EXPERIMENTAL AND NUMERICAL STUDIES ON PLATE FIN HEAT EXCHANGER

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE DEGREE OF

MASTER OF TECHNOLOGY

In

Mechanical Engineering

By

MITRAVANU SAHOO

Roll No: 213ME5454

DEPARTMENT OF MECHANICAL ENGINEERING
NATIONAL INSTITUTE OF TECHNOLOGY ROURKELA
INDIA

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CERTIFICATE

This is to certify that the thesis entitled**, "EXPERIMENTAL AND NUMERICAL STUDIES ON PLATE FIN HEAT EXCHANGER"** submitted by **Mr. Mitravanu Sahoo** in partial fulfilment of the requirements for the award of Master of Technology Degree in Mechanical Engineering with specialization in **Cryogenics and Vacuum Technology** at the National Institute of Technology, Rourkela is an authentic work carried out by him under my super vision and guidance.

To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other university/institute for the award of any degree or diploma.

Date:

Prof. Ranjit Kumar Sahoo
Department of Mechanical Engineering
National Institute of Technology, Rourkela
Rourkela-769008

Declaration

I hereby declare that the work which is being presented in this thesis entitled "**Experimental and Numerical Studies on Plate Fin Heat Exchanger**" in partial fulfilment of the requirements for the award of M.Tech. Degree, submitted to the Department of Mechanical Engineering, National Institute of Technology, Rourkela, is an authentic record of my own work under the supervision of Prof. Ranjit Kumar Sahoo. I have not submitted the matter embodied in this thesis for the award of any other degree or diploma to any other university or Institute.

Date: 1st June 2015

Mitravanu Sahoo

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I would also like to thank all staff members of **Mechanical Engineering Department** specially **Prof. S.S. Mahapatra** who has provided the **CFD Lab** Where I completed the maximum part of my project work.

Place:

Date: **Mitravanu Sahoo**

ABSTRACT

In the cryogenics field, high effectiveness heat exchangers of the order of 0.96 or higher are widely used for preserving the refrigeration effect produced. So, there will be no liquid yield if the effectiveness falls below that of the design value. Due to high effectiveness, low weight & compactness, the compact heat exchangers have their extensive applications in the air-conditioning system, oil industries, food industries & in the process industries. The plate fin heat exchangers (PFHE) is a type of compact heat exchanger which is manufactured by brazing a stack of alternate plates (parting sheets) & corrugated fins together. The exchange of heat occurs by the streams through the fins. Generally, aluminium is used for manufacturing PFHE due to their high thermal conductivity & low cost. In the plate fin heat exchanger, the pressure drop is also measured along with the effectiveness. The increase in pressure gradient can be outweighed by decreasing the passage length, so that an acceptable pressure drop can be achieved. There are enormous research is going on to make out the heat transfer phenomena & also to determine the dimensionless heat transfer coefficients that is the Colburn factor (j) and the friction factor (f). This thesis on the offset strip plate fin heat exchanger compares the effectiveness, overall thermal conductance & the pressure drop obtained from the experimental data with some correlations on plate fin heat exchanger i.e., Joshi-Webb correlation, Maiti-Sarangi correlation, Manglik-Bergles correlation and also with the numerically achieved data obtained by using CFD.

Key words: Offset or serrated fin, Computational fluid dynamics, Colburn factor, Friction factor

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NOMENCLATURE

A	Heat transfer area of the heat exchanger with subscripts h or c denoting hot and cold fluid, m^2
A_{ff}	Free flow area available for hot or cold fluid with subscripts h or c respectively, m^2
A_{fr}	Frontal area available for hot or cold fluid with subscripts h or c respectively, m^2
A_w	Frontal area available for hot or cold fluid with subscripts h or c respectively, m^2
a	Plate thickness, m
a_{ff}	Fin surface area, m^2
a_f	Free flow area/fin, m^2
a_{fr}	Frontal area/fin, m^2
a_w	Total wall cross sectional area for longitudinal conduction, m
C	Flow stream heat capacity rate with subscript h or c for hot and cold fluids, W/K.
C_d	Coefficient of discharge, dimensionless
C_{min}	Minimum of C_c or C_h , W/K
c_p	Specific heat at constant pressure, J/kg-K
C_r	Heat capacity rate ratio, dimensionless
D_e	Equivalent diameter of the flow passage, m
f	Fin frequency, Number of fins per meter length, fins/m
f	Fanning friction factor, dimensionless
G	Core mass velocity, kg/ m^2s
H	No flow height (stack height) of the heat exchanger core, m
h	Height of fins, m
h	Convective heat transfer coefficient, W/ m^2
j	The Colburn factor, non-dimensional heat transfer characteristic

K_c Contraction coefficient, no units

K_e Expansion coefficient, no units

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

K_w Conductivity of the wall material, W/m- K

L Fluid flow (core) length on one side of the heat exchanger, m

l Fin flow length on one side of a heat exchanger, m

l_e Effective fin length for efficiency determination

m Mass flow rate, kg/sec.

N Total number of layers or total number of fluid passages

NTU Number of heat transfer units, $UA/C_{\min}$, dimensionless

η_{tuc} Number of heat transfer units based on cold fluid side, $(\eta_0 hA)_c / C_c$

η_{tuh} Number of heat transfer units based on the hot fluid side, $(\eta_0 hA)_h / C_h$

P_f Fin pitch,1/ f, m

Pr Prandtl number of the fluid

Q Heat load, W

Re Reynolds number, dimensionless

Re^* Critical Reynolds number for heat transfer and pressure drop considerations

s Spacing between adjacent fins, m

T Temperature of the fluid (with subscripts c, h or i, o)

t Thickness of fin, m

Uo Overall heat transfer coefficient. W/m K

W Width of the core, m

CHAPTER 1

INTRODUCTION

1 INTRODUCTION

A heat exchanger is a device by which thermal energy or enthalpy is transferred between two or more fluids having different temperatures and which are also in thermal contact with each other. The enthalpy can transfer between two or more fluids, between fluid and solid particulates and between fluid and a solid surface which are in thermal contact with each other. Usually in heat exchangers there is no work interaction. The heat exchangers are also adiabatically insulated, so no heat transfer takes place. The cooling and heating of a fluid, condensation of a single or multi- compound fluid, evaporation of a single or multi-compound fluid are the main applications of the heat exchanger. Generally, high effectiveness heat exchangers are used in cryogenic applications. The effectiveness of heat exchangers used in liquefiers is of the order of .96 and above. There will be no liquid yield if the effectiveness of the heat exchangers falls below the design value. But in case of the use of heat exchangers in aircrafts, high effectiveness and performance is not so required rather the aim is to keep the weight and volume of the heat exchanger minimum. These requirements of low volume and weight of the heat exchanger lead to the generation of compact heat exchangers. In general Compact heat exchangers have large surface area density i.e. large surface area to volume ratio which is of the order 700 m^2/m^3or greater than this value for gas and it should be 300 $m^2/m^3$for two-phase streams and liquids.

1.1 Plate fin heat exchangers

Plate fin heat exchangers are the type of heat exchangers having triangular or rectangular corrugated fins, with the parting sheets or plates (spacers) sandwiched between the parallel plates. The plates and fins separate the two fluid streams from each flow passages. Two or more fluid sides can be formed in the heat exchangers by the connection of the alternative fluid passage using suitable headers. That means it is a stack of parting sheets placed alternatively and the corrugated plate fins brazed collectively in a single block. Since the flow of the passage is through the parting sheets which is controlled by the fins causes the heat transfer between the fluid streams. These fins are formed by the process of rolling or by using a die. The metal joining processes such as welding, brazing, soldering, extrusion etc. are used to attach the fins to the plates. The fins may be used on both sides in case of the gas-to-gas heat exchangers, but fins are usually used on the gas side only in case of the gas-to-liquid heat exchangers. The fins those are employed on the liquid side are used for flow mixing process and also give structural strength. The plate fin heat exchangers are also known as Matrix heat exchangers in Europe. The plate fin heat exchangers have the advantage of high effectiveness, low weight, compactness and moderate cost.

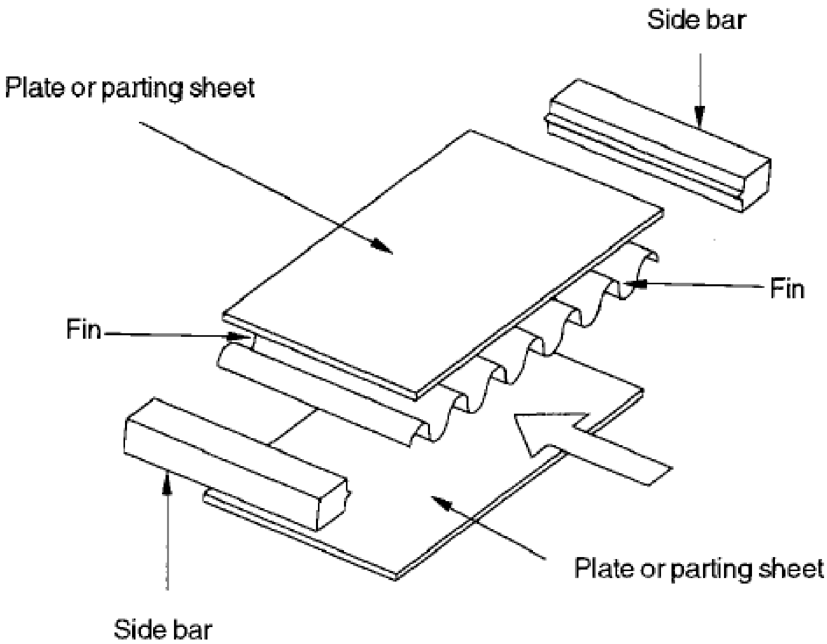


Figure 1. A stack of fins placed between the parting sheets (Shah and Sekulic [39])

The separating plate and the fins in the plate fin heat exchangers act as the primary and secondary heat transfer surfaces respectively. The spilling over of the fluid to the outside is prevented by the side bar of the plate fin heat exchangers. The side bars and fins are brazed to the parting sheets for providing mechanical stability and also to provide a good thermal link. A clear view of a compact plate fin heat exchanger with two layers is shown in figure 2.

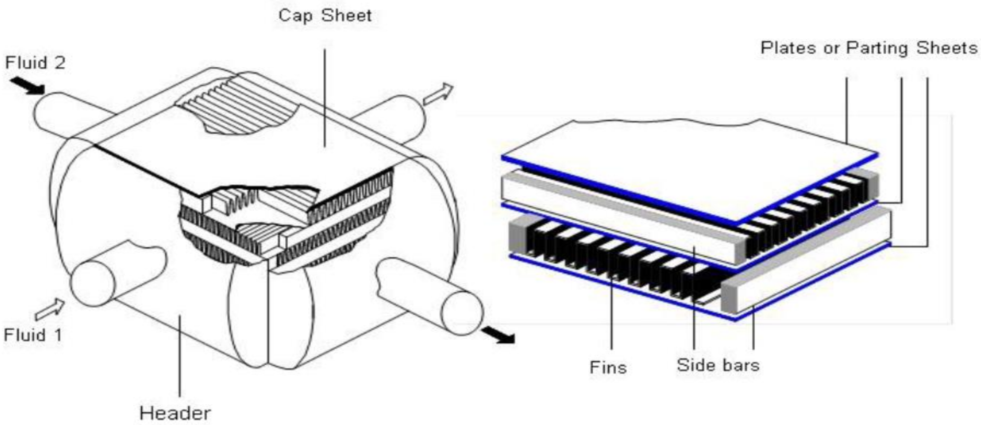


Figure 2. The assembly details of a plate fin heat exchanger (Shah and Sekulic[39])

The compact plate fin heat exchangers can be divided into various types depending on their fin structures. Some fin types are:

1. Triangular or rectangular cross-section plate fins(straight and uncut fins)
2. Wavy fins.
3. Various types of interrupted fins like offset strip fin, perforated fin, louver and pin fin.

The strip fins are also known as serrated or segmented fin lance offset fin and offset fin.

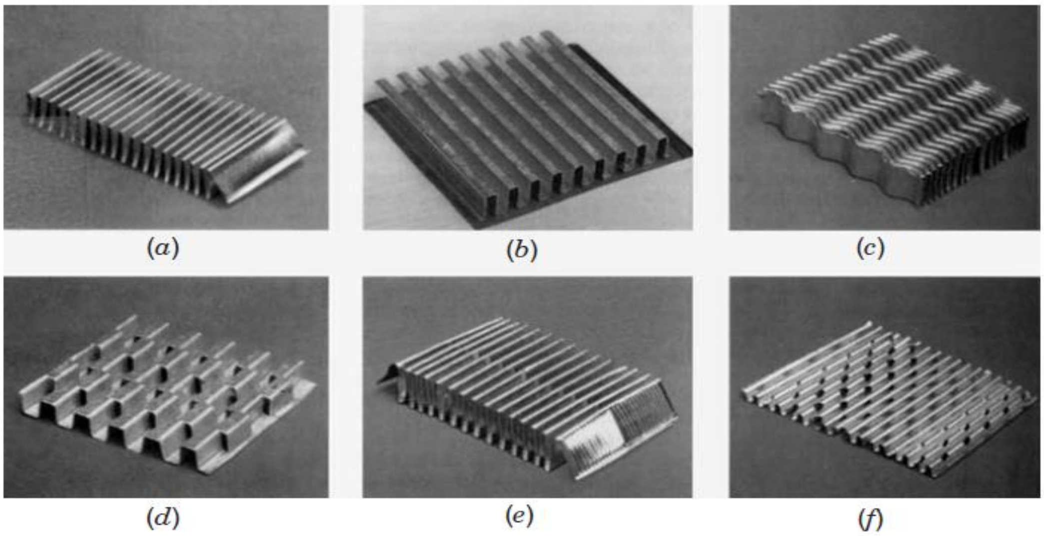


Figure 3. Plate fin heat exchangers having corrugated fin geometries (a) Triangular plain fin (b) Rectangular plain fin (c) Wavy fin (d) Serrated or offset strip fin (e) Multi-louver fin (f) Perforated fin (Shah and Sekulic [39])

1.1.1 Merit and demerit of compact plate fin heat exchangers

The advantages of plate fin heat exchangers over tube fin, shell & tube and other types of heat exchangers are:

1. Compactness: The plate fin Heat exchangers generally provide a very large value of the heat transfer area density which is typically 1000 m^2/m^3.
2. Good heat transfer area and high overall heat transfer coefficient: It is because of the narrow passage of the plate fin heat exchangers.
3. Effectiveness: High effectiveness of the order of the 95% or above can be achieved.
4. Control of temperatures: The temperature difference of the order of 3 K between two single phase fluid streams and of about 1 K in case of condensation and boiling can be achieved in plate fin heat exchangers.

5. Flexibility: Very broad range of condition and also various types of fluids can be adopted by modifying the design specifications. Multi-stream operation is also possible.
6. Counter Flow: Plate fin heat exchanger can operate counter flow which gives rise to better effectiveness, unlike shell & tube heat exchangers.
7. Compact Structure: Generally the surface area density (ratio of heat transfer area and unit volume) can be up to 2500 m^2/m^3 which is much more than other heat exchangers.
8. Light Weight: In addition to the compact structure, because of the fact that the size of the plate fin heat exchanger is one-tenth of shell and tube heat exchanger for the same heat transfer area indicates that the plate fin heat exchangers of light weight.
9. Wide Range of operation and strong adaptability: The plate fin heat exchanger has a very wide temperature range from 220 ℃ to absolute zero.

Since, the plate fin heat exchanger is generally made up of aluminium, the coefficient of thermal conductivity is high and when the plate fin heat exchanger is operated below absolute zero, its ductility and tensile strength can be improved.

The disadvantages of plate fin heat exchanger are:

1. In plate fin heat exchangers, flow passage is so small that the heat exchanger block causes the reduction in pressure. Along with this once the dirt is formed, the task of cleaning and maintenance is difficult.
2. It is necessary to notice that heat exchanger material should not be prone to corrosion.
3. In plate fin heat exchangers, if leakage between the passages occurs, then it is difficult to repair this.

1.1.2 Selection of material

The alloys of aluminium are the best-suited material and are extremely used for the manufacturing of plate fin heat exchanger. Stainless steel is also used for the manufacturing of plate fin heat exchangers. However, the selection of the material depends on the process temperature and pressure. For the application in low temperature or cryogenic applications, aluminium alloy is widely used in plate fin heat exchangers due to its low weight, high ductility and increasing strength under low-temperature conditions. The fins and the side bars (secondary surfaces) are joined with the separating plates by vacuum brazing techniques or by using dip brazing technique. An aluminium alloy of a lower melting point is the brazing material for the aluminium made heat exchangers. But a nickel-based alloy with suitable melting point and having good welding characteristics is used for a stainless steel made heat exchanger.

1.1.3 Manufacturing of the plate fin heat exchangers

In the manufacturing process of the plate fin heat exchangers, the basic principle is same for all materials and for all sizes. First a number of flat sheets are placed one above other and the corrugated fins are assembled like a sandwich construction. The parting sheets (the separating plates) are the primary heat transfer surface. To form a restraint between each layer, the parting sheets are placed in an alternative manner with the layers of the fins forming a stack. All the elements i.e. the partings sheets, side bars, the corrugations and the cap sheets are kept together by a jig under a predetermined load. Then, for forming a heat exchanger block it is placed in a brazing furnace. Then during welding process to ensure the brazed joints remain in contact or n't, the nozzles and the header tanks are welded to the block. The main factor lies in the fact that under which condition and in which manner the brazing process takes place. The commonly used methods are:

1. Salt bath brazing
2. Vacuum brazing

In salt bath brazing process, the assembled stack is kept inside a furnace and in the furnace it is heated to about 560℃, then assembled stacked plate is dipped inside a fused salt bath containing alkali metal chloride or fluorides. The chlorides or the fluorides fused salts functions both as the heating agent as well as the flux. It also maintains uniform temperature inside the furnace during the pre-heating period. For heat exchangers made up of aluminium alloys, the tenacious layer of aluminium oxides, grease etc. is removed by the molten salts. In salt bath brazing, the brazing takes place when temperature is increased above the melting point of the brazed alloy. Then, this brazed block kept inside water so that the residual solidified salts are separated. So, the brazed block is cleaned. Then the brazed block is allowed to completely dry.

In case of vacuum brazing technique, there is no requirement of preheating and also no flux is required. In this process, the assembled block is allowed to be heated till the brazing temperature is achieved. This heating of assembled block is generally done by the mode of radiation heat transfer using electric heaters. The heat transfer from the heated exposed surface of the assembled block to the inner side of the block occurs by the mode of conduction. By using high vacuum in the vacuum brazing process, the absence of oxygen can be increased. Since, vacuum brazing process is carried out in the absence of oxygen so there is no possibility of formation of metal oxide. In vacuum brazing the residual gas is further enhanced by alternative evacuation. The washing and drying of brazed block is not required in vacuum brazing process. Materials such as aluminium, copper, nickel alloys, stainless steel can be brazed effectively by using vacuum brazing techniques. Nowadays the process of vacuum brazing is extensively used in manufacturing of compact plate fin heat exchangers.

By using super elastic deformation and diffusion bonding a titanium plate fin heat exchanger is developed recently (as shown in the figure 4).

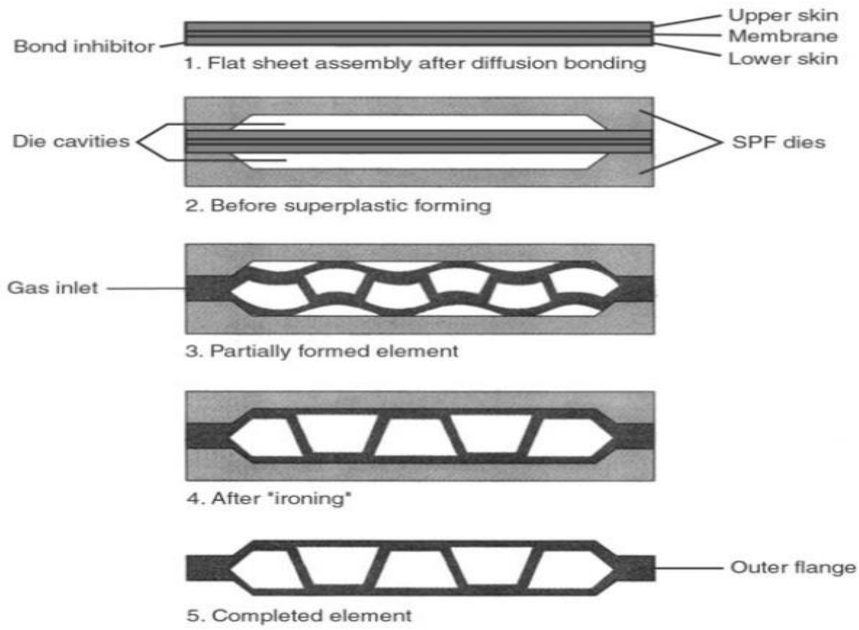


Figure 4. Process of manufacturing of a super elastically deformed diffusion bonded plate fin exchanger (Shah and Sekulic[39])

1.1.4 Applications of plate fin heat exchangers

The use of plate fin heat exchanger is very wide because it works over a very large range of pressure and temperature for gas-to-gas, gas-to-liquid and in multiphase applications. There is a wide variety of applications. In the cryogenic field, for the liquefaction of air, for the separation of air the plate fin heat exchangers are widely used. In the petrochemical industries and in very large refrigerating systems for the processing of natural gas, and their liquefaction, it is also widely used. The plate fin heat exchanger being used in cryogenic application, are very large and complex units with large dimensions. In the aerospace industries, aluminium brazed PFHEs are widely used because of its compactness and low weight-to-volume ratio. These plate fin heat exchangers are mainly used in control system of the aircraft, cooling system of the aircraft, hydraulic cooling of oil and fuel heating.

Because of the space consciousness of the automobile companies and the aircraft designers, there is more demand for the designing of the compact heat exchangers. For the application of preheating/precooling of air coming to the conditioned space, a treated hygroscopic paper with the operating limit temperature of 50℃ plate fin heat exchangers are used.

The other applications are:

a) Nuclear fuel cells, gas turbines
b) Fuel cells
c) Aero plane engines
d) Fuel processing and conditioning plants
e) Waste heat recovery plant

1.1.5 Flow arrangements

There can be two or more streams in the plate fin heat exchangers and the flow direction of these fluids streams affect the effectiveness of the heat exchanger. The arrangement of flow in plate fin heat exchanger can be of three types:

a) Parallel flow
b) Counter flow
c) Cross flow

On the basis of thermodynamics point of view, the counter flow arrangement of flow gives the most cold/heat recovery whereas the parallel flow delivers the lowest. The cross-flow arrangement delivers an intermediate action by supplying easier mechanical output and heat flow. Thus, there are mainly three types of plate fin heat exchangers.

a. **Cross-flow heat exchangers**: In these types of heat exchangers, the hot and the cold fluids flow in the perpendicular direction to each other. The effectiveness of cross flow systems lies in between the parallel flow and the counter-flow arrangement. In cross-flow heat exchangers, only two fluid streams can be handled that's why it eliminates the need of distributors. The header tanks are placed on all four sides of the core of the cross flow heat exchanger. This makes it cheap and simple. So only more effectiveness is not a major requirement. If the two fluid streams possess extremely different volume flow rates or if any one or both of the two fluid streams possesses constant temperature, then the cross flow arrangement is a good choice to be implemented. The radiators need in automobiles, aeroplane heat engines are some typical applications of this type of heat exchanger.

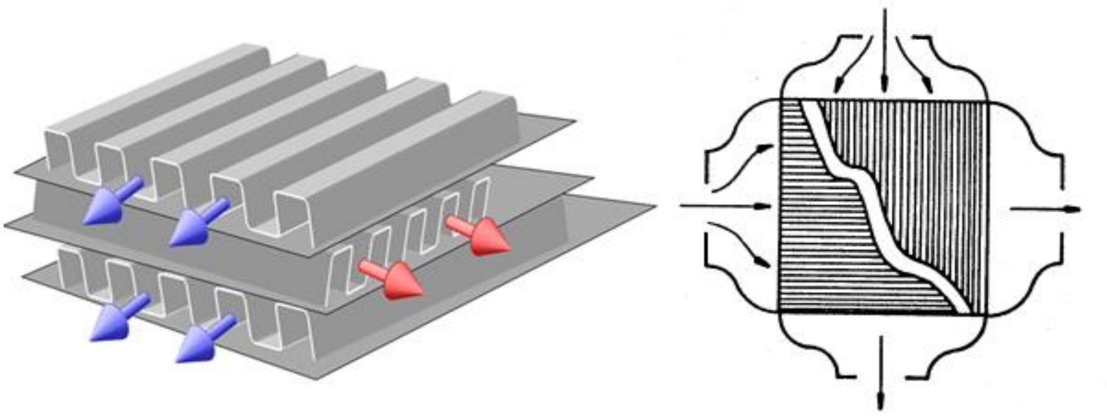


Figure 5. Flow arrangement in cross flow heat exchanger

b. **Counter flow heat exchangers**: Here the two fluid streams flow in opposite directions but the fluid streams flow parallel to the other one. These heat exchangers provide the highest efficiency and are most effective in recovery systems, for a constant overall thermal conductivity (UA), inlet temperature of the fluid, and rate of the fluid flow. The applications of these heat exchangers are vast in cryogenic refrigeration and liquefaction equipment. Due to the complicated geometries of headers, these heat exchangers are very complex in design.

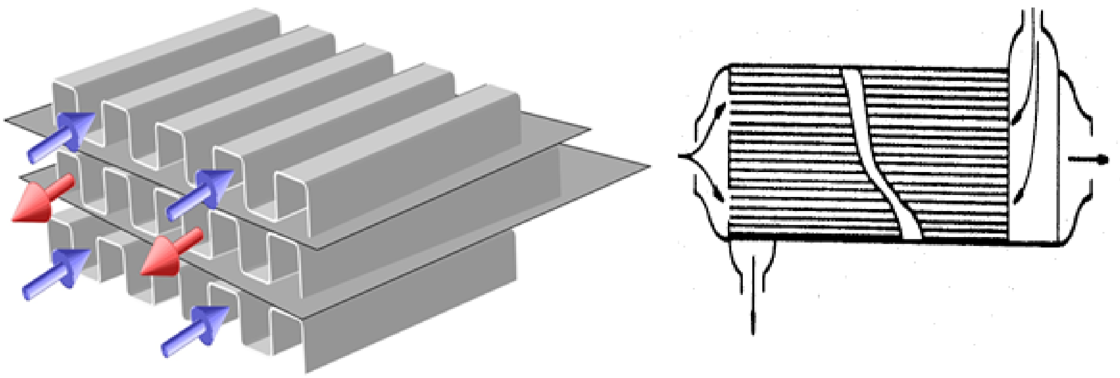


Figure 6. Flow arrangement in counter flow heat exchanger

c. **Cross-counter flow heat exchangers**: The cross-counter flow arrangements possess the combined features of both the systems. So it is a hybrid of both the counter-flow and cross-flow systems. The effectiveness of these heat exchangers is nearly in the same range as that of the counter flow arrangements. In addition, it possesses better heat transfer characteristics as that of the cross flow type. In cross-counter flow arrangements, one of the streams flow straight and the second stream flow in a zig-zag manner and in the normal direction to the first one. The second fluid stream takes the heat exchangers length while flowing along the zig-zag path opposite in direction to that of the direct stream. Cross-counter flow heat exchanger can be used in all those applications where cross flow heat exchangers are used with more flexibility in design and fabrication. These are mainly applied where two fluid streams possess different volume flow rates. The overall geometry can be optimised by this arrangement.

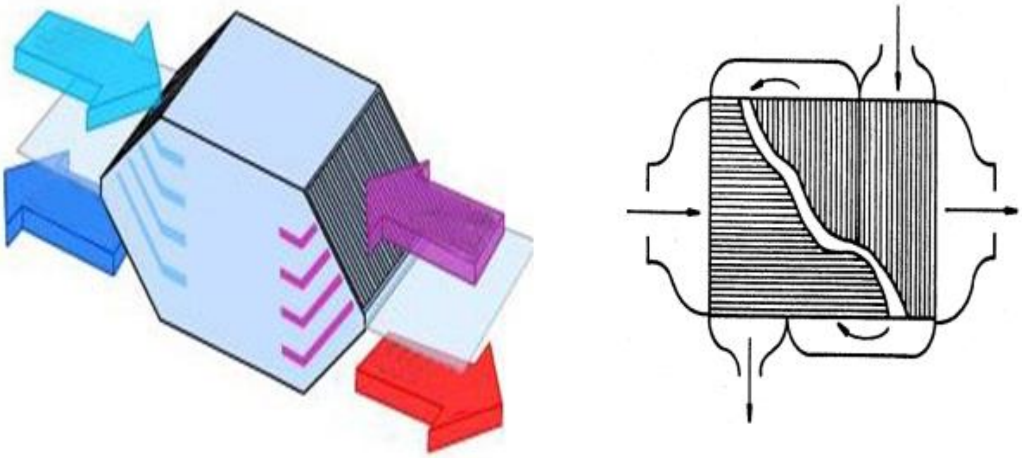
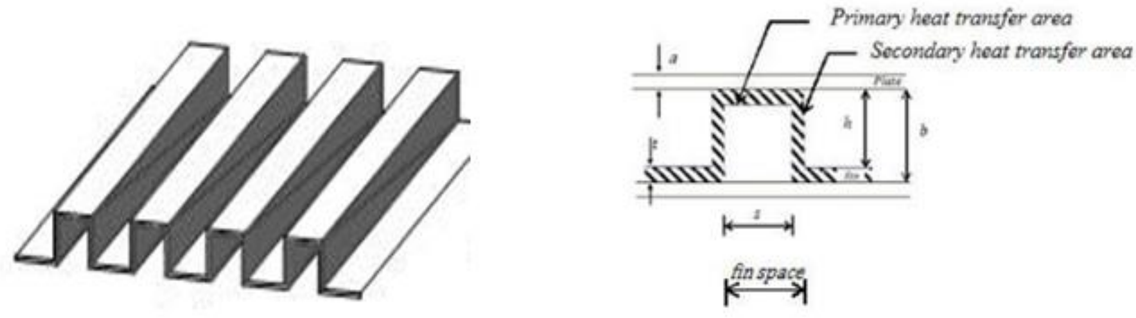


Figure 7. Flow arrangement in the cross-counter flow heat exchangers

1.2 Types of plate fin heat exchanger surfaces

Extended surfaces are generally used in plate fin heat exchangers due to small heat transfer coefficient in the gas side. That’s why the extended surfaces are configured for the increment of the heat transfer coefficients. For this purpose special geometries of the fins have been developed that gives very larger heat transfer coefficients compare to the extended plain surface. At the same time, these special sometimes cause high pressure drop. Till now, varieties of the extended surfaces have been developed i.e. plain trapezoidal, plain rectangular, wavy offset, louvered etc. We deal with the offset strip fin geometry for numerical and experimental work.

1. Plain Fins: Plain fins are straight and continuous fins. The plain fins are commonly in the form of triangular and rectangular cross-section. It is also easy to manufacture triangular design fins compared to other designs of fins. But, the limitations of the plain fins are lower heat transfer rate during laminar flow and structural weakness. For critical pressure drops, plain fins are widely used.

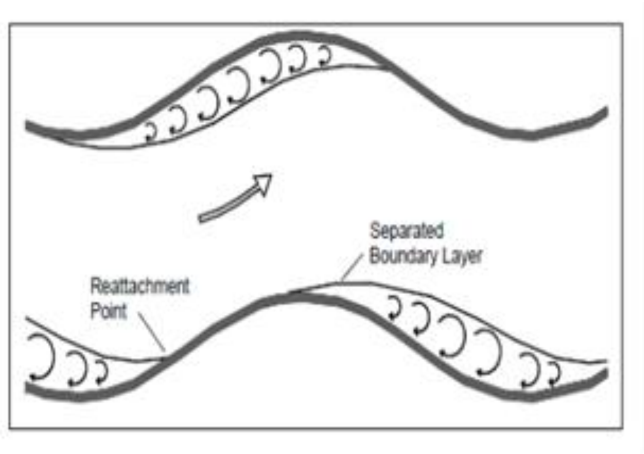


(a) (b)

Figure 8. (a) Plain fin (b) Plain rectangular fin geometry (Maiti and Sarangi [41]).

2. Wavy Fins: Wavy Fins have similar cross-sectional shapes to that of the plain fins, but is of curvy shapes in the normal to the fluid direction. The resulted wave-like structures in the wavy fins provide effective interruptions and induce a complex flow field that lead to the creation of vortex when the fluid pass over that of the concave wavy surface. These counter rotatory vortexes, those are developed results in a cork-screw pattern. The pressure drop and the heat transfer characteristics of these types of fin surfaces fall in between the plain fins and the offset fins, with the increasing Reynolds number. At high density, the thickness of the wavy fin is not limited, unlike offset strip fins. Due to this reason, wavy fins are often used in high-pressure streams particularly in those cases where poor heat transfer coefficient can be tolerated.

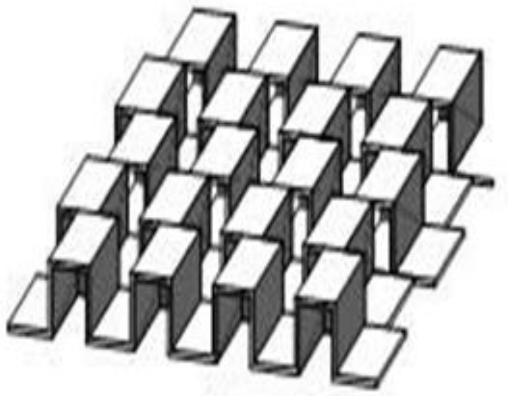
(a)



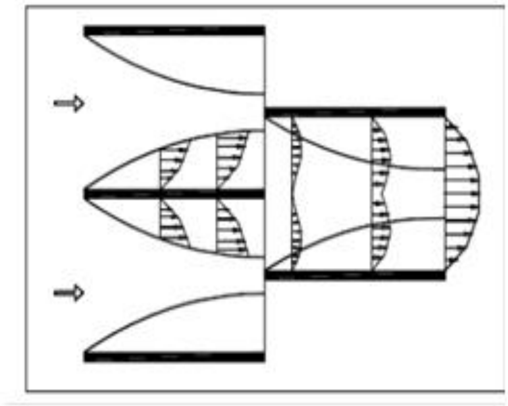
(b)

Figure 9. (a) Wavy fin (b) Boundary layer flow across the wavy fin (Maiti and Sarangi [41)].

3. Offset Strip (serrated fin): The offset type fin is similar to that of the plain rectangular fin except the fact that the fin flow length (L_f) on one side of the heat exchanger is discontinuous. This fin flow length is equal to the sum of all strip length (L_s). Due to these offsets, surface interruption is caused. So, these fins enhance the heat transfer by continuously interrupting the thermal boundary layer growth. Due to this, thin boundary layers are produced which is responsible for low resistance for heat conduction. It leads to higher heat transfer rate. The rate of heat transfer is better but the pressure drop is more in the offset arrangements.



(a)



(b)

Figure 10. (a) Offset strip fin (b) Boundary layer flow across the offset strip fin (Maiti and Sarangi [41]).

4. Louver fins: In Louver Fins, louver cuts are marked parallel and are marked from the base of the fin to its centre along the path of the heat conduction. For maintaining the structural length of the required fin, louvers are not extended towards the fin base. So ideally heat transfer across the individual louver is same as that of plain triangular fins. At low Reynolds number, the flow of the stream is parallel to the axial direction (duct flow) whereas at high Reynold number values the flow of the fluid occurs along the direction of the louvers (boundary layer flow).

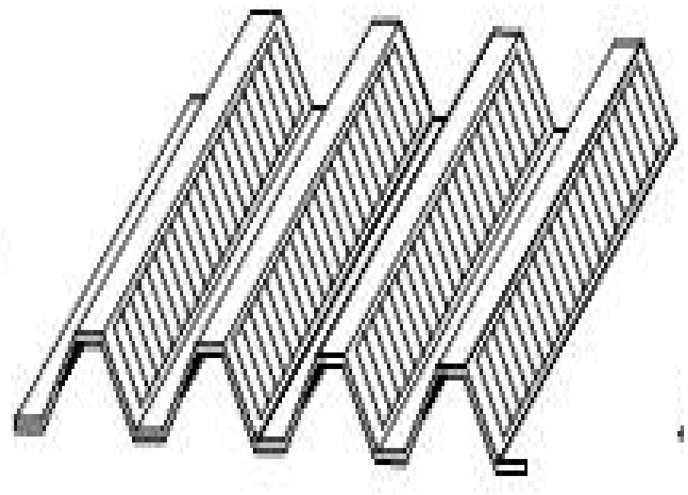


Figure 11. Louver fin (Maiti and Sarangi [41]).

5. Perforated Fin: In the fin material, a pattern of spaced holes are punched for the manufacturing of the perforated fins. Then, to form the flow channels the fin material is folded. The channels can be of rectangular or triangular in shape having rectangular or round perforations. Although these fins make enhance surface for the heat transfer, still it is manufactured in a wasteful manner because the material removed are thrown way as scraps.

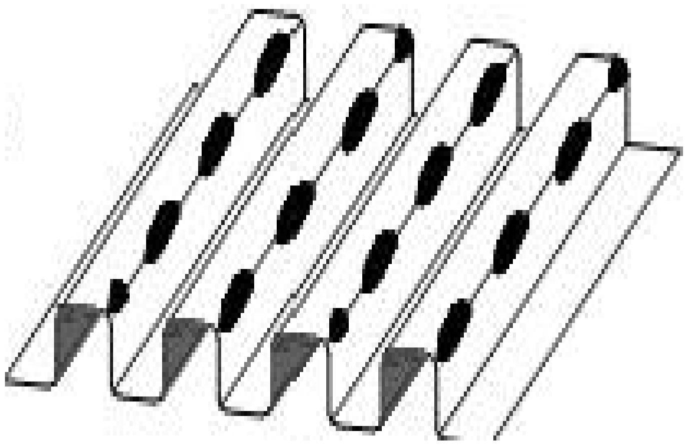


Figure 12. Perforated fins (Maiti and Sarangi [41]).

1.3 Characteristics of Heat Transfer and Flow Friction

Generally the non-dimensional forms are used for expressing the heat transfer and the flow friction characteristics of a heat exchanger and these are simply considered as the basic surface characteristic data. The correlations that are available in the in the literature which relate the Colburn Factor (j) with the Friction Factor (f) as a function Reynolds number(Re)and are given by:

$$\text{Colburn factor} = j = \frac{h}{Gc_p}\left(\Pr\right)^{2/3} \quad \text{and}$$

$$\text{Pressure drop} = \Delta p = \frac{4fLG^2}{2D_h\rho}$$

Where,

h = heat transfer coefficient, W/m^{2}K

G = Mass velocity of the basis of minimum flux flow area, Kg/m^{2}s

L = Flow passage length, m

D_h = Hydraulic Diameter, m

ρ = Mean Density, Kg/m^3

1.4 Measurement Principles

Kays and London experimented on the heat transfer characteristics along with the pressure drop characteristics of the Plate fin heat exchanger, in a most effective and reliable manner. Their experimental set-up dealt with a cross-flow type heat exchanger. But we did our experimental set-up for a counter flow type plate fin heat exchanger having offset strip fin type surface. Here the cool ambient air is drawn by the compressor through a channel and is being heated by using heat exchanger and a heater where air is heated & these heated air transfers heat to the cold air through the exchanger. The pressure gauges are used for measuring the pressure at the inlets of the both the fluids. Similarly Resistance Temperature Detectors (RTDs) are used for measuring the temperature at the inlet and outlet of both the fluids. By measuring the temperature and mass flow rate, effectiveness can be calculated. Overall thermal conductance and Number of Transfer units (NTU) can be determined from the measurement of effectiveness.

1.5 Objective of the study

For the rating and sizing offset strip plate fin heat exchanger, we have many correlations between the heat transfer factor in the literature coefficient and friction reviewed. The main objective is the evaluation of the performance parameters of a counter flow heat exchanger. This is done by following steps:

This is done by following steps:

i. Based on the chosen correlation, an offset strip plate fin heat exchanger have been designed.
ii. For a given design data, industrial fabrication of the plate fin heat exchanger is done.
iii. Test fabrication for testing.
iv. The effectiveness, heat transfer coefficient, pressure drop etc. are obtained for the experiment and these obtained values will be compared with the rating values of the plate fin heat exchanger based on various correlations and by CFD analysis. The correlations used for Plate Fin Heat Exchanger design are:
    a. Correlation by Maiti-Sarangi [41].
    b. Correlation by Manglik-Bergles [28].
    c. Correlation by Joshi-Webb [40].

CHAPTER 2

LITERATURE REVIEW

2 LITERATURE REVIEW

Fehle et al. [1] found the heat transfer behavior of the compact plate fin heat exchanger by applying holographic interferometry. For this, he enabled a non-invasive and inertialess visualization of the temperature field. Then from the constant temperature line at the wall of the temperature field, he determined the local Nusselt number. For determining the effect of corner radii of the heat metal sheets, they first investigated the transfer of heat in plane fin arrangements. They also examined the increase of turbulence by the effect of circular segments in the inclined, non-staggered and staggered arrangements. After the determining the average Nusselt number they got the conclusion that the rate of heat transfer is highest in the in the non-staggered geometry and the staggered geometry achieved the best volume goodness factor. **Ranganayakulau et al. [2]** analyzed the cross flow plate fin, cross flow tube fin, parallel plate fin and the counter-flow plate fin heat exchangers by taking the effect of heat conduction in the longitudinal direction across the heat exchanger wall by using finite element method. They observed that the performance declination of cross flow type heat exchangers is higher compared to the counter-flow and parallel flow heat exchangers for all the cases. This occurrence of the distribution of temperature is two dimensional.

Sanaye and Hajabdollahi [3] applied ε-NTU method for the thermal modeling of the compact heat exchanger. By applying ε-NTU method, they calculated the effectiveness and the pressure drop of the heat exchanger. The fin pinch, height of the pin, offset length of the fin, flow length of the cold stream, no-flow length and flow length of the hot stream are considered as the six design parameters. They used non-dominated sorting genetic algorithm (NSGA-II) for getting the maximum effectiveness and minimum total yearly cost as two objective functions.

Rao and Patel [4] thermodynamically optimized the cross flow PFHE by using particle swarm optimization (PSO). The reduction of entropy generation for a given space restriction for specific heat duty, reduction of total volume and reduction of total unwanted cost are the basic objective functions, which are treated individually. **Hajabdollahi at el. [5]** presented a thermal model and optimally designed a compact heat exchanger. Fin pitch, the height of the fin, flow length of the cold stream, no-flow length and flow length of the hot stream are the five deign parameters. By using CFD analysis along with ANN (Artificial Neural Network), they developed a relation between the Colburn factor (J) and Fanning friction factor (f). **Zhang et al. [6]** examined evaporative mist pre-cooling, deluge cooling and combined cooling schemes for enhancing the performance of the heat exchanger. **Pingaud et al. [7]** performed both the steady state simulation and the dynamic simulation of the plate fin exchanger. By using the modeling based on mass balance, momentum balance and energy balance, they developed an algorithm for multi-fluid and multi passage PFHEs for both the transient and steady-state simulations. For treating the counter current flow and for solving the model equations involving the partial differentiation, they adopted an integration scheme which is implicit in nature. **Menzel and Hecht [8]** found that the heat exchanger performance decreases by the slog flow reversal, two-phase up-flow in PFHEs and they liquid logging involved in the heat exchanger surfaces. So, they adopted a special care by selecting wide boiling range fluid mixtures which evaporates at high NTU values at relatively low gas fluxes. Flow condition of non-refluxes needs a large gas mass flux, thus a higher pressure drop.

Dubrovsky [9] investigated a new convective heat transfer augmentation law for PFHE surfaces. This is characterized by $(\zeta/\zeta_{sm}) \leq (Nu/Nu_{sm})$ with the comparison to the vortex promoted heat transfer surfaces having an equal small channel at the same Reynolds numbers. Dubrovsky developed heat exchanger cores with three different fin surfaces. Out of the three, two of the surfaces are sort equilateral triangular and rectangular cross-sectional offset channels and the third surface possess isosceles triangular cross-sectional channels with transverse grooves which have projections along the direction of channel length.

Ranganayakulau et al. [10] analyzed the effect of non-uniform flow distribution of two-dimensional inlet fluids of both the cold and hot sides of the fluid in a cross-flow PFHE using finite element method. They developed mathematical equations for different mal-distribution models for various types of fluid flow, for the effectiveness of the exchanger and its declination due to the flow non-uniformity for all ranges of design and operating conditions. **Kundu and Das [11]** determined the optimum dimensions of the fin for the fin tube type heat exchangers. They found the maximum heat dissipation for any given value of thickness or pitch length for a fixed fin volume. Averous et al. (1999) presented a dynamic simulation for a brazed plate heat exchanger.

Ranganayakulau et al. [12] analyzed the cross flow compact PFHE, accounting the combined effects of 2D heat conduction in the longitudinal direction through the wall of the exchanger, non-uniform flow of the inlet fluid and the temperature distribution using the finite element method. They observed that the declination of the thermal performance promotes the elimination of each other in the areas of higher NTU but promote each other in the areas of lower NTU, considering the heat conduction in the longitudinal direction, non-uniformity of the flow & the non-uniformity of the temperature. **Picon-Nunez et al. [13]** designed compact plate fin heat exchangers where main objective was the utilization of the pressure drop. Based on the performance of the volume performance index (VPI) the surface is selected. The PFHE sizing requires the specification of the type of the surface on each side of the streams that takes part in the process of heat transfer. So, the small exchanger volume was the basic design objective. They concluded that by utilizing the steam pressure drop & making highly thermal efficient heat transfer surfaces, this can be achieved.

Wen & Li [14] analyzed the performance on the flow distribution of the fluid in the PFHE headers. They found that the fluid flow mal-distribution is very serious at the header length direction in the industrial used convectional headers due to poor header configuration. They also found that by using baffles the flow absolute parameter can be reduced. **Kim et al. [15]** investigated an experimental study for the heat transfer characteristics of a plate fin & tube fin heat exchangers by using the method of lumped capacitance based on liquid crystal thermography. **Sahin et al. [16]** proposed a model to know the flow behavior within plate fin & tube fin heat exchangers where they are made of two parallel plates having a single cylinder which is located between the plates. They did their experiment for duct height to cylinder dia. ratio of 0.365 with the *Re* value 4000-7500.

Peng and Ling [17] presented used of ANNs (artificial neural networks) for the prediction of pressure drop & heat transfer characteristics in the plate fin heat exchangers. The estimated result indicates that ANN models can be used for providing satisfactory estimations of both Colburn factor (j) & friction factor (f) in PFHEs. **Li-Zhi Zhang [18]** investigated the mal-distribution of the flow and the declination of the thermal performance in the cross flow air to air heat exchangers. **Zhang et al. [19]** developed a universal 3D distributed parameter model (DPM), for the evaluation & prediction of the steady behavior of PFHE. The model is able to satisfy the calculation for humid air in both wet & dry conditions. By using the DPM, they got the outlet temperature of the hot fluid of the exchanger under the wet condition is lower compared to the value that is obtained in the dry conditions. **Wang et al. [20]** experimentally studied the flow distribution of two-phase fluid in a PFHE under various operating conditions. They found that the flow distribution of the two-phase fluid is much more non-uniform & complex compared to that of the single phase flow. They also got that the non-uniformity distribution of the liquid phase declines with the reduction in the value of Re_{gas} and Re_{liq}. And for the gas phase distribution uniformity reduces with Re_{liq} but increases with Re_{gas}.

Yousfei et al. [21] optimized a cross flow plate fin heat exchanger using an imperialist competitive algorithm (ICA). The minimization of the total annual cost & the total weight is the main objective. The length of the heat exchanger at the hot & cold side, height of the fin, frequency, thickness of the fin, strip length of the fin and the number of the hot side layer are the seven design parameters. **Goyal et al. [22]** presented a model for multi-stream plate fin heat exchanger for the applications in the cryogenic field, using finite volume analysis. They discretized the core of the heat exchanger both in the transverse as well as in the axial direction. The heat conduction that takes place axially through the metal matrix of the heat exchanger in those cases, heat leaks to the surrounding & the effects of the metal matrix conductivity are considered. **Feru et al. [23]** presented the modelling and model validation for a modular two-phase heat exchanger that recovers energy in heavy duty diesel engines. They developed this model for temperature and vapour quality production and for control design of the waste heat recovery system. This waste heat recovery system recovers energy both from the main exhaust line and from the exhaust gas circulation line.

Nagarajan et al. [24] developed a novel fin configuration for high-temperature ceramic plate fin heat exchanger using the 3D computational fluid dynamics (CFD) fluent code. They performed the numerical analysis for different types of fins & compare their results with the selected design. They studied the fluid flow, heat transfer pressure drop and the properties like Nusselt number, friction factor and Colburn factor for various fin configurations. **Jeong et al. [25]** developed a new shape of plate fin heat exchanger by creating holes & ceases on the fin. This is used in construction machinery under the poor environment with very high dust content & where the extraneous materials are produced. They comparatively analyzed the flow and the heat transfer phenomena on the heat transfer surface for the plate fin heat exchanger, louver fin heat exchanger and the newly proposed shape heat exchanger. **Guo et al. [26]** experimented for preventing the fluid leakage from the adjacent channel walls of the plate fin heat exchanger. For achieving the maximum heat transfer performance, they used a genetic algorithm (GA) for optimized along with the Monte-Carlo algorithm method for getting the suitable solution.

Taler and Oclon [27] presented a method for the calculation of the mean value of thermal contact resistance for a plain fin & tube fin exchanger by using experimental measurements & CFD simulations. It is necessary to know the thermal contact resistance to predict the total gas side temperature difference. **Manglik and Bergles [28]** investigated on the thermodynamic & the hydraulic design tools for compact heat exchanger of rectangular offset strip fin and the effect of convection related to this. They reanalyzed the *j* and *f* data for the actual cores and also identified the asymptotic behaviour in the extreme laminar and fully turbulent flow regimes. They correlated the asymptotes obtained by the power law expressions in terms of the Reynolds number (Re) and by the dimensionless geometric parameters α, δ and γ. Finally, they developed equations for *j* and *f* data for laminar, turbulent and transition flow regimes.

Mishra et al. [29] developed an optimization technique based on genetic algorithm. The minimization of the entropy generation for a given heat duty under a specified space restrictions is the main objective. **Bhowmik and Kwan-Soo Lee [30]** used a steady state 3D numerical model for studying the pressure drop and heat transfer characteristics of an offset strip fin heat exchanger. Water is taken as the heat transfer medium & they considered Reynold's number Re_{dh} ranged from 10 to 3500. They observed the fanning friction factor (*f*) variation and Colburn heat transfer factor (*j*) variation relative to Re_{dh}. Then, they developed correlation for the *f* and *j* factors and analyzed the heat transfer and fluid flow characteristics of offset strip fins in the laminar, turbulent and the transient regions. Finally, they adopted three performance criteria i.e.,(j/f) , $(j/f)^{1/3}$ and *jf*. **Ismail et al. [31]** found that the calculated dimensional performance of heat transfer surfaces (*j* and *f* vs. Re) strongly controls the thermodynamic design of compact heat exchangers. They studied the flow patterns of 3 types of offset fins and 16 types of wavy fins by using the fluent software.

Saad et al. [32] developed a correlation for the friction factor of the micro-channel having offset-strip fin geometry with single phase for laminar to the turbulent flow of air & water. Then, they did a comparison with 3D CFD simulations. Also by using a high-speed camera, they studied the distribution of pressure drop in the two-phase regime. **Seara et al. [33]** analyzed a titanium brazed PFHE with serrated fins in liquid-liquid heat transfer process experimentally. They conducted the experiment by taking the water on both sides of the exchanger & determined heat transfer characteristics and the pressure drop. They also developed a correlation to determine the convectional heat transfer coefficient of a single phase fluid as a function of Reynolds number. **Yang and Li [34]** numerically investigated the heat transfer coefficients and the flow friction coefficients for offset strip fins used in plate fin heat exchangers & compared it with well-validated 3D models. Based on the analysis by CFD simulations, they developed correlations for the general prediction of *j* and *f* factors which excellently correlate a variety of geometrical parameters with the comparison to the existing correlations. Their predictions for offset strip fins with different fin thickness over a broad range of blockage ratio.

Suzuki et al. [35] studied the numerical and experimental analysis of the heat transfer and flow characteristics of a 2D flat plate arranged in a staggered manner used in mixed (free flow) convection region at low Reynolds number. **Youcef-Ali [36]** used the low thermos-physical characteristic of air to be used as heat transfer fluid in solar collectors. This promotes the thermal heat transfer between the fluid & the absorber plate. This enhances the thermal performance of the solar collector. **Fischedick et al. [37]** experimented on heat recovery of a ceramic PFHE being used for heat exchanger applications. **Peng et al. [38]** predicted the thermodynamic performance for longitudinal & transverse direction flow through the offset fins by a steady state, 3D numerical model.

2.1 Correlations for Offset strip channels

1. Correlation by Kays [42]:

   Colburn factor = $j = 0.665\,\mathrm{Re}_1^{-0.5}$ and

   Friction factor = $f = 0.44\left(t/l\right) + 1.328\,\mathrm{Re}_1^{-0.5}$

   But, for $\mathrm{Re} < 1000$:

   Colburn factor = $j = 0.483\left(1/D_h\right)^{-0.162}\left(s/h\right)^{-0.184}\mathrm{Re}^{-0.536}$ and

   Friction factor = $f = 7.661\left(1/D_h\right)^{-0.384}\left(s/h\right)^{-0.092}\mathrm{Re}^{-0.712}$

2. Joshi-Webb Correlation [40]:

   For laminar range $\left(\mathrm{Re} \le \mathrm{Re}^*\right)$,

   Colburn factor = $j = 0.530\left(\mathrm{Re}\right)^{-0.50}\left(l/D_e\right)^{-0.150}\left(s/h\right)^{-0.14}$ and

   Friction factor = $f = 8.120\left(\mathrm{Re}\right)^{-0.740}\left(l/D_e\right)^{-0.41}\left(s/h\right)^{-0.02}$

   For turbulent range $\left(\mathrm{Re} \ge \mathrm{Re}^* + 1000\right)$,

   Colburn factor = $j = 0.210\left(\mathrm{Re}\right)^{-0.4}\left(l/D_e\right)^{-0.24}\left(t/D_h\right)^{0.02}$ and

   Friction factor = $f = 1.120\left(\mathrm{Re}\right)^{-0.360}\left(l/D_e\right)^{-0.65}\left(t/D_e\right)^{0.17}$

The critical Reynolds number for both heat transfer and pressure drop considerations is given by:

$$\mathrm{Re}^* = 257\left(l/s\right)^{1.23}\left(t/l\right)^{0.58} D_e\left[t + 1.328\left(\frac{\mathrm{Re}}{l.D_e}\right)^{0.5}\right]^{-1}$$

The free flow area & the heat transfer area of offset strip fin given by Joshi-Webb are:

Free flow area = $A_c = (s-t)h$ and

Heat transfer area = $A = 2(sl + lh + th)$

So, the hydraulic diameter is given by, $D_h = \dfrac{2h(s-t)}{(sl + lh + th)}$

$$Since,\ Hydraulic\ Dia. = D_h = \frac{4\times A_c}{P} = \frac{4\times Free\ flow\ volume}{Total\ heat\ transfer\ area} = \frac{4\times A_c}{A/l}$$

3. Maiti-Sarangi Correlation [41]:

For laminar range $\left(\mathrm{Re} \le \mathrm{Re}^*\right)$,

Colburn factor = $j = 0.360(\mathrm{Re})^{-0.510}\left(h/s\right)^{-0.275}\left(l/s\right)^{-0.270}\left(t/s\right)^{-0.063}$ and

Friction factor = $j = 4.67(\mathrm{Re})^{-0.70}\left(h/s\right)^{-0.196}\left(l/s\right)^{-0.181}\left(t/s\right)^{-0.104}$

For turbulent range $\left(\mathrm{Re} \ge \mathrm{Re}^*\right)$,

Colburn factor = $j = 0.180(\mathrm{Re})^{-0.420}\left(h/s\right)^{0.288}\left(l/s\right)^{-0.184}\left(t/s\right)^{-0.05}$ and

Friction factor = $j = 0.320(\mathrm{Re})^{-0.286}\left(h/s\right)^{0.221}\left(l/s\right)^{-0.185}\left(t/s\right)^{-0.023}$

The critical Reynolds number for heat transfer is given by:

$$\mathrm{Re}^* = 1568.580\left(h/s\right)^{-0.217}\left(l/s\right)^{-1.433}\left(t/s\right)^{-0.217}$$

The critical Reynolds number for pressure drop is given by:

$$\mathrm{Re}^* = 648.230\left(h/s\right)^{-0.06}\left(l/s\right)^{-0.1}\left(t/s\right)^{-0.196}$$

The free flow area & the heat transfer area of offset strip fin given by Maiti-Sarangi correlation are same as that given by Joshi-Webb i.e.

Free flow area = $A_c = (s-t)h$ and

Heat transfer area = $A = 2(sl + lh + th)$

So, the hydraulic diameter is given by, $D_h = \dfrac{2h(s-t)}{(sl+lh+th)}$

$$Since,\ Hydraulic\ Dia. = D_h = \frac{4\times A_c}{P} = \frac{4\times Free\ flow\ volume}{Total\ heat\ transfer\ area} = \frac{4\times A_c}{A/l}$$

4. Manglik-Bergles Correlation [28]:

$$Colburn\ factor = j = 0.6522(\mathrm{Re})^{-0.5403}\left(s/h\right)^{-0.1541}\left(t/l\right)^{0.1499}\left(t/s\right)^{-0.0678}\times$$
$$\left[1+\left(5.269\times10^{-5}\right)\mathrm{Re}^{1.340}\left(s/h\right)^{0.504}\left(t/l\right)^{0.546}\left(t/s\right)^{-1.055}\right]^{0.1}$$

$$Friction\ factor = f = 9.6243(\mathrm{Re})^{-0.7422}\left(s/h\right)^{-0.1856}\left(t/l\right)^{0.3053}\left(t/s\right)^{-0.2659}\times$$
$$\left[1+\left(7.669\times10^{-8}\right)\mathrm{Re}^{4.429}\left(s/h\right)^{0.920}\left(t/l\right)^{3.}\left(t/s\right)^{0.236}\right]^{0.1}$$

The free flow area & the heat transfer area of offset strip fin given by Manglik-Bergles are:

Free flow area = $A_c = s.h$ and

Heat transfer area = $A = 2(sl + lh + th) + ts$

So, the hydraulic diameter is given by, $D_h = \dfrac{4shl}{2(sl+lh+th)+ts}$

$$Since,\ Hydraulic\ Dia. = D_h = \frac{4\times A_c}{P} = \frac{4\times Free\ flow\ volume}{Total\ heat\ transfer\ area} = \frac{4\times A_c}{A/l}$$

CHAPTER 3

EXPERIMENTAL SET-UP

3 THE EXPERIMENTAL SET-UP

In the experimental set-up, a counter-flow plate fin heat exchanger having offset strip fin surface is the major component. In the set-up, the compressor draws the cold air from the surrounding and forces this air to flow through a channel whereas hot air coming from that of a heating unit is made to flow across the second channel in a counter direction of flow. The experimental set-up includes a layout of the experimental set-up, different components of the set-up, principles of measurement and the procedure for the calibration of the instruments.

3.1 Experimental set-up and operations

The experimental rig consist of the heat exchanger core, air supply system, heating unit and the measurement/instrumentation system as shown in the figure 13. In the experiment, the fluids are flowing in the counter flow directions. The working fluid for this experiment is air. A screw compressor is used for the continuous supply of dry air to the plate fin heat exchanger. For the regulation of the flow rate, a control valve is used. The inlet of the cold air is at the bottom of the heat exchanger. When the cold air passes through the heat exchanger it gets heated by the transfer of heat from the hot air to the cold air. Then, the air comes out of the heat exchanger & again gets heated when passing through the heating unit. Here the inlet of the hot air which comes out from the heater is supplied to the heat exchanger from the top end. At the inlet of the hot air i.e., at the inlet to the hot air, a valve is used for controlling the mass flow rate of the hot air. When both the fluids have equal mass flow rates, the bypass valve is closed.

The inlet and outlet pressure of both hot and cold fluids are measured by the pressure gauges. For measuring the pressure drop across the heat exchanger, at both the upstream & downstream of the heat exchanger pressure taps are used. The U-tube manometer that connected to the pressure tap gives the pressure drop. RTDs (Resistance temperature detectors) are used to measure the inlet & outlet fluid temperatures.

At the outlet of the heat exchanger, a rotameter is installed to measure the flow rate for the balanced flow. Orifice meters are used to measure the flow rate of both hot and cold fluids, for the unbalanced flow. The rotameter is also used for the calibration of the orifice meter when desired.

For resisting the heat loss from the system to the surrounding, the test section is thoroughly insulated by using polystyrene foam or thermocouple sheets and glass wool. On the outer surface of the insulation, a resistance temperature detector is placed to indicate the temperature difference. From this temperature difference, heat loss to the surrounding can be calculated.

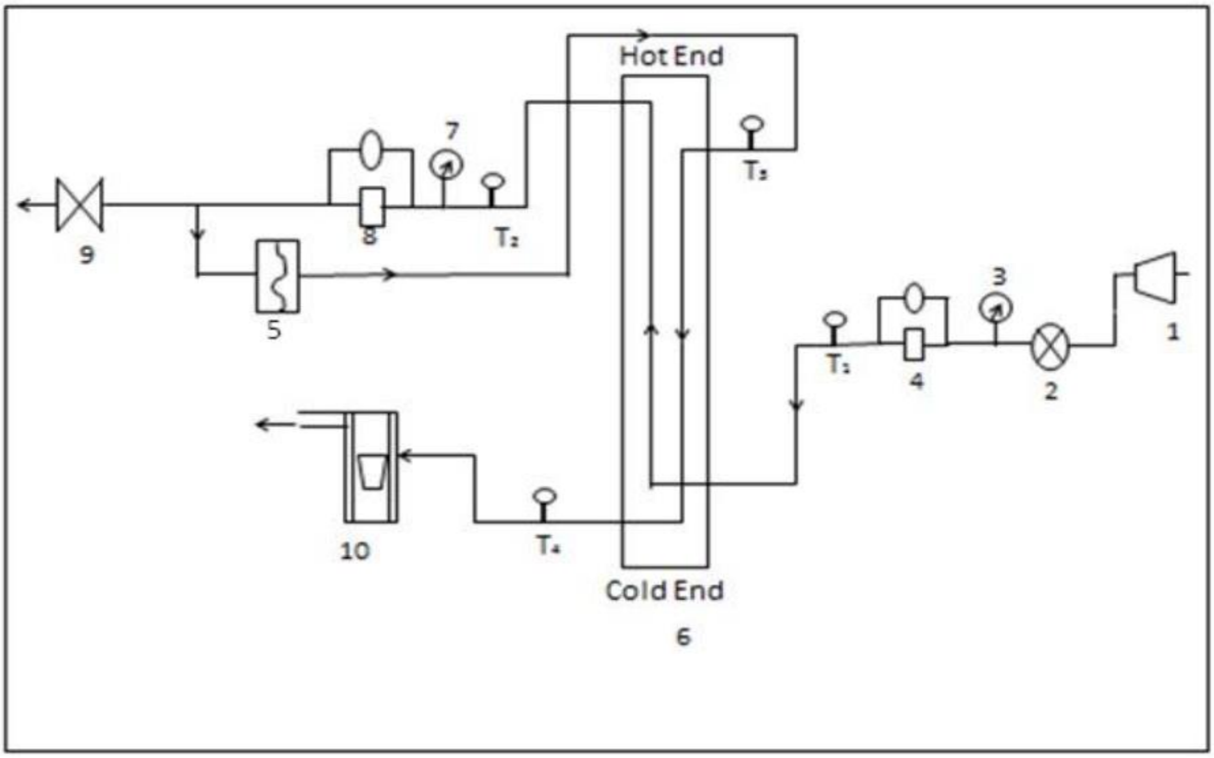


Figure 13. Schematic diagram showing Piping &Instrumentation of the experimental test rig

Table 1. Components shown in the Piping &Instrumentation of the experimental test rig

1-Compressor	2-Control Valve	3, 7- Pressure Taps
4, 8- U-tube manometer	5- Heater	6- Test Section
T_1, T_2, T_3, T_4 are RTDs	9- Bypass Valve	10- Flowmeter

The volume flow rate of the fluid across the test section is fixed at the desired value and this flow can be perceived by a rotameter placed at the outlet of the heat exchanger, at the balanced condition. Initially, at a low value of the variac is fixed & then increased progressively depending on the hot inlet temperature. By adjusting the wattage of the variac, the hot air temperature at the inlet of the heat exchanger is maintained at a desired temperature. This system is let to run till the study state condition is achieved. Then, by using RTDs the inlet & exit temperature of both fluid are measured. From the U-tube mercury manometer, the pressure drops of both the fluids are measured.

Figure 14. Picture of the experimental set-up.

3.2 Description of Equipment and Instruments

3.2.1 The offset strip plate fin heat exchanger

The counter flow offset strip plate fin heat exchanger is the major component of the test section. All the dimensions & arrangement of inlet and outlet parts are shown in the Table 4. The core data dimensions and the thermal data are given in the table 2 and 3. The fins of this offset strip plate fin heat exchanger are made of aluminium. It is manufactured by sandwiching the layers of the plates in which the cold stream flows in between the outer layers which are at a higher temperature. These offset strip PFHXs have a vast application in the cryogenic field.

Table 2. Details of the flow arrangement in the heat exchanger

	High Pressure Side (Hot Side)	Low Pressure Side (cold Side)
Fin	Offset strip fin	Offset strip fin
Number of passages	5	4
Number of passes	1	1
Flow rate	Counter Flow	Counter Flow

Table 3. All dimensions of the plate fin heat exchanger

Length of the core	0.9 m	Total length	1.0 m
Width of the core	0.073 m	Total width	0.085 m
Height of the core	0.093 m	Total height	0.105 m

Table 4. Fin dimensions used in the heat exchanger

	Fin geometry	**High pressure Side**	**Low pressure Side**
01	Fin frequency, f	714 fins per metre	588 fins per metre
02	Length of the fin, l	0.003 m	0.005 m
03	Thickness of the fin, t	0.0002 m	0.0002 m
04	Height of the fin, h	0.0093 m	0.0093 m
05	No. of layers	05	04

3.2.2 Twin Screw compressors

A positive displacement type screw compressor is used for the supply of air to the heat exchanger. Since, it is flooded compressor, the lubricating oil flows inside the rotors & provides a hydraulic sealing. The gas enters through the suction side & gets compressed with the rotation of the screws. In comparison to other positive displacement type compressor, these compressors have a relatively high rotational speed. So, these are compact. The twin screw compressor also maintains high volumetric efficiency over a wide range of flow rates & operating pressures.

3.2.3 Heating device

The heating device consists of a number of heating elements enclosed in a shell. The cold stream enters from one side of the shell and then flows over the heating element & leaves the shell at the other end. The capacity of each heating tube is 1500W & there are 17 numbers of tubes within the shell.

3.2.4 Resistance Temperature Detector

The resistance of a conductor or a semiconductor changes with the change in temperature. Resistance Temperature Detectors, also known as RTDs, use this property to measure this temperature. Platinum, copper, lead, or indium wires are used in metallic RTDs. Non-metallic RTDs use GaAlAs diodes, carbon glass and ruthenium oxide. The schematic diagram of platinum RTD is as shown in the figure 15.

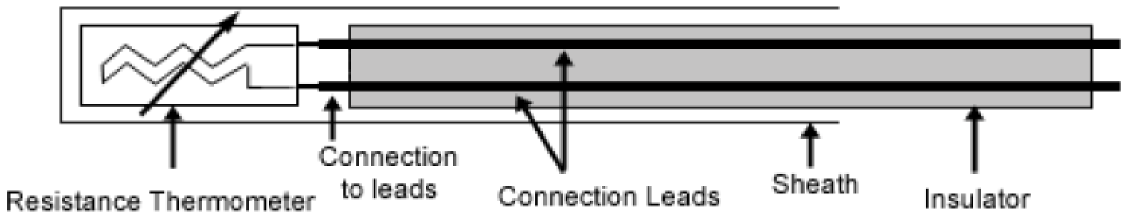


Figure 15. Schematic of platinum RTD

A conducting wire, say platinum wire of very long length is wound on a notched mica insulator. This assembly is housed inside a closed platinum sheath. An ohm meter is used to measure the resistance, thereby the corresponding temperature. The typical size of RTD is $3mm \times 1.84mm \times 0.98mm$. The choice of the wire material is dependent on the variation of the resistance (R_t/R_0) with temperature *(t)* where R_t and R_0 are resistance at any temperature $t\ ^0c$ and nominal temperature $0\ ^0c$ respectively. It is desired to choose a material whose resistance varies linearly with temperature. One of the correlations for RTD is

$$\frac{R_t}{R_0} = 1 + At + Bt^2 + ct^3\left(t - 100\right)$$

The constants A, B and C are found by the calibration of RTD at any three standard temperatures.

The unwanted mechanical and thermal strains cause a change in the electrical resistance. These changes induce an error in the measurement. These changes induce an error in the measurement. The effect of lead wire resistance is very crucial in the accuracy of RTDs. In order to minimize this error, two different wirings are used. They are:

- Two wire arrangement and
- Four wire arrangement

Some of the common used metallic RTDs are PT 100, PT 1000. PT 100 implies the sensor has 100 ohms resistance at $0\ ^0c$.

3.2.5 Orifice Meter

An orifice meter is used for the measurement of mass flow rate of the air through the pipes. Orifice meter works on the principle that a pressure difference between the two sections can be developed by reducing the cross-sectional area of the flow passage and from the measurement of the pressure difference, the discharge through the pipe can be determined.

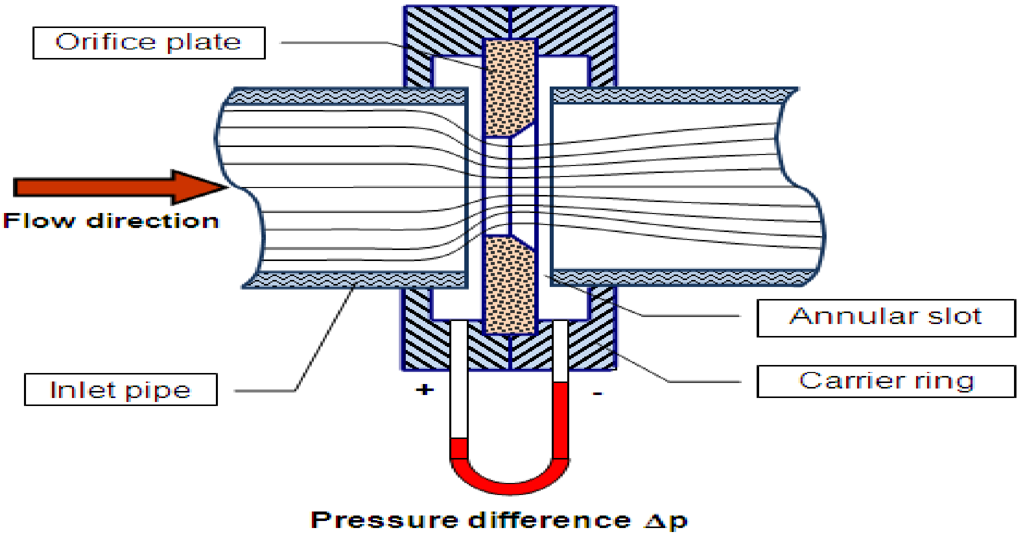


Figure 16. Schematic diagram of Orifice meter

An orifice meter is made up of a circular flat plate with a circular hole concentric with the pipe axis called orifice. The thickness of the plate is less than or equal to 0.005 times the diameter of the pipe. The edge thickness of the orifice from the upstream face of the plate is less than equal to 0.02 times of the pipe. For the remaining thickness of the plate, it is bevelled with the bevel angle lying between 30° to 45°.

The volume flow rate, Q can be calculated by $Q = CC_d\sqrt{2gh_a}$

Where C = Area constant = $\dfrac{A_2}{\sqrt{1-\left(A_1/A_2\right)^2}}$

C_d = Coefficient of discharge and

h_a = head of air = $h_w\left(\dfrac{\rho_w}{\rho_a}-1\right)$

The mass flow rate = $\dot{m} = \rho_a \times Q$

3.2.6 Autotransformer or Variac

An autotransformer (or Variac) is a single winding electrical transformer. It is single coil acting on itself. The same winding acts both as the primary as well as the secondary. There are at least three taps where electrical connections are made. The step-up & the step-down voltages of an autotransformer in India is 220-230 or 220-240 volt range. The function of the autotransformer is to regulate the voltage of the heating element to achieve the desired temperature of the heating unit.

CHAPTER 4

DESIGN OF PLATE FIN EXCHANGER

4 DESIGN OF THE PLATE FIN HEAT EXCHANGER

The basic design considerations of a plate fin heat exchanger include:

i. Process & design specifications
ii. Hydraulic & thermal design
iii. Mechanical design
iv. Manufacturing considerations
v. Trade-off factors and system based optimization

i. Process & design specifications:

In heat exchanger design, the process & problem specification is one of the most important steps. Any specification for process and design procedure counts all the required information for designing and optimizing the exchanger for a particular application. It includes:

- Specification of the problem for operating conditions
- Type of heat exchanger
- Type of flow arrangement
- Materials
- Considerations for design/manufacturing/operation
- Information on the minimum input specifications

The selection of design conditions is the first and important consideration. Then, the next is the off-design and design point condition. The specification for operating conditions and the operating environment should be mentioned which includes:

- ➢ Mass flow rates
- ➢ Fluid types and their thermos-physical properties
- ➢ Inlet temperature & pressure of both fluid streams
- ➢ Maximum allowable pressure drop on both fluid sides
- ➢ Inlet temperature & pressure fluctuation due to variation in the process or environmental parameters
- ➢ Corrosiveness
- ➢ Fouling characteristic of fluids and
- ➢ Operating environment

The heat exchanger specification includes heat exchanger includes exchanger construction type, flow arrangement, core geometry, & fin type. For compact plate fin heat exchangers, one can choose offset strip fin, louver fin or other fin geometry.

ii. Hydraulic & Thermal design:

The hydraulic and thermal design of the heat exchanger includes exchanger rating & exchanger sizing. The two important relations that hold good for the entire thermal design procedure are:

1. Enthalpy rate equation: $q = q_j = \dot{m}_j \, \Delta h_j$ (j= 1, 2…)
2. Heat transfer rate equation: $q = U.A.\Delta T_m$

4.1 Rating problem of heat exchanger

Rating problem is the calculation of pressure drop & heat transfer of an already sized heat exchanger or an existing heat exchanger. The rating problem is also known as performance problem or simulation problem. The inputs to the rating problem are:

(a) Heat exchanger construction (b) flow arrangement (c) overall dimensions (d) Both side surface geometries (e) pressure drop and heat transfer characteristics (f) mass flow rate (g) inlet temperature and (h) fouling factor

The parameters to be determined are:

(a)The fluid inlet temperature, (b) the total rate of heat transfer & the pressure drop on each side of the heat exchanger.

The calculation of various surface geometrical properties of the heat exchanger is the first step in the rating procedure. Our project on plate fin heat exchanger with offset strip fin geometry is described by the following parameters:

(i) Spacing between the fins (*s*) (excluding thickness of fin)

(ii) Height of the fin (*h*), (excluding thickness of fin)

(iii) Thickness of the fin(*t*) and

(iv) Strip length of the fin (*l* or L_f)

The basic geometric dimensions are shown by a schematic view of the rectangular type offset strip fin surface (Fig.17) but the present PFHE has different fin geometries for the cold and hot side fluids.

The fin specification for hot and cold side of the heat exchanger is shown in the Table 5.

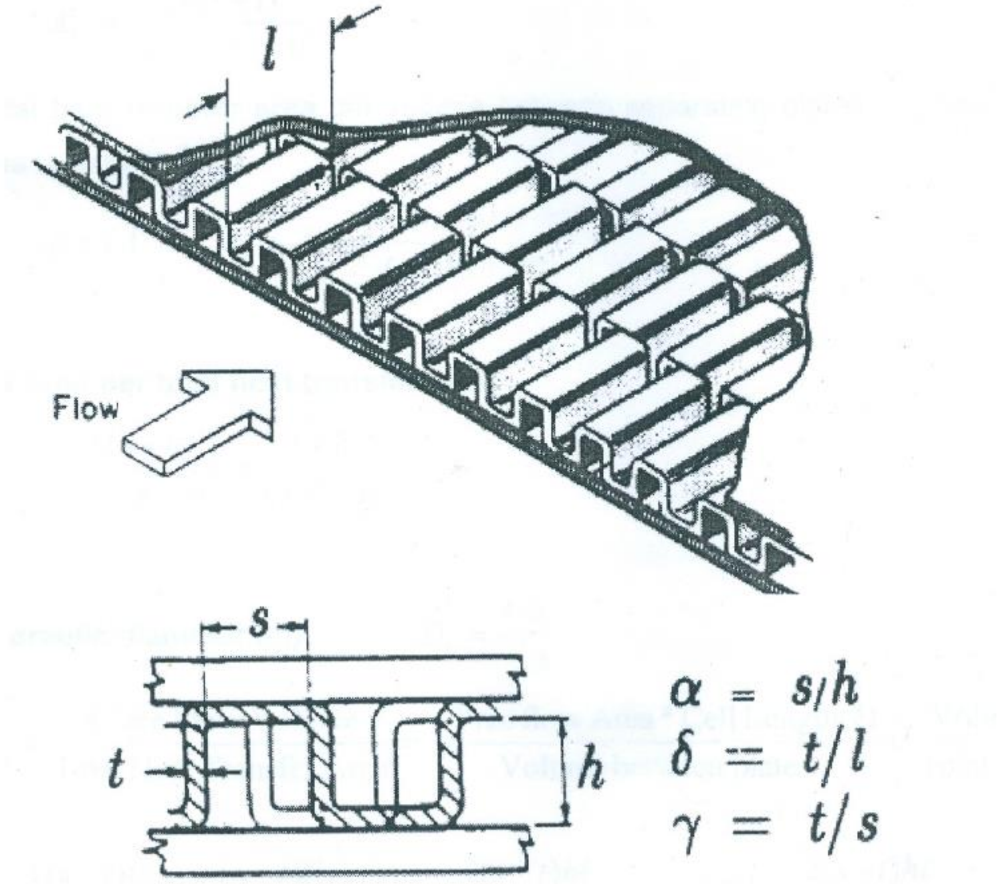


Figure 17. A typical offset strip surface geometry (Shah and Sekulic[39])

Table 5. Fin dimensions used in the heat exchanger

	Fin geometry	High pressure Side	Low pressure Side
01	Fin frequency, f	714 fins per metre	588 fins per metre
02	Length of fin, l	0.003 m	0.005 m
03	Thickness of the fin, t	0.0002 m	0.0002 m
04	Height of the fin, h	0.0093 m	0.0093 m
05	No. of layers	05	04

I. Calculation of secondary fin geometries

From the given basic fin geometries, the secondary geometrical parameters can be derived. The following calculation is done for the hot side only. The calculation can also be done on the cold fluid side by following the same steps:

(i) Fin spacing, $\mathrm{s} = \frac{[1-(f \times t)]}{f} = 0.012 \ m$

(ii) $\frac{Free\ flow\ area}{Frontal\ area} =$

$$\sigma = {a_{ff}}/{a_{fr}} = \frac{h(s-t)}{(s+t)(h+t)} = \frac{0.0093 \times (0.012 - 0.0002)}{(0.0012 + 0.0002)(0.0093 + 0.0002)} = 0.6992$$

(iii)Heat transfer area per fin,

$$a_s = 2hl + 2th + 2ls = (2 \times 0.0093 \times 0.003) + (2 \times 0.0002 \times 0.0093) + (2 \times 0.003 \times 0.0012)$$

$= 0.00006672 \ \ m^2$

(iv)Ratio of fin area to heat transfer area of fin,

$$= \frac{2(l+t)h}{2(hl+ls+th)} = \frac{2 \times (0.003 + 0.0002) \times 0.0093}{2(0.0093 \times 0.003) + (0.003 \times 0.0012) + (0.0002 \times 0.0093)} = 0.8920$$

(v) Equivalent Diameter, $D_e = \frac{(4 \times free\ flow\ area \times length)}{Heat\ transfer\ area}$

$$= \frac{2hl(s-t)}{(hl+ls+th)} = \frac{2 \times 0.0093 \times 0.003 \times (0.0012 - 0.0002)}{(0.0093 \times 0.003) + (0.003 \times 0.0012) + (0.0002 \times 0.0093)}$$

$= 0.001672662 \ m$

(vi) Gap between plates, $\mathrm{b} = t + h = 0.0095 \ m$

II. Heat transfer area calculation

The various heat transfer area in the hot side are calculated as follows:

(i) The high pressure side heat transfer area is calculated as follows:

Total area between the plates, $\mathrm{A}_{frh} = b \times N_h \times W = 0.00950 \times 5 \times 0.0730 = 0.0035 \ m^2$

(ii) Total free flow area, $\mathrm{A}_{ffh} = \sigma \times A_{frh} = 0.002425 \ m^2$

(iii)Wall conduction area on the hot side, $a_{wh} = A_{frh} - A_{ffh} = 0.0010430 \ m^2$

(iv)Total heat transfer area, $A_h = \frac{4 \times A_{ffh} \times L}{D} = \frac{4 \times 0.002425 \times 0.9}{0.00167266} = 5.215 \ \ m^2$

(v) Total wall conduction area $a_{wh} + a_{wc} = 0.001043 + 0.00069736 = 0.00174\ m^2$

Similarly, the cold side heat transfer areas and the free flow area can be calculated.

III. Heat exchanger input data

The hot gas inlet temperature = 368.80 K

The cold gas inlet temperature = 311.92 K

The inlet pressure of cold gas = 1.20 bar

The inlet pressure of hot gas = 1.170 bar

The cold gas mass flow rate = 0.0095 kg/s

The hot gas mass flow rate = 0.0095 kg/s

IV. Estimation of average temperature

The mean temperatures of the fluids have to be estimated first because the fluid outlet temperatures are unknown for the rating problem. So, by using the property package GASPAK, the fluid properties at the estimated mean temperatures of 344.15 K and 340.05 K for hot and cold fluid.

The properties of hot nitrogen gas at the mean film temperature are:

Specific heat, $C_p = 1040.8$ J/Kg K

Viscosity, $\mu = 0.0000199$

Prandtl number, $pr = 0.7170$

Density, $\rho = 1.046$ Kg/m^3

The properties of cold nitrogen gas at the mean film temperature are,

Specific heat, $C_p = 1040.7$ J/Kg K

Viscosity, $\mu = 0.0000197$

Prandtl number, $pr = 0.7170$

Density, $\rho = 1.076$ Kg/m^3

V. Heat Transfer coefficients and surface effectiveness of fins

The heat transfer coefficients calculations for the hot and cold gases are similar. The following calculations are presented for the cold nitrogen gas.

(i) The core mass velocity, $G = \frac{m_c}{A_{ffh}} = \frac{0.00577}{0.002425} = 2.38\ \frac{kg}{m^2 s}$

(ii) The Reynolds number, $\mathrm{Re} = \frac{GD_e}{\mu} = \frac{2.38 \times 0.001674}{0.0000197} = 202.3$

(iii) The critical Reynolds number for heat transfer coefficient is given as:

$$\mathrm{Re}^* = 1568.580\left(h/s\right)^{-0.2170}\left(l/s\right)^{-1.4330}\left(t/s\right)^{-0.2170}$$

$$= 1568.58(7.743)^{-0.217}(2.498)^{-1.433}(0.1665)^{-0.217} = 399.77$$

(iv) The Colburn factor, j (for $\mathrm{Re} > \mathrm{Re}^*$) is given by correlation proposed by Maiti and Sarangi as:

$$j_c = 0.180(\mathrm{Re})^{-0.420}\left(h/s\right)^{0.2880}\left(l/s\right)^{-0.1840}\left(t/s\right)^{-0.050}$$

$$= 0.180(202.298)^{-0.420}(7.743)^{0.2880}(2.498)^{-0.1840}(0.1665)^{-0.050} = 0.03685$$

(v) The convective heat transfer coefficient is given as:

$$h_c = \frac{j_c \times c_c \times G_c}{(pr)^{0.667}} = \frac{(0.0369 \times 1040.70 \times 2.380)}{(0.7169)^{0.667}} = 113.95 \ \frac{W}{m^2K}$$

(vi) The fin parameter is calculated as:

$$M = \sqrt{\frac{(2 \times h_c)}{(K_f \times t)}} = \sqrt{\frac{(2 \times 113.954)}{(170 \times 0.0002)}} = 81.87 \ m^{-1}$$

(vii) l_c = fin heights for the inner layers of the cold fluid =b/2= 4.75 *mm*

l_c = fin heights for outer layers of the cold fluid = b = 9.5 *mm*

l_h = fin heights for both inner and outer layers of the hot fluid = b/2 = 4.75 *mm*

(viii) The fin effectiveness is given by,

$$\eta_f = \tanh(Ml_c)/(Ml_c) = 0.9525$$

(ix) The overall surface effectiveness of fins on hot side is: $\eta_{oh} = 1 - \left(a_f/a_s\right) \times (1 - \eta_f) = (1 - 0.8656) \times (1 - 0.9626) = 0.9676$

$$\eta_{oc} = 1 - \left(a_f/a_s\right) \times (1 - \eta_{fi})\frac{(N_p - 2)}{N_p} - \left(a_f/a_s\right) \times (1 - \eta_{fo})\left(\frac{2}{N_p}\right)$$

$$= 1 - (0.8920) \times (1 - 0.9524)\frac{(5-2)}{5} - (0.8920) \times (1 - 0.8375)\left(\frac{2}{5}\right)$$

$= 0.9166$

VI. Overall heat transfer coefficient and NTU

The overall heat transfer coefficient is given as:

$$\frac{1}{\left(U_o A_o\right)_h} = \frac{1}{\left(\eta_{oh} h_h A_h\right)} + \frac{a}{K_w A_w} + \frac{1}{\left(\eta_{oc} h_c A_c\right)}$$

Where, lateral conduction area

$$A_w = W \times L \times \left(2N_p + 2\right) = 0.0730 \times 0.9\left(2 \times 4 + 2\right) = 0.657$$

$$\frac{1}{\left(U_o A_o\right)_h} = \frac{1}{\left(0.9676 \times 88.464 \times 3.452\right)} + \frac{0.0008}{170 \times 0.657} + \frac{1}{(0.9166 \times 113.954 \times 5.215}$$

$$= 0.00522 \ \frac{K}{W}$$

So, $\left(U_o A_o\right)_h = 191.32 \ \frac{W}{K}$

Overall heat transfer coefficient, $U_{oc} = \frac{\left(U_o A_o\right)}{A_{oc}} = \frac{191.32}{5.215} = 36.68 \ \frac{W}{m^2 K}$

Number of transfer units, $\mathrm{NTU} = \frac{U_o A_o}{C_{\min}} = \frac{191.32}{6.0084} = 31.84$

VII. Effectiveness of heat exchanger (neglecting longitudinal heat conduction)

Neglecting longitudinal heat conduction, the effectiveness of heat exchanger is given by:

$$\varepsilon = \frac{1 - e^{-NTU\left(1-C_r\right)}}{1 - C_r e^{-NTU\left(1-C_r\right)}} = \frac{1 - e^{-32.18956(1-0.9999)}}{1 - 0.9999 e^{-32.18956(1-0.9999)}} = 0.96989$$

VIII. Effect of longitudinal heat conduction

The longitudinal heat conduction results in the reduction of the effectiveness of heat exchanger. By using Kroeger's equation this reduction in the effectiveness of heat exchanger can be found.

(i) Wall conduction area, $a_w = 0.00175 \ m^2$

(ii) Fin conductivity, $K_w = 150 \ \frac{W}{mK}$

(iii) Wall conduction parameter, $\lambda = \frac{K_w a_w}{L C_{\min}} = \frac{150 \times 0.00174}{0.9 \times 25.883} = 0.0112$

(iv) $y = \lambda(NTU) C_r = 0.01120 \times 34.20 \times 0.9960 = 0.3817$

(v) $\gamma = \frac{(1-C_r)}{(1-C_r)(1+y)} = \frac{(1-0.9960)}{(1-0.996)(1+0.3817)} = 0.00145$

(vi) $\phi = \gamma \left(y \big/ (1+y)^{1/2} \right) \left[\frac{y(1+\gamma)}{1-\gamma y(1+\gamma)} \right] = 0.00029$

(vii) $\varphi = \frac{(1+\phi)}{(1-\phi)} = \frac{(1+0.001011)}{(1-0.001011)} = 1.000$

(viii) $r_1 = \frac{NTU(1-C_r)}{1+\lambda(NTU)C_r} = \frac{31.84(1-0.996)}{1+(0.05467\times 31.84\times 0.996)} = 0.001228$

(ix) $(1-\varepsilon) = \frac{(1-C_r)}{\varphi \exp(r_1)-C_r} = \frac{(1-0.996)}{1\times \exp(0.001228)-0.996} = 0.0761$

(x) $\varepsilon = [1-(1-\varepsilon)] = 0.9238$

This effectiveness is the actual effectiveness of heat exchanger considering longitudinal heat conduction. Based on this value of effectiveness the outlet temperatures of fluids are calculated as follows:

The outlet temperature of hot fluid is calculated by,

$$T_{ho} = T_{hi} - \frac{\varepsilon C_{\min}(T_{hi}-T_{ci})}{C_h} = 368.96 - \frac{0.9238\times 6.0084(368.96-315.24)}{6.0084} = 319.34\ K$$

The outlet temperature of cold fluid is calculated by,

$$T_{co} = T_{ci} + \frac{\varepsilon C_{\min}(T_{hi}-T_{ci})}{C_c} = 315.24 + \frac{0.9238\times 6.0084(368.96-315.24)}{6.0084} = 364.87\ K$$

Then, the mean temperatures of hot & cold fluid are calculated.

The mean temperature of hot fluid, $T_{hm} = \frac{T_{hi}+T_{ho}}{2} = \frac{368.96+319.34}{2} = 344.15\ K$

The mean temperature of cold fluid, $T_{cm} = \frac{T_{ci}+T_{co}}{2} = \frac{315.24+364.87}{2} = 340.05\ K$

IX. Estimation of pressure drop, Δp

The pressure drop in the cold fluid stream side is high & more complex. So, the pressure drop calculation for the cold fluid stream side is calculated here.

(i) For the pressure drop the critical Reynolds number is calculated

$$\mathrm{Re}^{**} = 648.230\left(h/s\right)^{-0.060}\left(l/s\right)^{0.1}\left(t/s\right)^{-0.1960}$$

$$= 648.23(7.7435)^{-0.06}(2.5)^{0.1}(0.1667)^{-0.196} = 892.77$$

(ii) For $\mathrm{Re}^{**} > \mathrm{Re}$, the friction factor is given by

$$f = 0.320(\mathrm{Re})^{-0.2860}\left(h/s\right)^{0.2210}\left(l/s\right)^{-0.1850}\left(t/s\right)^{-0.0230}$$

$$= 0.320(202.298)^{-0.2860}(7.743)^{0.2210}(2.5)^{-0.1850}(0.1665)^{-0.0230}$$

$$= 0.1731$$

(iii) The pressure drop, $\Delta p = \dfrac{4fLG^2}{2D_e\rho_b} = \dfrac{4\times 0.1732\times 0.9\times(2.3804)^2}{2\times 0.001674\times 1.07551} = 980.67\ pa$

Since, value of the pressure drop is less than the allowed pressure drop value which is 5 *kPa*, the design of the heat exchanger meets the hydraulic requirement.

CHAPTER 5

PERFORMANCE ANALYSIS

5 PERFORMANCE ANALYSIS

The performance parameters of a plate fin heat exchanger mainly include effectiveness, overall thermal conductance & pressure drop. A number of experiments have been performed at different mass flow rates & at different hot fluid inlet temperatures under balance conditions by S. Alur. We have compared these experimentally obtained results with Joshi-Webb correlation, Maiti-Sarangi, Manglik-Bergles correlation and by using CFD analysis.

5.1 Numerical analysis by CFD

The computational fluid dynamics is used for the prediction of fluid flows & heat transfer using the computation method.

1. Description to the problem & geometry:

In the present thesis, an offset strip fin plate fin heat exchanger is investigated numerically & is compared with the experimentally obtained results.

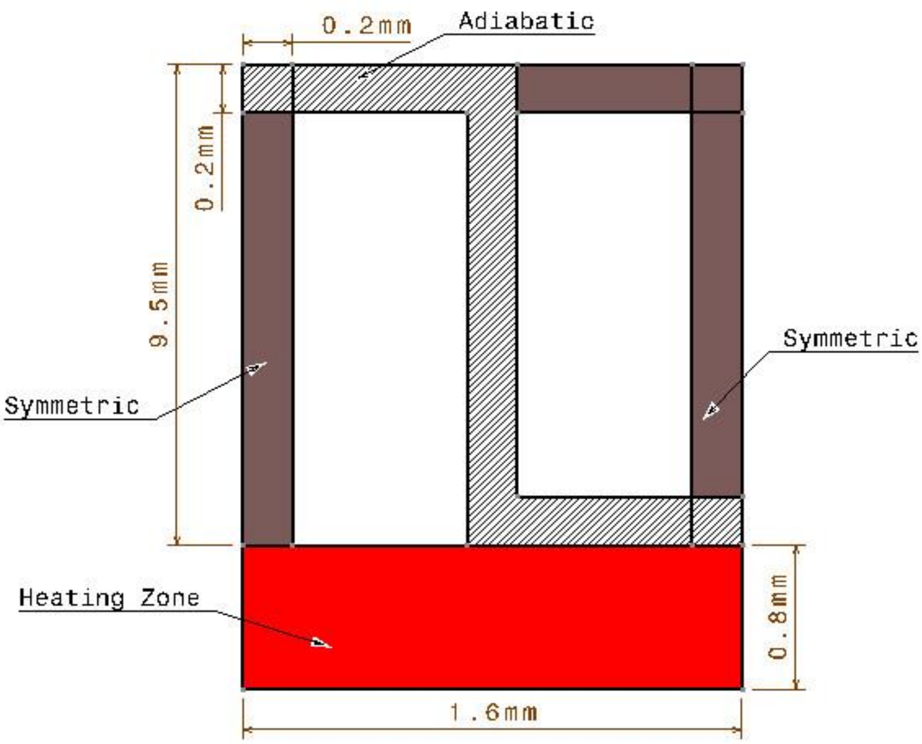


Figure 18. The geometry of the offset fin with the dimensions

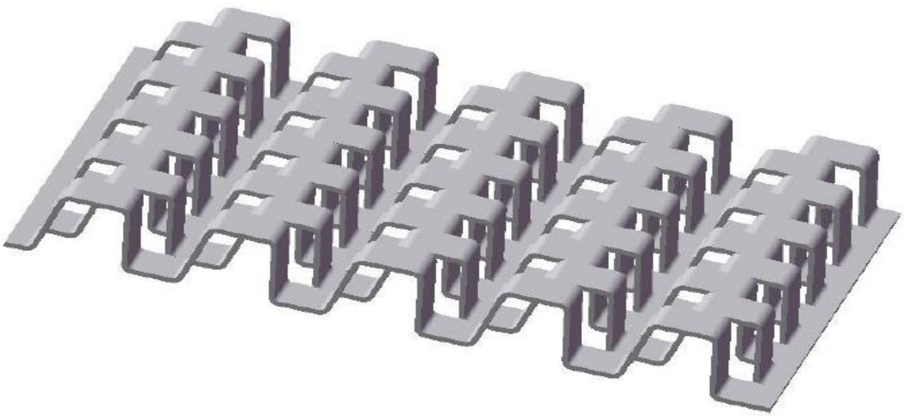


Figure 19. Offset strip fin geometry used in the heat exchanger

2. Materials properties

The fin material is made of aluminium or aluminium based alloys. Aluminium is widely used in offset strip fin heat exchangers because of its good thermos-physical characteristics, small density & relatively low price. The following constant thermos-physical properties of aluminum are used for the numerical calculations:

Density of aluminium= ρ_{Al} =2719 Kg/m^3

Specific heat of aluminium= $C_{p,Al}$ =871 J/Kg K

Thermal conductivity of aluminium=202.4 W/m K

The fluid that flows through the offset strip fin heat exchanger is air. The thermo-physical properties of air that taken for CFD calculation are:

Density of air = ρ_{air} =1.225 Kg/m^3

Specific heat of air= $C_{p,air}$ =1006.43 J/Kg K

Thermal conductivity of air= 0.0242 W/m K

Coefficient of viscosity= $\mu = 1.7894 \times 10^{-5}$ Kg/m s

3. Boundary conditions:

Boundary conditions are given to the physical models in order to solve the numerical problem.

- Inlet boundary condition: For the given offset strip fin surface, velocity is defined as the inlet boundary condition. The uniform velocity value is obtained from the

Reynolds number based on the hydraulic diameter of the offset strip fin and viscosity at a constant temperature of 333 K.

- Outlet boundary condition: The static pressure is defined as the outlet boundary condition.
- Symmetric boundary condition: The symmetric conditions means there is zero flux across the boundary. The symmetric boundary conditions are given to the side walls of the heat exchanger.
- Thermal boundary wall: At the bottom surface a constant heat flux of 20000 W/m^2 is applied.
- Adiabatic wall: The top surface of the offset- strip fin PFHX is adiabatically insulated from the heat flow.

The temperature contour of the offset strip plate fin heat exchanger for the given conditions is shown in the figure 20.

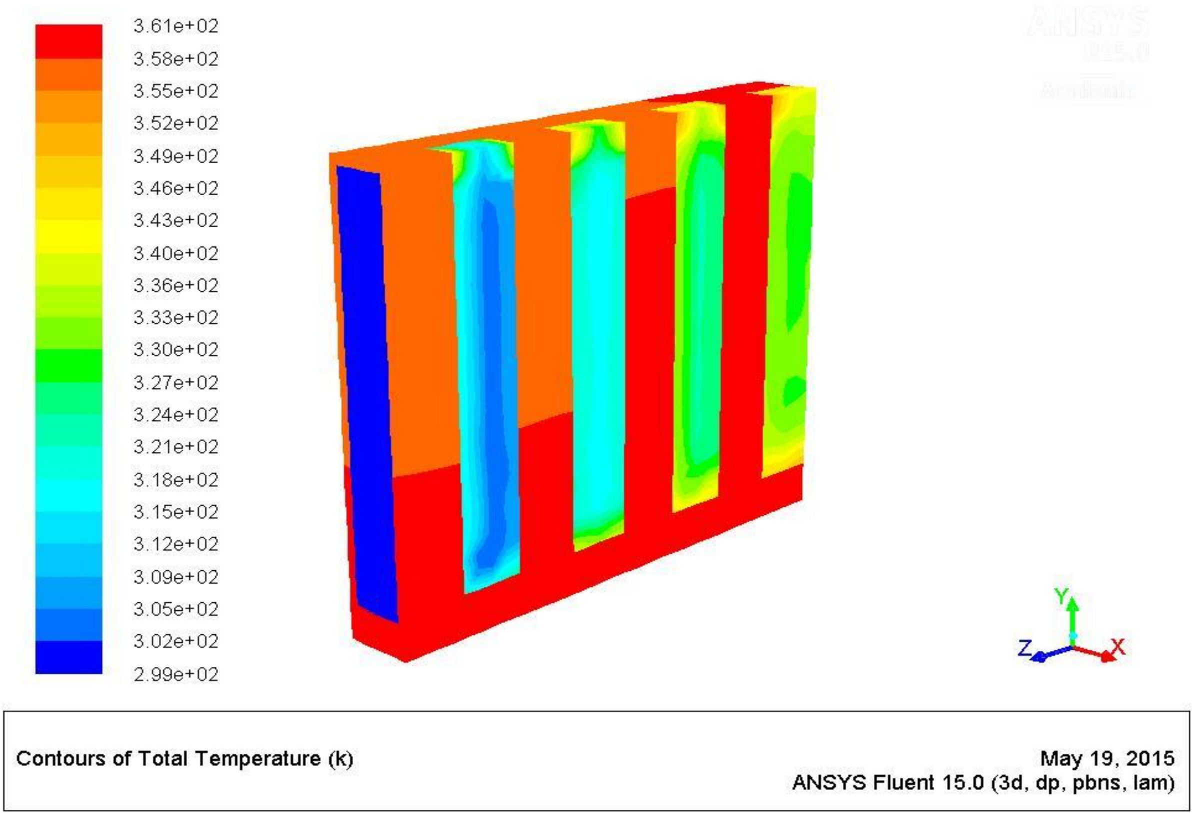


Figure 20. Temperature Contour of a offset strip fin

The experimentally obtained results are compared with the theoretical correlations e.g., Joshi-Webb correlation, Maiti-Sarangi correlation, Manglik-Bergles correlation & also compared with the plots obtained by the CFD analysis. The plots are shown below:

5.2 Variation of effectiveness with the mass flow rate

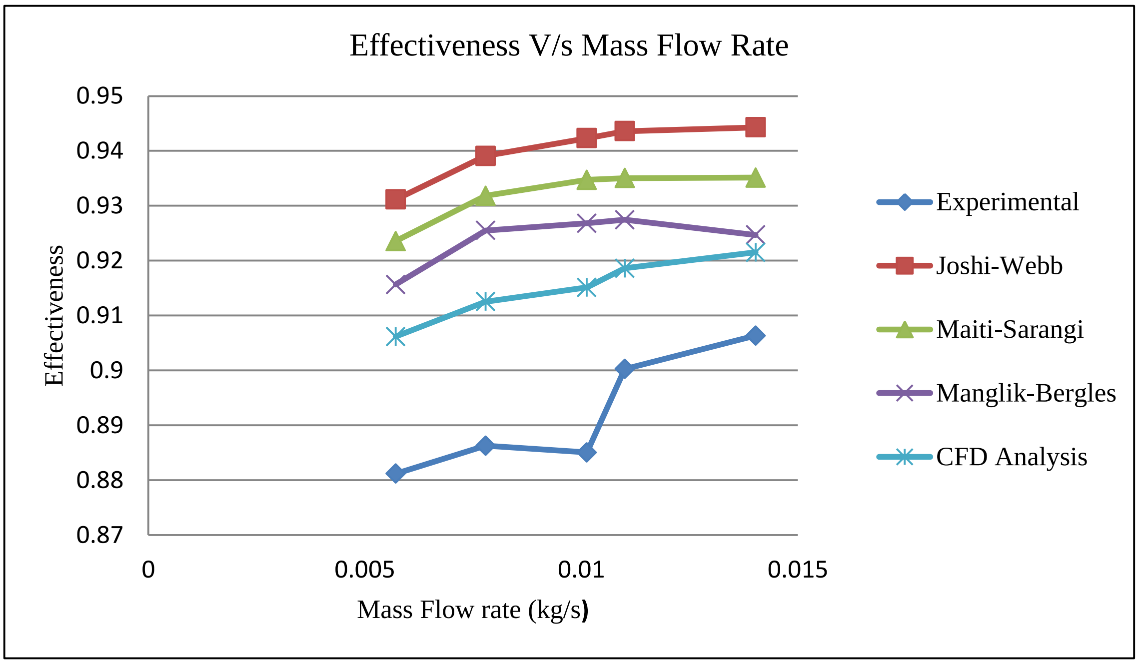


Figure 21. Effectiveness variation with the mass flow rate (at hot inlet temperature of 66 ℃ or 339 K)

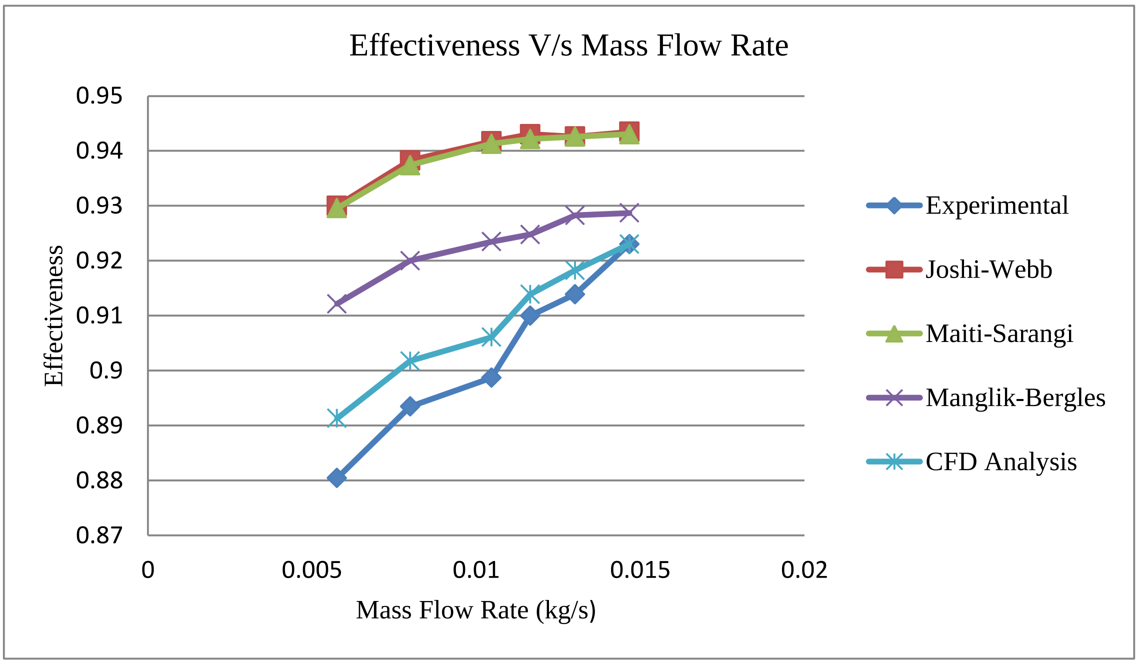


Figure 22. Effectiveness variation with the mass flow rate (at hot inlet temperature of 86 ℃ or 359 K)

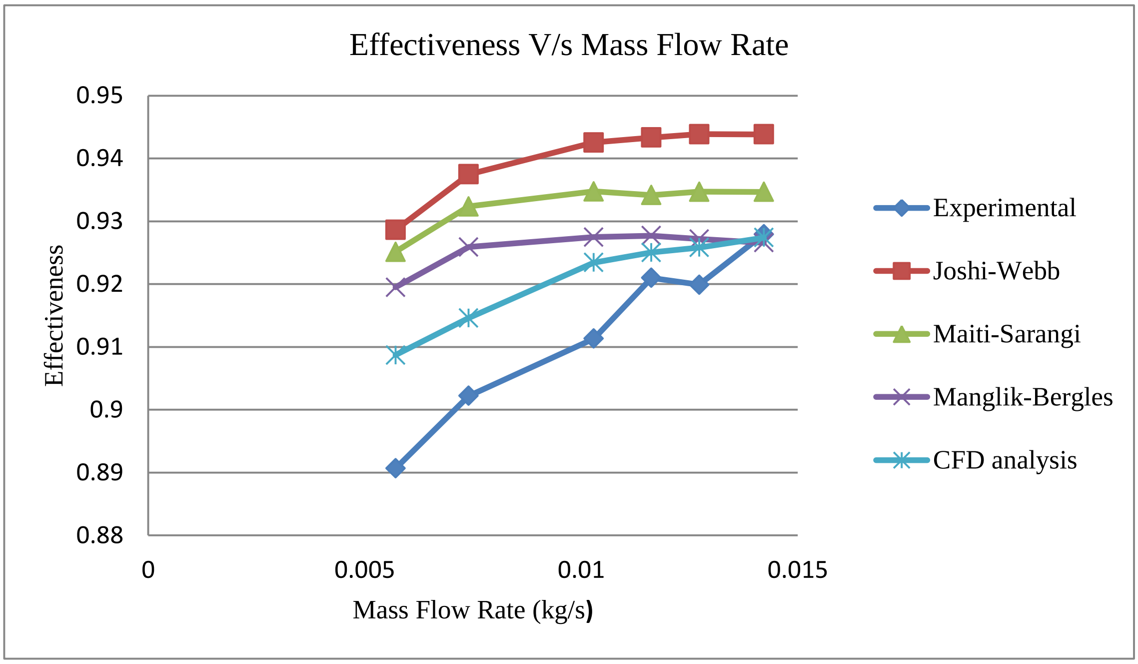


Figure 23. Effectiveness variation with the mass flow rate (at hot inlet temperature of 96 ℃ or 369 K)

From the graphs, it can be observed that with the mass flow rate, the effectiveness increases. It can be also observed that the effectiveness value is more at the hot inlet temperature of 369 K in comparison to the effectiveness value at the hot inlet temperature is 339 K. The effectiveness of the heat exchanger at hot inlet temperature of 359 K lies within 339 K and 369 K.

5.3 Variation of overall thermal conductance with the mass flow rate

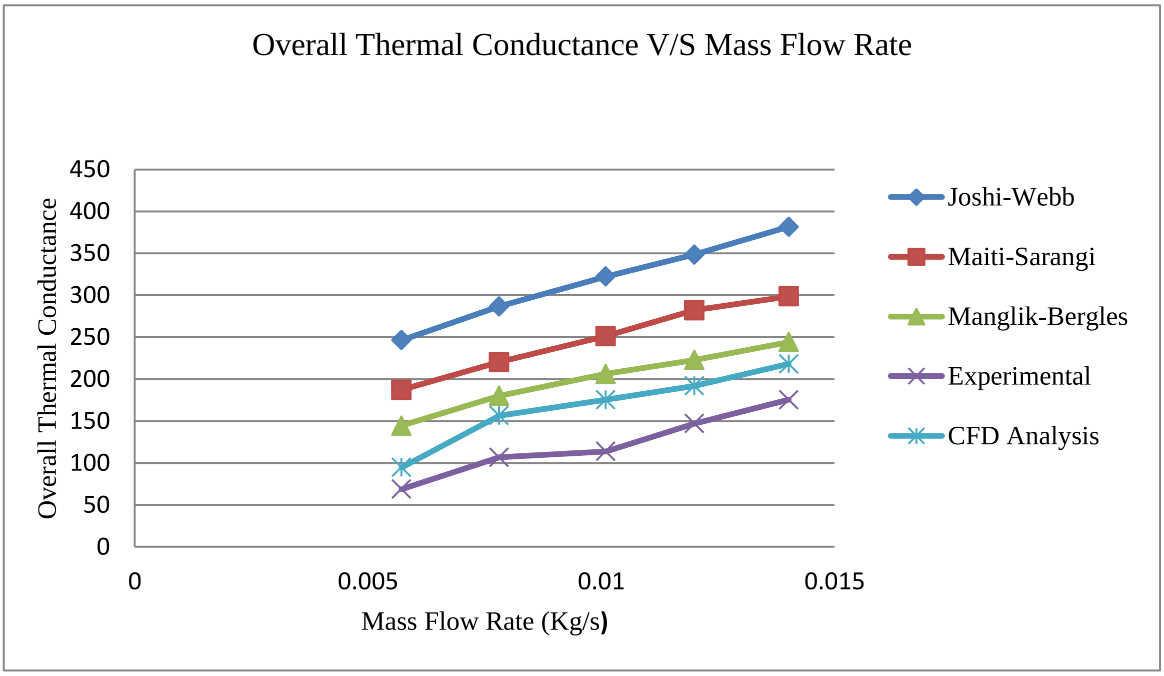


Figure 24. Variation of Overall thermal conductance with the mass flow rate (at hot inlet temperature of 66 ℃ or 339 K)

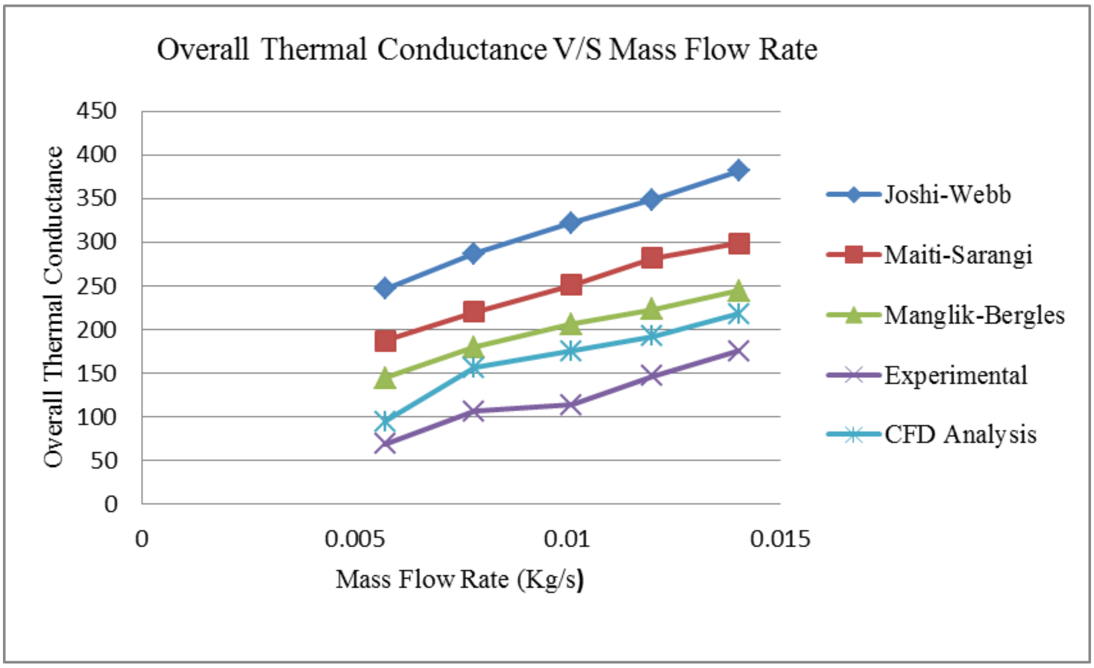


Figure 25.Variation of Overall thermal conductance with the mass flow rate (at hot inlet temperature of 96 ℃ or 369 K).

The overall thermal conductance variation with the mass flow rate for hot inlet temperature of 339 K & 369 K are shown in the figures 24 & 25. It can be observed that the theoretical as well as the experimental overall heat transfer coefficient increases with increase in the mass flow rate. It is because of the fact that the Reynolds number increases with the increase in the mass flow rate. As a result the Colburn factor (*j*) also increases. Since, the Colburn factor is proportional to the heat transfer coefficient, the overall thermal conductance increases

.

5.4 Variation of pressure drop with the mass flow rate

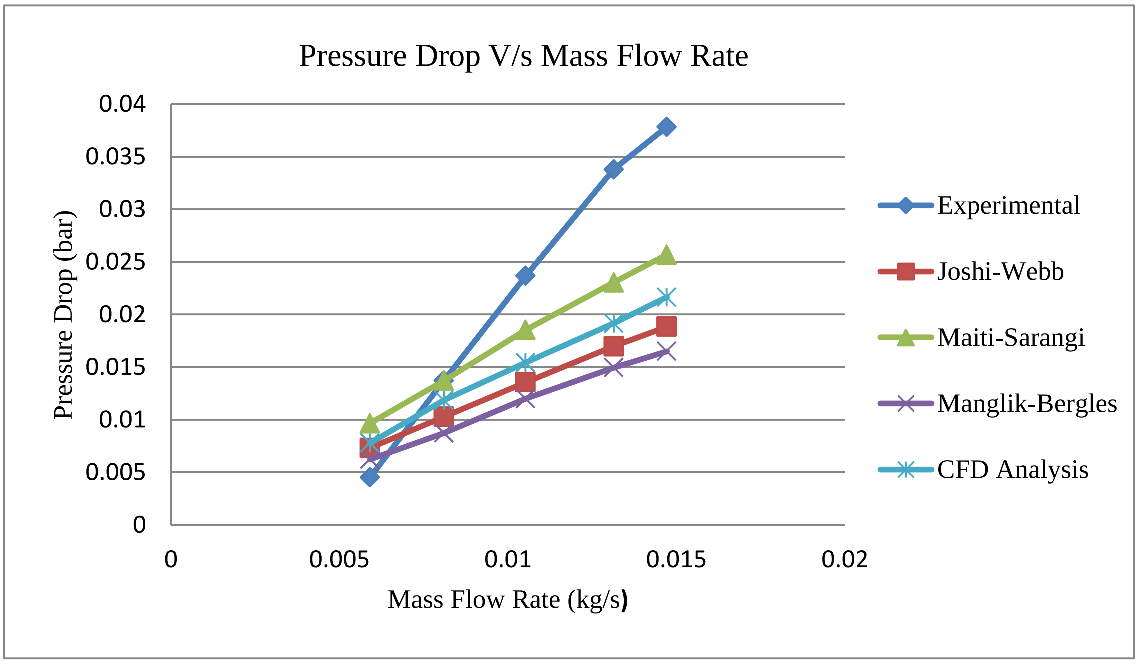


Figure 26. Variation of pressure with the mass flow rate (at hot inlet temperature of 66 °C or 339 K)

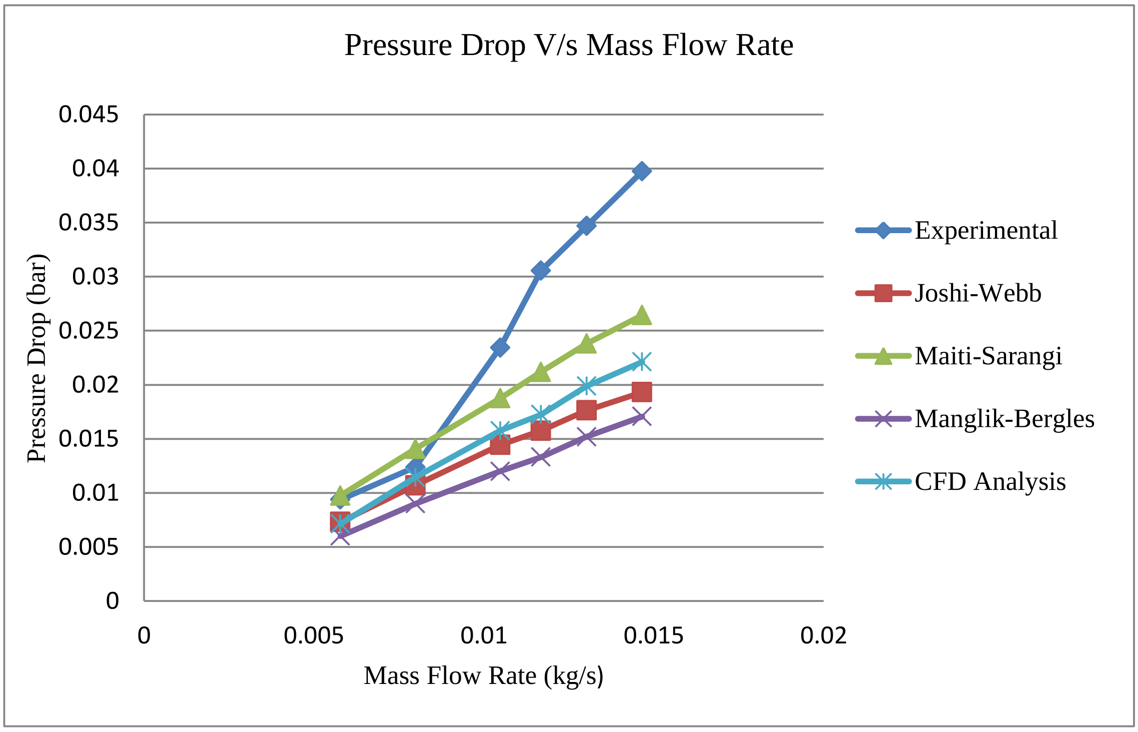


Figure 27. Variation of pressure with the mass flow rate (at hot inlet temperature of 86 °C or 359 K)

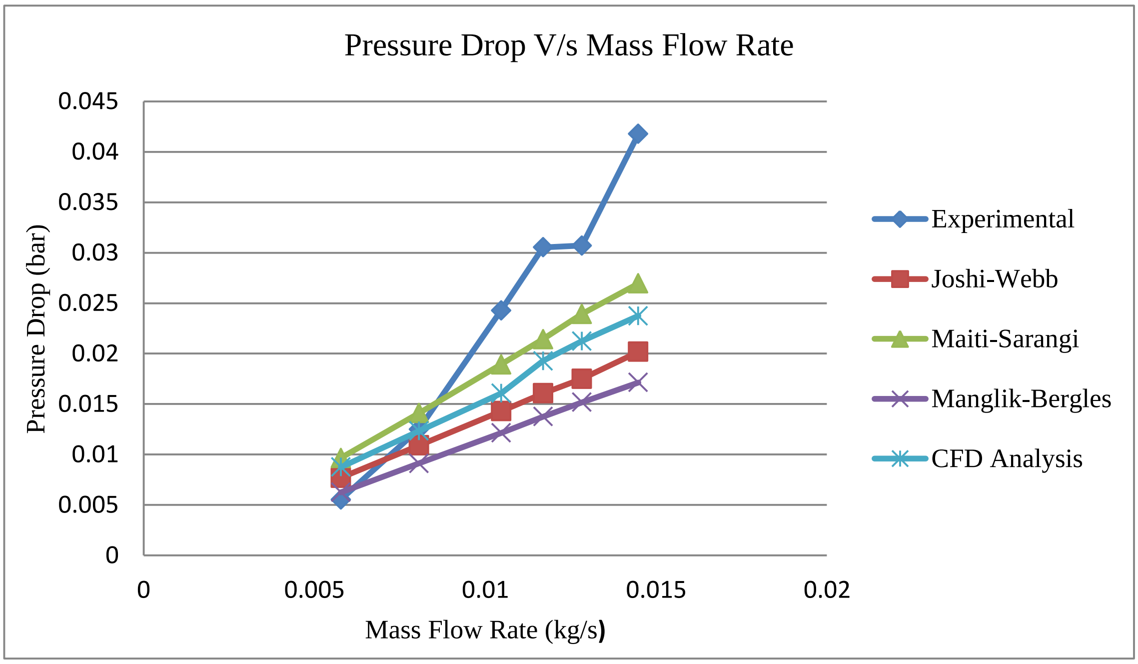


Figure 28. Variation of pressure with the mass flow rate (at hot inlet temperature of 96 ℃ or 369 K)

It can be observed from the plot that the pressure drop in the heat exchanger varies with the varying mass flow rate. The pressure drop obtained experimentally is much more compared to the theoretical pressure drop because in the theoretical pressure drop calculation the header loss, the manufacturing irregularities, pressure drop in the piping etc. are ignored.

CHAPTER 6

CONCLUSIONS

6 CONCLUSIONS

The experimentally obtained results are compared with various correlation results and also with the results obtained from the simulation software of the CFD-fluent. The effectiveness v/s mass flow rate, overall thermal conductance v/s mass flow rate & pressure drop v/s mass flow rate for different hot inlet temperature are evaluated by using the correlations and by using CFD, fluent simulation software. The correlations used for the comparison of the performance parameters with the experimental results are Joshi-Webb correlation, Maiti- Sarangi correlation and Manglik-Bergles correlation. The comparison of the experimental results with the results obtained from the correlations & from the simulation software of Ansys fluent gives the following points:

(i) There is a percentage deviation of the effectiveness of experimentally obtained results from that of the value obtained by the simulation software Ansys fluent.

(ii) There are also deviations between the experimental value & the predicted values of effectiveness calculated by using Maiti-Sarangi, Joshi-Webb and Manglik-Bergles.

(iii) It is observed that the correlation developed by Maiti-Sarangi is better suited to the experimental results compared to the other correlations.

(iv) Up to the Reynolds number 500 the pressure drop of the fluids is below the allowable pressure drop of 0.05 bar. Thereafter, the pressure drop increases rapidly. Between the theoretical pressure drop obtained from various correlations and the pressure drop obtained from the experimental results, there is a large amount of deviation.

6.1 Future Work

A cold fluid test can be carried out using fluids at cryogenic temperatures for checking the validity of the used correlations at cryogenic temperature. Since, in the hot-test the heat loss to the surrounding should n't be neglected, for plate fin heat exchanger having large surface area the heat loss should be considered, when calculating the effectiveness. So, some new correlations can be developed considering the heat loss.

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