1+Rw+UA2)

disp('Thermal design of heat exchanger: ')

disp('Heat Capacity Ratio: ')
CR=Cmin/Cmax
disp('Mean Density of hot fluid(Kg/m^3): ')
rho_hot_m =(2*rho_hot_i*rho_hot_o)/(rho_hot_o + rho_hot_i)
disp('Mean Density of cold fluid(Kg/m^3): ')
rho_cold_m =(2*rho_cold_i*rho_cold_o)/(rho_cold_o + rho_cold_i)
disp('Heat transfer rate of hot fluid(in W): ')
q_hot = m_dot_hot*cph*(Th1-Th2)
disp('Heat transfer rate of cold fluid(in W): ')
q_cold = m_dot_cold*cpc*(Tc2-Tc1)
disp('Number of Transfer Unit: ')
NTU =(UA)/Cmin
disp('Effectiveness of Heat Exchnager: ')
epsilon = 1-exp((1/CR)*(NTU^0.22)*(exp(-CR*(NTU^0.78))-1))
disp('Maximum Heat transfer rate(in W): ')
q_max = epsilon*Cmin*(Th1-Tc1)
Re = input('Reynold no.: ')

disp('Friction factor used for circulation: ')
```

```
if(Re>2300)
   fd = 0.049*(Re^(-0.2))
else
   fd= 16/Re
end

if(Re>2300)
   kd_tube = 1.09068*(4*fd)+0.05884*((4*fd)^0.5)+1
else
   kd_tube = 1.33
end

disp('Velocity distribution coefficient: ')

if(Re>2300)
   kd_square = 1+1.17*(kd_tube-1)
else
   kd_square = 1.39
end

disp('Expansion coefficient for hot fluid side: ')
ke_square1 = 1-(2*kd_square*(Sigma_1))+((Sigma_1)^2)
disp('Expansion coefficient for cold fluid side: ')
ke_square2 = 1-(2*kd_square*(Sigma_2))+((Sigma_2)^2)
disp('Jet contraction ratio for hot fluid side: ')
Cc_tube1 = 4.374*(10^(-4))*(exp(6.737*((Sigma_1)^0.5)))+0.621
disp('Jet contraction ratio for cold fluid side: ')
Cc_tube2 = 4.374*(10^(-4))*(exp(6.737*((Sigma_2)^0.5)))+0.621

disp('Contracrtion coefficient for hot fluid side: ')

kc_square1 = (1-(2*(Cc_tube1))+((Cc_tube1)^2)*((2*kd_square)-
1))/((Cc_tube1)^2)
```

```
disp('Contracrtion coefficient for cold fluid side: ')

kc_square2 = (1-(2*(Cc_tube2))+((Cc_tube2)^2)*((2*kd_square)-
1))/((Cc_tube2)^2)

disp('Pressure drop on hot fluid side(in KPa): ')

delta_p_hot = ((G1^2)/(2*rho_hot_i))*((1-
((Sigma_1)^2)+kc_square1)+2*((rho_hot_i/rho_hot_o)-
1)+(((4*f1*L2)/Dh1)*(rho_hot_i/rho_hot_m))-((1-((Sigma_1)^2)-
ke_square1)*(rho_hot_i/rho_hot_o)))

disp('Pressure drop on cold fluid side(in KPa): ')

delta_p_cold = ((G2^2)/(2*rho_cold_i))*((1-
((Sigma_2)^2)+kc_square2)+2*((rho_cold_i/rho_cold_o)-
1)+(((4*f2*L2)/Dh2)*(rho_cold_i/rho_cold_m))-((1-((Sigma_2)^2)-
ke_square2)*(rho_cold_i/rho_cold_o)))
```