DESIGN OF COMPACT PLATE FIN HEAT EXCHANGER

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

Bachelor of Technology

In

Mechanical Engineering

By

JAINENDER DEWATWAL

(ROLL.NUMBER: 10503059)

Department of Mechanical Engineering

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Under the Guidance of
PROF. R.K.Sahoo

Department of Mechanical Engineering

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2009

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Rourkela

CERTIFICATE

This is to certify that the project entitled, "Design of compact plate fin heat exchanger " submitted by Jainender Dewatwal in partial fulfilment of the requirements for the award of Bachelor of Technology, Rourkela (Deemed University) is an authentic work carried out by him under my supervision and guidance.

To the best of my knowledge, the matter embodied in the project has not been submitted to any other University / Institute for the award of any Degree or Diploma.

Date:

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Rourkela

ACKNOWLEDGEMENT

I would like to articulate my deep gratitude to my project guide Prof. R.K.Sahoo who has always been my motivation for carrying out the project.

An assemblage of this nature could never have been attempted without reference to and inspiration from the works of others whose details are mentioned in reference section. I acknowledge my indebtedness to all of them.

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ABSTRACT

Plate fin heat exchangers, because of their compactness, low weight and high effectiveness are widely used in aerospace and cryogenic applications. This device is made of a stack of corrugated fins alternating with nearly equal number of flat separators known as parting sheets, bonded together to form a monolithic block. Appropriate headers are welded to provide the necessary interface with the inlet and the exit streams. While aluminum is the most commonly used material, stainless steel construction is employed in high pressure and high temperature applications.

The performance of a plate fin heat exchanger is determined, among other things, by the geometry of the fins. The most common fin configurations are - (1) plain (straight and uninterrupted) rectangular or trapezoidal fins (2) uninterrupted wavy fins and (3) interrupted fins such as offset strip, louver and perforated fins. The interrupted surfaces provide greater heat transfer at the cost of higher flow impedance.

Here I have designed rectangular offset plate fin heat exchanger. I have assumed some data and based on them I have designed heat exchanger . The flowing fluid in heat exchanger is liquid nitrogen and material of heat exchanger is Al. After designing the heat exchanger, rating is also necessary .

The heat transfer and flow friction characteristics of plate fin surfaces are presented in terms of the Colburn factor j and the Fanning friction factor f vs. Reynolds number Re, the relationships being different for different surfaces.

The laminar flow model under predicts **j** and **f** values at high Reynolds number, while the 2-Layer k-e turbulence model over predicts the data throughout the range of interest. Because most industrial heat exchangers operate with Re less than 3000, and because the j and f data predicted by the laminar and the 2-layer k-e turbulence model differ little from each other at low Reynolds numbers, we have used the laminar flow model up to Reynolds number of 10,000, which is considered to be the limit for plate fin heat exchangers operating with gases. Velocity, pressure and temperature fields have been computed and **j** and **f** factors determined over appropriate range of Reynolds number and geometric dimensions.

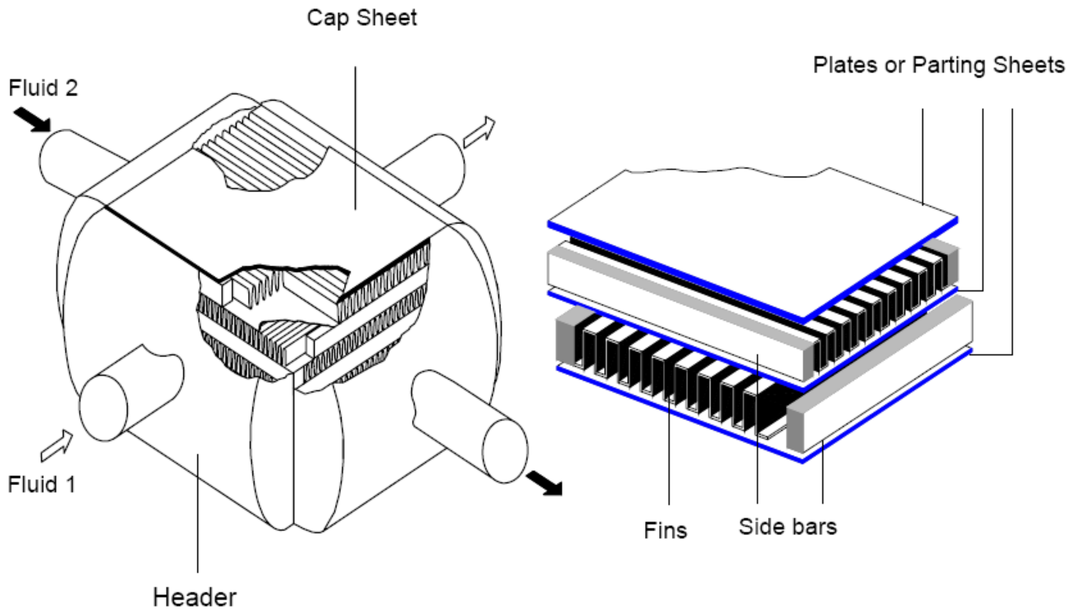
INTRODUCTION

· Plate Fin Heat Exchangers
· Fin Geometries
· Flow Friction and Heat Transfer Characteristics
1

Plate fin heat exchangers are widely used in automobile, aerospace, cryogenic and chemical industries. They are characterized by high effectiveness, compactness (high surface area density), low weight and moderate cost. Although these exchangers have been extensively used around the world for several decades, the technologies related to their design and manufacture remain confined to a few companies in developed countries. Recently efforts are being made in India towards the development of small plate fin heat exchangers for cryogenic and aerospace applications.

Plate Fin Heat Exchangers

A plate fin heat exchanger is a form of compact heat exchanger consisting of a block of alternating layers of corrugated *fins* and flat separators known as *parting sheets*. A schematic view of such an exchanger is given in Fig. 1.1. The corrugations serve both as secondary heat transfer surface and as mechanical support against the internal pressure between layers.



2 **Figure 1.1:** Plate fin heat exchanger assembly and detailsSide bars Plates or Parting Sheets FinsFluid 1 Fluid 2 Cap Sheet Header

Steam exchanges heat by flowing along the passege corrugations between the parting sheets. The edges of the corrugated layers are sealed by side-bars. Corrugations and side-bars are brazed to the parting sheets on both sides to form rigid pressure-containing voids. The first and the last sheets, called *cap sheets*, are usually of thicker material than the parting sheets to support the excess pressure over the ambient and to give protection against physical damage. Each stream enters the block from its own header via ports in the side-bars of appropriate layers and leaves in a similar fashion. The header tanks are welded to the side-bars and parting sheets across the full stack of layers

Merits and Drawbacks

Plate fin heat exchangers offer several advantages over competing designs.

(1) High thermal effectiveness and close temperature approach. (Temperature approach as low as 3K between single phase fluid streams and 1K between boiling and condensing fluids is fairly common.),

(2) Large heat transfer surface area per unit volume (Typically 1000 m^2/m^3),

(3) Low weight,

(4) Multi-stream operation (Up to ten process streams can exchange heat in a single heat exchanger.), and

(5) True counter-flow operation (Unlike the shell and tube heat exchanger, where the shell side flow is usually a mixture of cross and counter flow.).

The principal disadvantages of the plate fin geometry are :

(1) Limited range of temperature and pressure,

(2) Difficulty in cleaning of passages, which limits its application to clean and relatively non-corrosive fluids, and

(3) Difficulty of repair in case of failure or leakage between passages

Materials

Plate fin heat exchangers can be made in a variety of materials. Aluminium is preferred in cryogenic and aerospace applications because of its low density, high thermal conductivity and high strength at low temperature. The maximum design pressure for brazed aluminium plate fin heat exchangers is around 90 bar. At temperatures above ambient, most aluminium alloys lose mechanical strength. Stainless steels, nickel and copper alloys have been used at temperatures up to 500^0 C. The brazing material in case of aluminium exchangers is an aluminium alloy of lower melting point, while that used in stainless steel exchangers is a nickel based alloy with appropriate melting and

.

Manufacture

The basic principles of plate fin heat exchanger manufacture are the same for all sizes and all materials. The corrugations, side-bars, parting sheets and cap sheets are held together in a jig under a predefined load, placed in a furnace and brazed to form the plate fin heat exchanger block. The header tanks and nozzles are then welded to the block, taking care that the brazed joints remain intact during the welding process. Differences arise in the manner in which the brazing process is carried out. The methods in common use are salt bath brazing and vacuum brazing. In the salt bath process, the stacked assembly is preheated in a furnace to about 550^{0} C, and then dipped into a bath of fused salt composed mainly of fluorides or chlorides of alkali metals. The molten salt works as both flux and heating agent, maintaining the furnace at a uniform temperature. In case of heat exchangers made of aluminium, the molten salt removes grease and the tenacious layer of aluminium oxide, which would otherwise weaken the joints. Brazing takes place in the bath when the temperature is raised above the melting point of the brazing alloy. The brazed block is cleansed of the residual solidified salt by dissolving in water, and then thoroughly dried.

In the vacuum brazing process, no flux or separate pre-heating furnace is required. The assembled block is heated to brazing temperature by radiation from electric heaters and by conduction from the exposed surfaces into the interior of the block. The absence of oxygen in the brazing environment is ensured by application of high vacuum (Pressure $\approx 10^{-6}$ mbar). The composition of the residual gas is further improved (lower oxygen content) by alternate evacuation and filling with an inert gas as many times as experience dictates. No washing or drying of the brazed block is required. Many metals, such as aluminium, stainless steel, copper and nickel alloys can be brazed satisfactorily in a vacuum furnace.

Applications

Plate-fin and tube-fin heat exchangers have found application in a wide variety of industries. Among them are air separation (production of oxygen, nitrogen and argon by low temperature distillation of air), petro-chemical and syn-gas production, helium and hydrogen liquefiers, oil and gas processing, automobile radiators and air conditioners, and environment control and secondary power systems of aircrafts. These applications cover a wide variety of heat exchange scenarios, such as:

(1) exchange of heat between gases, liquids or both,

(2) condensation, including partial and reflux condensation,

(3) boiling,

(4) sublimation, and

(5) heat or cold storage

Flow Arrangement

A plate fin heat exchanger accepts two or more streams, which may flow in directions parallel or perpendicular to one another. When the flow directions are parallel, the streams may flow in the same or in opposite sense. Thus we can think of three primary flow arrangements – (i) parallel flow, (ii) counterflow and (iii) cross flow. Thermodynamically, the counterflow arrangement provides the highest heat (or cold) recovery, while the parallel flow geometry gives the lowest. The cross flow arrangement, while giving intermediate thermodynamic performance, offers superior heat transfer properties and easier mechanical layout. Under certain circumstances, a hybrid cross – counterflow geometry provides greater heat (or cold) recovery with superior heat transfer performance. Thus in general engineering practice, plate fin heat exchangers are used in three configurations: (a) cross flow, (b) counterflow and (c) cross-counter flow.

(a) Cross flow (Fig. 1.2(a))

In a cross flow heat exchanger, usually only two streams are handled, thus eliminating the need for distributors. The header tanks are located on all four sides of the heat exchanger core, making this arrangement simple and cheap. If high effectiveness is not necessary, if the two streams have widely differing volume flow rates, or if either one or both streams are nearly isothermal (as in single component condensing or boiling), the cross flow arrangement is preferred. Typical applications include automobile radiators and some aircraft heat exchangers.

(b) Counter flow (Fig. 1.2 (b))

The counterflow heat exchanger provides the most thermally effective arrangement for recovery of heat or cold from process streams. Cryogenic refrigeration and liquefaction equipment use this geometry almost exclusively. The geometry of the headers and the distributor channels is complex and demands proper design.

(c) Cross-Counter flow (Fig.1.2 (c))

The cross-counterflow geometry is a hybrid of counterflow and cross flow arrangements, delivering the thermal effectiveness of counterflow heat exchanger with the

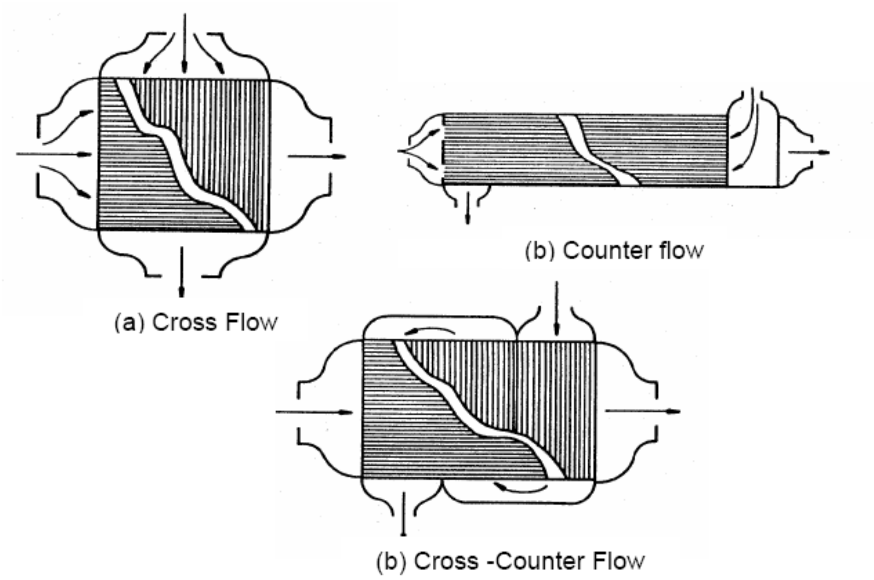


Figure 1.2: Heat exchanger flow arrangements

Figure 1.2: Heat exchanger flow arrangements(a) Cross Flow (b) Counter flow(b) Cross counter flow

superior heat transfer characteristics of the cross flow configuration. In this arrangement, one of the streams flows in a straight path, while the second stream follows a zigzag path normal to that of the first stream. Up to six such passes have been employed. While negotiating the zigzag path, the fluid stream covers the length of the heat exchanger in a direction opposite to that of the direct stream. Thus the flow pattern can be seen to be globally counterflow while remaining locally cross flow. Cross-counter flow PFHEs are used in applications similar to those of simple cross flow exchangers, but allow more flexibility in design. They are particularly suited to applications where the two streams have considerably different volume flow rates, or permit significantly different pressure drops. The fluid with the larger volume flow rate or that with the smaller value of allowable pressure drop flows through the straight channel, while the other stream takes the zigzag path. For example, in a liquid-to-gas heat exchanger, the gas stream with a large volume flow rate and low allowable pressure drop is assigned the straight path, while the liquid stream with a high allowable pressure drop flows normal to it over a zigzag path. This arrangement optimises the overall geometry.

1.2 Fin Geometries

The performance of a plate fin heat exchanger is determined, among other things, by the geometry of the fins. The most common fin configurations are – (1) plain (straight and uninterrupted) fins with rectangular, trapezoidal or triangular passages, (2) uninterrupted wavy fins and (3) interrupted fins such as offset strip, louvered, perforated and pin fins. The details of each fin type are given below.

Plain Fins

These are straight fins that are continuous in the fluid flow direction (Fig.1.3(a, b). Although passages of triangular and rectangular cross section are more common, any desired shape can be given to the fins, considering only manufacturing constraints. Straight fins in triangular arrangement can be manufactured at high speeds and hence are less expensive than rectangular fins. But generally they are structurally weaker than rectangular fins for the same passage size and fin thickness. They also have lower heat transfer performance compared to rectangular fins, particularly in laminar flow

Plain fins are used in those applications where core pressure drop is critical. An exchanger with plain fins requires a smaller flow frontal area than that with interrupted fins for specified pressure drop, heat transfer and mass flow rate. Of course, the required passage length is higher leading to a larger overall volume.

Wavy Fins

Wavy fins are uninterrupted fin surfaces with cross-sectional shapes similar to those of plain fins, but with cyclic lateral shifts perpendicular to the flow direction (Fig.1.3 (c)). The resulting wave form provides effective interruptions and induces a complex flow field. Heat transfer is enhanced due to creation of Goertler vortices. These counter-rotating vortices form while the fluid passes over the concave wave surfaces, and produce a corkscrew-like flow pattern.

The heat transfer and pressure drop characteristics of a wavy fin surface lie between those of plain and offset strip fins. The friction factor continues to fall with increasing Reynolds number. Wavy fins are common in the hydrocarbon industry where exchangers are designed with high mass velocities and moderate thermal duties. Unlike offset strip fins, the thickness of wavy fins is not limited at high fin densities. Therefore, wavy fins are often used for streams at high pressure, particularly those which can tolerate somewhat poor heat transfer coefficient.

Offset Strip Fins

This is the most widely used fin geometry in high performance plate fin heat exchangers. It consists of a type of interrupted surface, which may be visualised as a set of plain fins cut normal to the flow direction at regular intervals, each segment being offset laterally by half the fin spacing (Fig. 1.3 (d)). Surface interruption enhances heat transfer by two independent mechanisms. First, it prevents the continuous growth of thermal boundary layer by periodically interrupting it. The thinner boundary layer offers lower thermal resistance compared to continuous fin types. Above a critical Reynolds number, interrupted surfaces offer an additional mechanism of heat transfer enhancement. Oscillations in the flow field in the form of vortices shed from the trailing edges of the interrupted fins enhance local heat transfer by continuously bringing in fresh fluid

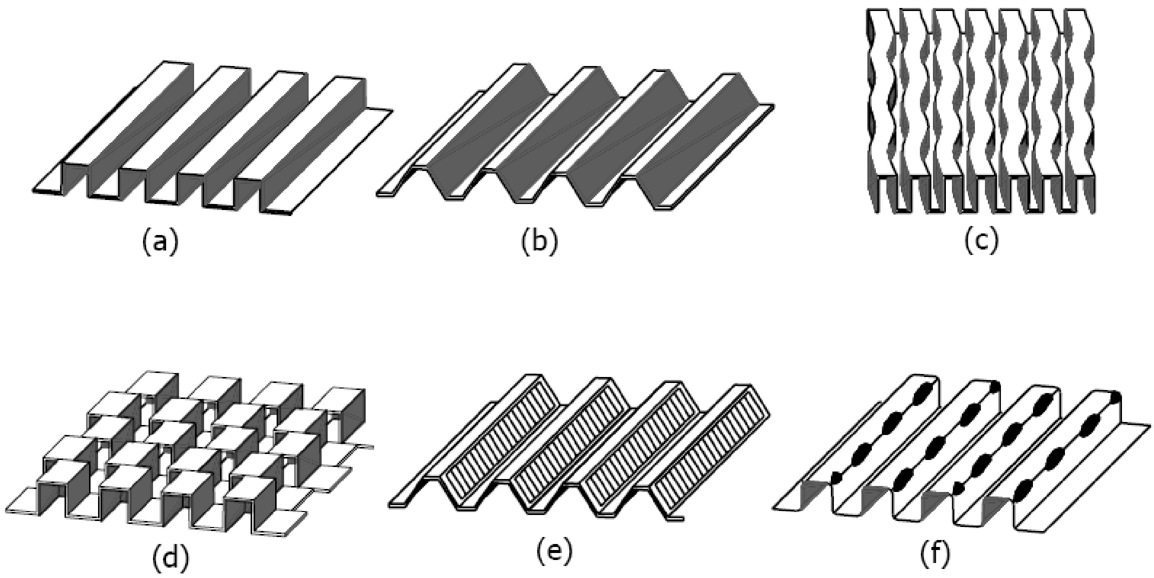


Figure 1.3: Types of plate fin surfaces: (a) plain rectangular (b) plain trapezoidal (c) wavy (d) serrated or offset strip fin (e) louvered (f) perforated

towards the heat transfer surfaces. This enhancement is accompanied by an increase in pressure drop.

The heat transfer performance of offset strip fin is often as much as 5 times that of a plain fin surface of comparable geometry, but at the expense of higher pressure drop. For specified heat transfer and pressure drop requirements, the offset strip fin surface demands a somewhat higher frontal area compared to those with plain fin, but results in a shorter flow length and lower overall volume. An undesirable characteristic of this type of fin is that at high Reynolds numbers the friction factor remains nearly constant (because of the higher contribution of form drag), while the heat transfer performance goes down. Therefore, offset strip fins are used less frequently in very high Reynolds number applications. On the other hand, they are extensively used in air separation and other cryogenic applications where mass velocities are low and high thermal effectiveness is essential.

The louvered fin geometry shown in Fig. 1.3 (e) bears a similarity to the offset strip fin. Instead of shifting the slit strips laterally, small segments of the fin are slit and rotated 20 to 45 degrees relative to the flow direction. The base surface of the louvered fin geometry can be of triangular or rectangular shape, and louvers can be cut in many different forms.

The multilouvered fin has the highest heat transfer enhancement relative to pressure drop in comparison with most other fin types. Flow over louvered fin surfaces is similar in nature to that through the offset strip fin geometry, with boundary layer interruption and vortex shedding playing major roles. An important aspect of louvered fin performance is the degree to which the flow follows the louver. At low Reynolds number the flow is nearly parallel to the axial direction (duct flow), whereas at high Reynolds number the flow is in the direction of the louvers (boundary layer flow). Louvered fins are extensively used in automotive heat exchangers.

Perforated fins shown in Fig.1.3 (f) are made by punching a pattern of spaced holes in the fin material before it is folded to form the flow channels. The channels may be triangular or rectangular in shape with either round or rectangular perforations. While this geometry, with boundary layer interruptions, is a definite improvement over plain fins, its performance is generally poorer than that of a good offset strip fin. Furthermore, the perforated fin represents a wasteful way of making an enhanced surface, since the material removed in creating the perforations is thrown out as scrap. Perforated fins are now used only in limited number of applications such as turbulators in oil coolers.

In a pin fin exchanger, a large number of small pins are sandwiched between plates in either an inline or staggered arrangement. Pins may have a round, an elliptical, or a rectangular cross section. These types of finned surfaces are not widely used due to low compactness and high cost per unit surface area compared to multilouvered or offset strip fins. Due to vortex shedding behind the pins, noise and flow-induced vibration are produced, which are generally not acceptable in most heat exchanger applications. The potential application of pin fin surfaces is at low flow velocities (Re < 500), where pressure drop is negligible. Pin fins are used as electronic cooling devices with free-convection flow on the pin fin side.

Heat Transfer and Flow Friction

Characteristics

The heat transfer and flow friction characteristics of a heat exchanger surface are commonly expressed in non-dimensional form and are simply referred to as the basic characteristics or basic data of the surface. These characteristics are presented in terms of the Colburn factor **j** and Friction factor **f** vs. Reynolds number Re, the relationships being different for different surfaces. The Colburn and Friction factors are defined by the relations:

$$J = h(Pr)^{(2/3)}/GCp$$

$$P = 4fLG^2/(2D_h\rho)$$

where, h = heat transfer coefficient ($W/m^2 K$)

G = mass velocity ($kg/m^2 s$) [on the basis of minimum free flow area]

L = length of flow passage (m)

D_h = hydraulic diameter (m), and

ρ = mean density of fluid (kg/m^3).

The friction factor **f** takes both viscous shear (skin friction) and pressure forces (form drag) into consideration. This approach is somewhat arbitrary since geometric variables, other than the hydraulic diameter, may have a significant effect on surface performance. It also becomes necessary to present j and f data separately for each surface type. The **j** and **f** data so presented are applicable to surfaces of any hydraulic diameter, provided a complete geometric similarity is maintained.

One of the earliest and the most authoritative sources of experimental **j** and **f** data on plate fin surfaces is the monograph *Compact Heat Exchangers* by Kays and London [1]. Although nearly two decades have passed after the latest edition, there has not been any significant addition to this database in open literature. Attempts have been made towards numerical prediction of heat transfer coefficient and friction factor; but they have generally been unable to match experimental data. Several empirical correlations, however, have been generated from the data of Kays and London, which have found extensive application in industry, particularly in less-critical designs. For critical applications, direct experimental determination of **j** and **f** factors for each fin geometry remains the only choice.

In a plate fin heat exchanger, the hydraulic diameter of the flow passage is generally small due to closely spaced fins. Operation with low density gases leads to excessive pressure drop unless the gas velocity in the flow passage is kept low. These factors imply operational Reynolds number less than 10,000, the common range being between 500 and 3000 for most ground based applications

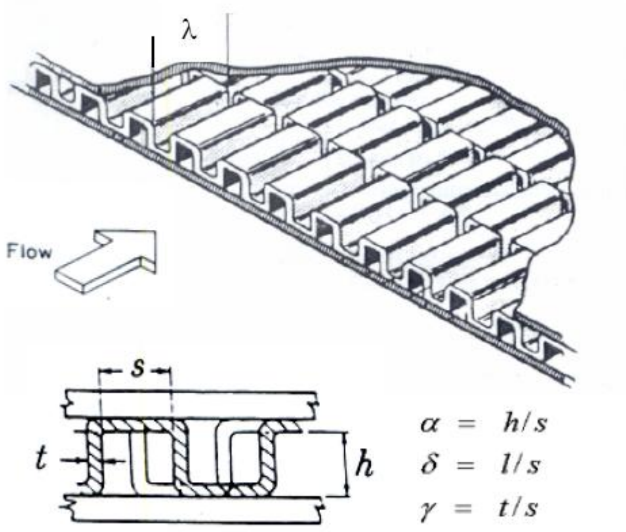
RECTANGULAR OFFSET STRIP FIN SURFACES

The offset strip fin is one of the most widely used finned surfaces, particularly in high effectiveness heat exchangers employed in cryogenic and aircraft applications. These fins are created by cutting a set of plain rectangular fins periodically along the flow direction, and shifting each strip thus generated by half the fin spacing alternately left and rightward. The flow is thus periodically interrupted, leading to creation of fresh boundary layers and consequent heat transfer enhancement. Interruption of flow also leads to greater viscous pressure drop, manifested by a higher value of effective friction factor. In addition to the effect of wall shear, resistance to flow also increases due to *form drag* over the leading edges of the fin sections facing the flow, and due to trailing edge vortices. The effective heat transfer coefficient and friction factor are composite effects of the above mechanisms.

2.1 The Offset Strip Fin Geometry

The geometry of the offset strip fin surface is described by the following parameters:
(i) fin spacing (s), excluding the fin thickness,
(ii) fin height (h), excluding the fin thickness,
(iii) fin thickness (t), and
(iv) the strip length (_), in the flow direction.
The lateral fin offset is generally uniform and equal to half the fin spacing (including fin thickness). Figure 3.1 shows a schematic view of the rectangular offset strip fin surface and defines the geometric parameters. The following are some commonly used secondary parameters derived from the basic fin dimensions.

Figure



Fin Parameters

s - fin spacing
h - fin height
t - fin thickness
λ - strip length
α = h/s
δ = λ/s
γ = t/s

Figure 3.1: Geometry of a typical offset strip fin surface

Design of compact plate fin heat exchanger using Magahanic correlation

(1) Heat transfer data specification

a. fin thickness (t)=.2 mm
b. fin frequency(f) =714.25 fin per meter
c. fin length (l)=1.5 mm
d. fin height (h)=9.3 mm
e. fin spacing (s)=(1/f)-t
   = (1/714.25)-.2
   = 1.2 mm

f. plate thickness(b)= h + t
   = 9.3+.2
   =9.5 mm
g. free flow area (A_{ff}) =(s-t)h
   =(1.2-.2)*9.3
   =9.6*10-^{6} m^2
h. frontal area (A) =(h+t)(s+t)
   =(9.3+.2)(1.2+.2)
   =.0000133 m^2
i. heat transfer area (A_s)=2*h*l+2*s*l+2*h*l
   =2*9.3*1.5+2*1.2*1.5+9.3*.2*2
   =35.22 mm^2

j. Fin area (A_f) =2*h*l+2*h*l
   =2*9.3*1.5+9.3*2*.2
   =31.62 mm^2

k. eq. Dia. =D_h =((2lh(s-t))/(ls+hl+ht)
   =(2*1.5*9.3(1.2-.2))/(1.5*1.2+9.3*1.5+9.3*.2)
   =1.58 mm

l. fin area/total surface area=(A_f /A_s)=31.62/35.22=.8977

m. frontal area ratio (σ)=A_f /A_s=9.3/13.3=.69924

n. α =h/s=9.3/1.2=7.75
o. δ =l/s=1.5/1.2=1.25
p. ν = t/s=.2/1.2=.166

(2.) DATA INPUT

material of the fin =Al

conductivity of the fin material(K_f)=150 W/mK

end plate of thickness=6 mm

end bars thickness=6 mm

	hot fluid	cold fluid
inlet temp	310 k	99.716 K
Outlet temp	124.26 K	301.54 K
Mass flow rate	.0822 Kg/s	.07791 Kg/s
Pressure inlet	8 bar	1.15 bar
Allowable pressure drop	.05 bar	.05 bar
Density at avg. temp	1.583	1.711

(3) ASSUMPTION

avg. wall temp	200 K	
width(w)	.115 mm	.115 mm
no of layers	5	4
area between plate	.0054625 m^2	.00437m^2

A=(wbn)

For hot fluid =.115*9.5*.5

For cold fluid =.115*9.5*4

Free flow area	.003819643 m^2	.003055714 m^2

($A_{ff}=A*\sigma$)

For hot fluid =.0054625*.699

For cold fluid=.00437*.699

(4) CONVECTIVE HEAT TRNSFER CO-EFFICIENT

(a) bulk temp =(inlet temp+outlet temp)/2

=(310+124.26)/2

=217.13 (hot fluid)

= (301.54+124.26)/2

=200.628 (cold fluid)

(b) mean film temp.=(wall temp+bulktemp)/2

= (200+217.13)/2

=208.56 k (hot fluid)

=(200+200.628)/2

=200.314 (cold fluid)

Properties

	Hot fluid	cold fluid
Sp. Heat (c_p)	1043 J/Kg-K	1043 J/Kg-K
Viscosity(μ)	.0000134 N/m^2-s	.00001295 N/m^2- s
Predelt number	.74767	.75

(c) core mass velocity(G)=m_{ff} / A_{ff}

G=(.0822/.003819643)

=21.5203 Kg/sm^2 (hot fluid)

=(.07791/.003055714)

=25.4964 Kg/sm^2 (cold fluid)

(d) Reynolds no. (R_e) =GD/μ

=(21.5203*1.58)/.0000134

=2544.421806 (hot fluid)

=(25.4964*1.58)/.0000134

=3119.288461 (cold fluid)

$Re_f^*=648.25(h/s)^{-.06}(l/s)^{.1}(t/s)^{-.196}$

=832.874

$Re_j^*=1568.58(h/s)^{-.217}(l/s)^{-1.433}(t/s)^{-.217}$

=1077.7424

Since Re> Re*

$F=.32(Re)(h/s)^{.221}(l/s)^{-0.185}(t/s)^{-0.023}$

=.053 (hot fluid)

F= .02039 (cold fluid)

$J=.18(Re)^{-0.42}(h/s)^{.288}(l/s)^{-0.184}(t/s)^{-0.05}$

=.012 (hot fluid)

J=.01164 (cold fluid)

Pressure drop /length=$(.5fG^2)D_{eq}$

$=(.053*21.5203^2*1000)/(2*1.583*1.58)$

=4931.727799 N/m^3

$h=jc_pl/p_r^{(2/3)}$

=344.704 (hot fluid)

=374.129 (cold fluid)

fin parameter

$M=\sqrt{2h/k_ft}=\sqrt{(2*344.704*1000)/(150*.2)}$

=151.5925 (hot fluid)

$=\sqrt{2*374.129*1000/(150*.2)}$

= 157.930 (cold fluid)

Ml_f =Mb/2

=151.5925*9.5/2=.7200644

=157.930*9.5/2=.75016

n_f =tanh(ml)/ml

=tanh(.7200644)/.7200644=.856797 (hot fluid)

=tanh(.75016)/.75016=.84680918 (cold fluid)

Overall efficiency=№ =1-(A_f/A_s)(1- n_f)

=1-.8977(1-.856797)

=.871434 (hot fluid)

=1-.8977(1-.84680)

=.862467 (cold fluid)

(5)overall heat transfer coefficient(w/m^2-k)

(a) total area/separating wall area(A_0/A_w)=(1-ft)/(1-A_f/A_w)

=(1-.71425*.2)/(1-.8977)

=8.3857 m^2/m^3

(b) overall thermal resistance

$$(1/U_0)=(n_c w_c/n_h w_h(№_h h_h))+(aA_o/K_w A_w)+(1/№_c h_c)$$

$$=.005807\ m^2K/W$$

$$U_o = 172.204\ W/m^2K$$

(6) heat transfer area m/m^2

UA for heat exchanger = 1088 W/K

Required heat transfer area = 6.318074 m^2

Required heat transfer area /length

A/L=7.714857 m^2/m =$4A_{min(ff)}/D_{eq}$

Required length of heat exchanger

L=.81894 m

(7) pressure drop

p /L=($fG^2/2\rho D_{eq}$)

=4038.829 Pa (hot fluid)

=4948.206647 Pa (cold fluid)

(8) final dimensions

Core length =819 mm (without longitudinal heat conduction)

Core width =115 mm

Total width =115+2*6=127 mm

no of HP side =5

no of LP side =4

core height =(n_c+n_h)*b+(n_c+n_h)*a

=92.7 mm

Total height =92.7+2*6=104.7 mm

EFEECT OF LONGITUDNAL HEAT CONDUCTION

Heat conduction area = A_w=core width*total height-free flow area of hot side

- free flow area of cold side

=115*104.7-3819-3055

=5165 mm^2

C_{min} = .07791*1043=81.26

FOS = 2.47

UA = UA_o*FOS

= 1088*2.47=2687.36

NTU = UA/C_{min}

= (2687.36)/81.26=33.071

$\lambda = (K_W A_W)/LC_{min}$

=(150*.005165)/(2.022 *81.26)=.011641

$Y = \lambda*NTU*C_R$

=.0116*33.071*.947

=.3649

$\gamma=(1-C_R)/(1-C_R)(1+Y)$

=(1-.9478)/(1+.9478)(1+.3649)

=.0196

$\Phi=\gamma(Y/(1+Y))^{.5}((1+\gamma)Y/(1-\gamma(1+\gamma)Y))$

=.0038

$\Psi=(1+\Phi)/(1-\Phi)$

=1.0076

$r1=(1-C_R)*NTU/(1+\lambda NTUC_R)$

=1.264

$1-\varepsilon=(1-C_R)/(\Psi e^{r1}-C_R)$

=.0199

ε =.9800

Heat exchanger area=15.605m^2

Required length = 2.022 m

RATING OF HEAT EXCHANGER

free flow area (A_{ff}) $=9.6*10^{-6}$ m^2

heat transfer area (A_s)= 35.22 mm^2

l=1.5 mm

D_{eq} =1.58 mm

frontal area ratio (σ)=A_f /A_s=9.3/13.3=.69924

mean film temp. =(wall temp+bulktemp)/2

= (200+217.13)/2

=208.56 k (hot fluid)

=(200+200.628)/2

=200.314 (cold fluid)

UA for heat exchanger = 1088 W/K

C_{hot}=85.73

C_{cold}=81.26

$C^*=C_{min}/C_{max}$=81.26/85.73=.94

$NTU=UA/C_{min}$=13.38

$\varepsilon=(1-e^{-NTU(1-C^*)})/(1-c^* e^{-NTU(1-C^*)})$

=.95

$T_{h,o} = T_{h,I} - \varepsilon\, C_{min}/C_{max}\,(T_{h,I} - T_{c,I})$

=310-.95(81.26/85.73(310-99.7156)

=121 K

$T_{c,o} = T_{c,I} + \varepsilon\, C_{min}/C_{max}\,(T_{h,I} - T_{c,I})$

=99.716+.94(310-99.716)

=300 K

$Q = \varepsilon\, C_{min}(T_{h,I} - T_{c,I})$

=.95*81.26(310-91.716)

=16233.29 J

pressure drop

p /L=$(fG^2/2\rho D_{eq})$

=4038.829 Pa (hot fluid)

=4948.206647 Pa (cold fluid)

Design of compact plate fin heat exchanger using Wieting correlations

(2) Heat transfer data specification

a. fin thickness (t)=.2 mm
b. fin frequency(f) =714.25 fin per meter
c. fin length (l)=1.5 mm
d. fin height (h)=9.3 mm
e. fin spacing (s) =(1/f)-t
   = (1/714.25)-.2
   = 1.2 mm plate thickness

f. plate thickness (b) = h + t
   = 9.3+.2
   =9.5 mm
g. free flow area (A_{ff}) =(s-t)h
   =(1.2-.2)*9.3
   =$9.6*10^{-6}$ m^2
h. frontal area (A) =(h+t)(s+t)
   =(9.3+.2)(1.2+.2)
   =.0000133 m^2
i. heat transfer area (A_s)=2*h*l+2*s*l+2*h*l
   =2*9.3*1.5+2*1.2*1.5+9.3*.2*2
   =35.22 mm^2

j. Fin area (A_f) =2*h*l+2*h*l
   =2*9.3*1.5+9.3*2*.2
   =31.62 mm^2

k. eq. Dia. =D_h =(2sh)(s+h)
   =(2*1.2*9.3)(1.2+9.3)
   =2.125 mm

l. fin area/total surface area=(A_f /A_s)=31.62/35.22=.8977

m. frontal area ratio (σ)=A_f /A_s=9.3/13.3=.69924

n. α =s/h=1.2/9.3=.129
o. δ =t/l=.2/1.5=.133

p. ν = t/s=.2/1.2=.166

(2.) DATA INPUT

material of the fin =Al

conductivity of the fin material(K_f)=150 W/mK

end plate of thickness=6 mm

end bars thickness=6 mm

	hot fluid	cold fluid
inlet temp	310 k	99.716 K
Outlet temp	124.26 K	301.54 K
Mass flow rate	.0822 Kg/s	.07791 Kg/s
Pressure inlet	8 bar	1.15 bar
Allowable pressure drop	.05 bar	.05 bar
Density at avg. temp	1.583	1.711

(3) ASSUMPTION

avg. wall temp	200 K	
width(w)	.115 mm	.115 mm
no of layers	7	6
area between plate	.007647 m^2	.006555m^2

A=(wbn)

For hot fluid =.115*9.5*7

For cold fluid =.115*9.5*6

Free flow area	.005347 m^2	.004583 m^2

(A_{ff}=A*σ)

For hot fluid =.007647*.699

For cold fluid=.006555*.699

(4) CONVECTIVE HEAT TRNSFER CO-EFFICIENT

(a) bulk temp =(inlet temp+outlet temp)/2

=(310+124.26)/2

=217.13 (hot fluid)

= (301.54+124.26)/2

=200.628 (cold fluid)

(b) mean film temp.=(wall temp+bulktemp)/2

= (200+217.13)/2

=208.56 k (hot fluid)

=(200+200.628)/2

=200.314 (cold fluid)

Properties

	Hot fluid	cold fluid
Sp. Heat (c_p)	1043 J/Kg-K	1043 J/Kg-K
Viscosity(μ)	.0000134 N/m^2-s	.00001295 N/m^2- s
Predelt number	.74767	.75

(c) core mass velocity$(G)=m_{ff}/A_{ff}$

$G=(.0822/.005347)$

$=15.37166 Kg/sm^2$ (hot fluid)

$=(.07791/.004583)$

$=16.9976 Kg/sm^2$ (cold fluid)

(d) Reynolds no. $(R_e) = GD/\mu$

$=(15.37166*2.125)/.0000134$

$=2438.49$ (hot fluid)

$=(16.9976*2.125)/.0000134$

$=2790.12$ (cold fluid)

Re<1000

$J=.483Re^{-0.536}(l/D_h)^{-0.162}(\alpha)^{-0.184}$

$F=7.661Re^{-0.712}(l/D_h)^{-.384}(\alpha)^{-0.092}$

Re>2000

$J=.242Re^{-0.368}(l/D_h)^{-.322}(t/D_h)^{0.089}$

$F=1.136Re^{-0.198}(l/D_h)^{-0.781}(t/D_h)^{.534}$

Since Re>2000

$F=1.136(2438.49)^{-0.1998}(1.5/2.125)^{-0.781}(.2/2.125)^{.534}$

$=.09012$ (hot fluid)

$F= .08775$ (cold fluid)

$J=.242Re^{-0.368}(l/D_h)^{-.322}(t/D_h)^{0.089}$

$=.242(2438.49)^{-0.368}(1.5/2.125)^{-.322}(.2/2.125)^{0.089}$

$=.01244$ (hot fluid)

$J=.011184$ (cold fluid)

$h=jc_pl/p_r^{(2/3)}$

$=242.076$ (hot fluid)

$=254.2087$ (cold fluid)

fin parameter

$M=\sqrt{2h/k_ft}=\sqrt{(2*242.076*1000)/(150*.2)}$

$=127.0371$ (hot fluid)

$=\sqrt{(2*254.20*1000)/(150*.2)}$

$= 130.181$ (cold fluid)

$Ml_f = Mb/2$

$=127.0371*9.5/2=.6034$ (hot fluid)

$=130.181*9.5/2=.6183$ (cold fluid)

$n_f = \tanh(ml)/ml$

$=\tanh(.6034)/.6034=.8940$ (hot fluid)

$=\tanh(.6183)/.6183=.8894$ (cold fluid)

Overall efficiency=№ $=1-(A_f/A_s)(1- n_f)$

$=1-.8977(1-.8940)$

$=.9048$ (hot fluid)
$=1-.8977(1-.8894)$
$=.9007$ (cold fluid)

(5)overall heat transfer coefficient(w/m^2-k)

(a) total area/separating wall area$(A_0/A_w)=(1-ft)/(1-A_f/A_w)$
$=(1-.71425*.2)/(1-.8977)$
$=8.3857\ m^2/m^3$

(b) overall thermal resistance

$$(1/U_0)=(n_c w_c/n_h w_h(№_h h_h))+(aA_o/K_w A_w)+(1/№_c h_c)$$
$$=.008325\ m^2K/W$$
$$U_o=120.118\ W/m^2K$$

(6) heat transfer area m/m^2

UA for heat exchanger = 1088 W/K
Required heat transfer area = 6.057 m^2
Required heat transfer area /length
$A/L=(4*.0045)*(.002125)\ m^2/m=8.625=4A_{min(ff)}/D_{eq}$
Required length of heat exchanger
$L=1.051$ m

(7) pressure drop

$p/L=(fG^2/2\rho D_{eq})$
$=3323.829$ Pa (hot fluid)
$=3666.487$ Pa (cold fluid)

(8) final dimensions

Core length =1051 mm (without longitudinal heat conduction)
Core width =115 mm
Total width =115+2*6=127 mm
no of HP side =7
no of LP side =6
core height $=(n_c+n_h)*b+(n_c+n_h)*a$
$=133.58$mm
Total height =133.58+2*6=145.58 mm

EFEECT OF LONGITUDNAL HEAT CONDUCTION

Heat conduction area = A_w=core width*total height-free flow area of hot side

- free flow area of cold side

=115*104.7-3819-3055

=6810 mm^2

C_{min} = .07791*1043=81.26

FOS = 2.5

$UA = UA_o*FOS$

= 1088*2.5=2720W/K

$NTU = UA/C_{min}$

= (2720)/81.26=33.4727

$\lambda = (K_W A_W)/LC_{min}$

=(150*.005165)/(2.72 *81.26)=.0119

$Y = \lambda*NTU*C_R$

=.0119*33.4727*.94

=.3794

$\gamma=(1-C_R)/(1-C_R)(1+Y)$

=(1-.9478)/(1+.9478)(1+.3794)

=.0194

$\Phi=\gamma(Y/(1+Y))^{.5}((1+\gamma)Y/(1-\gamma(1+\gamma)Y))$

=.0039

$\Psi=(1+\Phi)/(1-\Phi)$

=1.0079

$r1=(1-C_R)*NTU/(1+\lambda NTUC_R)$

=1.2666

$1-\varepsilon=(1-C_R)/(\Psi e^{r1}-C_R)$

=.0199

ε =.98000

Heat exchanger area=22.64

Required length=2.62 m

Design of compact plate fin heat exchanger using Joshi & Webb correlation

(3) Heat transfer data specification

a. fin thickness (t)=.2 mm
b. fin frequency(f) =714.25 fin per meter
c. fin length (l)=1.5 mm
d. fin height (h)=9.3 mm
e. fin spacing (s)=(1/f)-t
= (1/714.25)-.2
= 1.2 mm plate thickness

f. plate thickness (b) = h + t
= 9.3+.2
=9.5 mm
g. free flow area (A_{ff}) =(s-t)h
=(1.2-.2)*9.3
=9.6*10-^{6} m^2
h. frontal area (A) =(h+t)(s+t)
=(9.3+.2)(1.2+.2)
=.0000133 m^2
i. heat transfer area (A_s)=2*h*l+2*s*l+2*h*l
=2*9.3*1.5+2*1.2*1.5+9.3*.2*2
=35.22 mm^2

j. Fin area (A_f) =2*h*l+2*h*l
=2*9.3*1.5+9.3*2*.2
=31.62 mm^2

k. eq. Dia. =D_h =2*(s-h)/((s+h)+(th/l))
=2*(1.2-.2)/((1.2+9.3)+(.2*9.3/1.5))
=1.58432 mm

l. fin area/total surface area=(A_f /A_s)=31.62/35.22=.8977

m. frontal area ratio (σ)=A_f /A_s=9.3/13.3=.69924

n. α =h/s=9.3/1.2=7.75

o. =l/s=1.5/1.2=1.25

p. ν = t/s=.2/1.2=.166

(2.) DATA INPUT

material of the fin =Al

conductivity of the fin material(K_f)=150 W/mK

end plate of thickness=6 mm

end bars thickness=6 mm

	hot fluid	cold fluid
inlet temp	310 k	99.716 K
Outlet temp	124.26 K	301.54 K
Mass flow rate	.0822 Kg/s	.07791 Kg/s
Pressure inlet	8 bar	1.15 bar
Allowable pressure drop	.05 bar	.05 bar
Density at avg. temp	1.583	1.711

(3) ASSUMPTION

avg. wall temp	200 K	
width(w)	.115 mm	.115 mm
no of layers	6	5
area between plate	.00655 m^2	.00546m^2

A=(wbn)

For hot fluid =.115*9.5*.5

For cold fluid =.115*9.5*4

Free flow area .00458m^2 .003819 m^2

(A_{ff}=A*σ)

For hot fluid =.00655*.699

For cold fluid=.00546*.699

(4) CONVECTIVE HEAT TRNSFER CO-EFFICIENT

(a) bulk temp =(inlet temp+outlet temp)/2

=(310+124.26)/2

=217.13 (hot fluid)

= (301.54+124.26)/2

=200.628 (cold fluid)

(b) mean film temp.=(wall temp+bulktemp)/2

= (200+217.13)/2

=208.56 k (hot fluid)

=(200+200.628)/2

=200.314 (cold fluid)

Properties

	Hot fluid	cold fluid
Sp. Heat (c_p)	1043 J/Kg-K	1043 J/Kg-K
Viscosity(μ)	.0000134 N/m^2-s	.00001295 N/m^2- s
Predelt number	.74767	.75

(c) core mass velocity(G)=m_{ff}/A_{ff}

G=(.0822/.004583)

=17.9333Kg/sm^2 (hot fluid)

=(.07791/.0038)

=20.39Kg/sm^2 (cold fluid)

(d) Reynolds no. (R_e) =GD/μ

=(17.93*1.58)/.0000134

= 2120.35 (hot fluid)

=(20.39*1.58)/.0000134

= 2495.33 (cold fluid)

$Re<Re^*$(laminar flow)

$J=.53Re^{0.5}(l/D_h)^{-0.15}(\alpha)^{-0.14}$

$F=8.12Re^{-0.74}(l/D_h)^{-0.41}(\alpha)^{-0.02}$

$Re>Re^*+1000$(turbulent flow)

$J=.21Re^{-0.4}(l/D_h)^{-0.24}(t/D_h)^{0.02}$

$F=1.12Re^{-0.36}(l/D_h)^{-0.65}(t/D_h)^{0.17}$

Where

$Re^*=257(l/s)^{1.23}(t/l)^{0.58}D_h(t+1.328(Re/lD_h)^{-0.5})^{-7}$

=678.484

Since Re> Re*

$F=1.12(2120.35)^{-0.36}(1.5/1.58)^{-0.65}(.2/1.58)^{0.17}$

=.05180 (hot fluid)

F=0.04885 (cold fluid)

$J=.21(2120.35)^{-0.4}(1.5/1.58)^{-0.24}(.2/1.58)^{0.02}$

=.00954 (hot fluid)

$J=.00893$ (cold fluid)

$h=jc_p l/p_r^{(2/3)}$
$= 216.537$ (hot fluid)

$=230.26$ (cold fluid)

fin parameter
$M=\sqrt{2h/k_f t}=\sqrt{(2*216.537*1000)/(150*.2)}$

$= 120.148$ (hot fluid)

$=\sqrt{2*230.26*1000/(150*.2)}$

$= 123.89$ (cold fluid)

$Ml_f =Mb/2$
$=120.148*9.5/2=.0570$ (hot fluid)

$=123.89*9.5/2=.0588$ (cold fluid)

$n_f \quad =\tanh(ml)/ml$
$=\tanh(.0570)/.0570=.9039$ (hot fluid)

$=\tanh(.0588)/.0588$ (cold fluid)

$=.8985$ (cold fluid)

Overall efficiency=№ $=1-(A_f/A_s)(1- n_f)$
$=1-.8977(1-.9060)$
$=.9137$ (hot fluid)
$=1-.8977(1-.90)$
$=.9089$ (cold fluid)

(5)overall heat transfer coefficient(w/m^2-k)

(a) total area/separating wall area$(A_0/A_w)=(1-ft)/(1-A_f/A_w)$
$=(1-.71425*.2)/(1-.8977)$
$=8.3857\ m^2/m^3$

(b) overall thermal resistance

$(1/U_0)=(n_c w_c/n_h w_h(№_h h_h))+(aA_o/K_w A_w)+(1/№_c h_c)$
$=.0092\ m^2K/W$

$U_o = 110.688 W/m^2K$

(6) heat transfer area m/m^2

UA for heat exchanger = 1088 W/K

Required heat transfer area = 9.829 m^2

Required heat transfer area /length

$A/L=9.6435 = 4A_{min(ff)}/D_{eq}$

Required length of heat exchanger

L=1019 m

(7) pressure drop

$p\ /L=(fG^2/2\rho D_{eq})$

=3385.008 Pa (hot fluid)

=3821.286 Pa (cold fluid)

(8) final dimensions

Core length =1019 mm (without longitudinal heat conduction)

Core width =115 mm

Total width =115+2*6=127 mm

no of HP side =6

no of LP side =5

core height $=(n_c+n_h)*b+(n_c+n_h)*a$

=113.14mm

Total height =113.14+2*6=125.14 mm

EFEECT OF LONGITUDNAL HEAT CONDUCTION

Heat conduction area = A_w=core width*total height-free flow area of hot side
- free flow area of cold side

=115*104.7-3819-3055

=5987 mm^2

C_{min} = .07791*1043=81.26

FOS = 2.41

UA = UA_o*FOS

= 1088*2.41=2622.08W/K

NTU = UA/C_{min}

= (2622.08)/81.26=32.26

$\lambda = (K_W A_W)/LC_{min}$

=(150*.005987)/(2.41 *81.26)=.0108

Y = λ*NTU*C_R

=.0108*32.26*.94

=.3314

$\gamma=(1-C_R)/(1-C_R)(1+Y)$

=(1-.9478)/(1+.9478)(1+.3314)

=.0201

$\Phi=\gamma(Y/(1+Y))^{.5}((1+\gamma)Y/(1-\gamma(1+\gamma)Y))$

=.003

$\Psi=(1+\Phi)/(1-\Phi)$

=1.006

$r1=(1-C_R)*NTU/(1+\lambda NTU C_R)$

=1.265

$1-\varepsilon=(1-C_R)/(\Psi e^{r1}-C_R)$

=.0199

ε =.98

Heat exchanger area = 23.688 m^2

Heat exchanger length=2.45 m

Design of compact plate fin heat exchanger using Deepak & Maity correlations

(4) Heat transfer data specification

a. fin thickness (t)=.2 mm
b. fin frequency(f) =714.25 fin per meter
c. fin length (l)=1.5 mm
d. fin height (h)=9.3 mm
e. fin spacing (s)=(1/f)-t
= (1/714.25)-.2
= 1.2 mm plate thickness

f. plate thickness (b) = h + t
= 9.3+.2
=9.5 mm

g. free flow area (A_{ff}) =(s-t)h
=(1.2-.2)*9.3
=9.6*10-^{6} m^2

h. frontal area (A) =(h+t)(s+t)
=(9.3+.2)(1.2+.2)
=.0000133 m^2

i. heat transfer area (A_s)=2*h*l+2*s*l+2*h*l
=2*9.3*1.5+2*1.2*1.5+9.3*.2*2
=35.22 mm^2

j. Fin area (A_f) =2*h*l+2*h*l
=2*9.3*1.5+9.3*2*.2
=31.62 mm^2

k. eq. Dia. =D_h ==((2lh(s-t))/(ls+hl+ht)
=(2*1.5*9.3(1.2-.2))/(1.5*1.2+9.3*1.5+9.3*.2)
=1.58 mm

l. fin area/total surface area=(A_f /A_s)=31.62/35.22=.8977

m. frontal area ratio (σ)=A_f /A_s=9.3/13.3=.69924

n. α =h/s=9.3/1.2=7.75
o. δ =l/s=1.5/1.2=1.25

p. ν = t/s=.2/1.2=.166

(2.) DATA INPUT

material of the fin =Al

conductivity of the fin material(K_f)=150 W/mK

end plate of thickness=6 mm

end bars thickness=6 mm

	hot fluid	cold fluid
inlet temp	310 k	99.716 K
Outlet temp	124.26 K	301.54 K
Mass flow rate	.0822 Kg/s	.07791 Kg/s
Pressure inlet	8 bar	1.15 bar
Allowable pressure drop	.05 bar	.05 bar
Density at avg. temp	1.583	1.711

(3) ASSUMPTION

avg. wall temp	200 K	
width(w)	.2 m	.2m
no of layers	10	9
area between plate	.019 m^2	.0171m^2

A=(wbn)

For hot fluid =.2*.0095*10

For cold fluid =.2*.0095*9

Free flow area	.0132 m^2	.0119 m^2

(A_{ff}=A*σ)

For hot fluid =.019*.699

For cold fluid=.0171*.699

(4) CONVECTIVE HEAT TRNSFER CO-EFFICIENT

(a) bulk temp =(inlet temp+outlet temp)/2

=(310+124.26)/2

=217.13 (hot fluid)

= (301.54+124.26)/2

=200.628 (cold fluid)

(b) mean film temp.=(wall temp+bulktemp)/2

= (200+217.13)/2

=208.56 k (hot fluid)

=(200+200.628)/2

=200.314 (cold fluid)

Properties

	Hot fluid	cold fluid
Sp. Heat (c_p)	1043 J/Kg-K	1043 J/Kg-K
Viscosity(μ)	.0000134 N/m^2-s	.00001295 N/m^2- s
Predelt number	.74767	.75

(c) core mass velocity$(G)=m_{ff}/A_{ff}$

$G=(.0822/.0132)$

$=6.187$ Kg/sm^2 (hot fluid)

$=(.07791/.0119)$

$=6.58$ Kg/sm^2 (cold fluid)

(d) Reynolds no. $(R_e)=GD/\mu$

$=(6.18*.00158)/.0000139$

$=705.207$ (hot fluid)

$=(6.58*.00158)/.0000131$

$=796.014$ (cold fluid)

$Re^* = 1568.58*(\alpha)^{-.217}*(\delta)^{-1.433}*(\gamma)^{-.217}$

$=1077.74$

$= 1077.74$

$J=.18Re^{-0.42}(\alpha)^{0.288}(\delta)^{-.184}(\gamma)^{-.05}$

$=.02350$ (hot fluid)

$=.02209$ (cold fluid)

$Re^* = 648.23*(\alpha)^{-.06}*(\delta)^{.1}*(\gamma)^{-.196}$

$=832.874$(hot fluid)

$=832.874$(cold fluid)

$F=.32Re^{-0.286}(\alpha)^{0.221}(\delta)^{-.185}(\gamma)^{-.023}$

$=.08118$ (hot fluid)

$=.07523$ (cold fluid)

Pressure drop /length$=(4fG^2)/2\rho D_{eq}$

$=312.95$ N/m^3 (hot fluid)

$=2201.312$ N/m^3 (cold fluid)

Pressure drop $=387.061$N/m^2 (hot fluid)

$=2722.59$ N/m^2 (cold fluid)

$h=jc_p l/p_r^{(2/3)}$

$=194.118$ (hot fluid)

$=190.447$ (cold fluid)

fin parameter

$M=\sqrt{2h/k_f t}=\sqrt{(2*194.118*1000)/(150*.2)}$

$=113.75$ (hot fluid)

$=\sqrt{(2*190.447*1000)/(150*.2)}$

$= 112.67$ (cold fluid)

$Ml_f=Mb/2$

$=113.75*9.5/2=.5289$ (hot fluid)

$=112.67*9.5/2=.5239$ (cold fluid)

$n_f \;=\tanh(ml)/ml$

$=\tanh(.5289)/.5289=.9161$ (hot fluid)

$=\tanh(.5239)/.5239=.9175$ (cold fluid)

Overall efficiency=№ $=1-(A_f/A_s)(1-n_f)$

$=1-.8977(1-.9161)$

$=.9246$ (hot fluid)

$=1-.8977(1-.9175)$

$=.9259$ (cold fluid)

(5)overall heat transfer coefficient(w/m^2-k)

(a) total area/separating wall area(A_0/A_w)=$(1-ft)/(1-A_f/A_w)$

$=(1-.71425*.2)/(1-.8977)$

$=8.3857\ m^2/m^3$

(b) overall thermal resistance

$$(1/U_0)=(n_c w_c/n_h w_h(№_h h_h))+(aA_o/K_w A_w)+(1/№_c h_c)$$

$=.0107\ m^2K/W$

$U_o=93.20\ W/m^2K$

(6) heat transfer area m/m^2

FOS=1.91

UA for heat exchanger = 3479.92 W/K

Required heat transfer area =37.33 m^2

Required heat transfer area /length

$A/L=(4*.0035)*(.002125)\ m^2/m=30.188=4A_{min(ff)}/D_{eq}$

Required length of heat exchanger

L=1.237 m

(7) Pressure drop /length=$(4fG^2)/2\rho D_{eq}$

$=312.95\ N/m^3$ (hot fluid)

$=2201.312\ N/m^3$ (cold fluid)

Pressure drop =387.061N/m^2 (hot fluid)

$=2722.59\ N/m^2$ (cold fluid)

(8) final dimensions

Core length =1237 mm

Core width =200 mm

Total width =200+2*6=212 mm

no of HP side =10

no of LP side =9

core height $=(n_c+n_h)*b+(n_c+n_h)*a$

=195 mm

Total height =195+2*6=207 mm

EFEECT OF LONGITUDNAL HEAT CONDUCTION

Heat conduction area = A_w=core width*total height-free flow area of hot side - free flow area of cold side

=115*104.7-3819-3055

=5165 mm^2

C_{min} = .07791*1043=81.26

FOS = 1.91

UA = UA_o*FOS

= 1088*1.91=3479.92 W/K

NTU = UA/C_{min}

= (3479.92)/81.26=42.27

$\lambda = (K_W A_W)/LC_{min}$

=(150*.005165)/(1.236 *81.26)=.023

$Y = \lambda*NTU*C_R$

=.023*42.27*.94

=.9413

$\gamma=(1-C_R)/(1-C_R)(1+Y)$

=(1-.9478)/(1+.9478)(1+.9413)

=.0172

$\Phi=\gamma(Y/(1+Y))^{.5}((1+ \gamma)Y/(1- \gamma(1+ \gamma)Y))$

=.0116

$\Psi=(1+ \Phi)/(1- \Phi)$

=1.0236

$r1=(1-C_R)*NTU/(1+ \lambda NTUC_R)$

=1.4084

$1-\varepsilon=(1-C_R)/(\Psi e^{r1}-C_R)$

=.019

ε =.98

PLATE FIN HEAT EXCHANGER DESIGN USING MANGAHANIC CORRELATION

1) Heat Exchanger Design Specifications

a)	Type of heat exchanger :	Plain plate fin heat exchanger							
b)	Fin type	Offset Serrated							
c)	Fin thickness, t	0.0002	m	0.2	mm	0.0002	m	0.2	mm
d)	Fin frequency, f	714.25	fpm			714.25	fpm		
e)	Fin length, Lf	0.0015	m	1.5	mm	0.0015	m	1.5	mm
f)	Fin height, h	0.0093	m	9.3	mm	0.0093	m	9.3	mm
g)	Plate thickness, a	0.0008	m	0.8	mm	0.0008	m	0.8	mm
h)	Fin spacing, s	0.0012	m	1.2	mm	0.0012	m	1.2	mm
i)	Plate Spacing, b	0.0095	m	9.5	mm	0.0095	m	9.5	mm
j)	Free flow area per fin, a_{ff}	0.0000093	m^2			0.0000093	m^2		
k)	Frontal area per fin, a_{fr}	0.0000133	m^2			0.0000133	m^2		
l)	Heat transfer area, A_s	0.00003522	m^2			0.00003522	m^2		
m)	Fin area, A_f	0.00003162	m^2			0.00003162	m^2		
n)	Equivalent diameter,De	0.001584327	m	1.58432709	mm	0.001584327	m	1.584	mm
o)	Fin area /total surface area, A_f / A_s	0.897785349	m^2/m^2			0.897785349	m^2/m^2		
p)	Frontal area ratio,σ = Aff / Afr	0.69924812	m^2/m^2			0.69924812	m^2/m^2		
q)	$\alpha = h/s$	7.75				7.75			
r)	$\delta = l/s$	1.25				1.25			
s)	$\gamma = t/s$	0.166666667				0.166666667			
t)	Conductivity of fin material, K_f	150	W/m-K			150	W/m-K		
u)	End Plate Thickness	6	mm						
v)	End Bars/ End Bars Thickness	6	mm						

2) Data input:

	hot fluid:		cold fluid:	
Inlet temperature,	310	K	99.716	K
Outlet temperature,	124.26	K	301.54	K
Mass flow rate,	0.0822	kg/s	0.07791	kg/s
Pressure at inlet,	8	bar	1.15	bar
Allowable pressure drop	0.05	bar	0.05	bar
Density at average temperature,	1.583	Kg/m^3	1.711	Kg/m^3

3) Assumptions:

a)	Average wall temperature, Tw =	200	K				
b)	Width, W	0.115	m	115	mm	0.115	m
c)	Number of Layers	5				4	
a)	Area between plates	0.0054625	m^2			0.00437	
b)	Free flow area, A_{ff}	0.003819643	m^2			0.003055714	

4) Convective heat transfer coefficients

		hot fluid side		cold fluid side	
a)	Average or bulk temperature,Tavg	217.13	K	200.628	K

b)	The mean film temperature,Tm	208.565	K	200.314	K
c)	Properties at the mean film temperature				
	Specific heat, Cp	1043	J/Kg-K	1043	J/Kg-K
	Viscocity, μ	0.0000134	N/m^2-s	0.00001295	N/m2-sec
	Prandtl number, Pr	0.74767		0.75	
d)	Core mass velocity, G	21.52033661	kg/s-m^2	25.49649369	kg/s-m2
e)	The Reynold's number, Re	2544.421806		3119.288461	
f)	Critical Reynold's number Re*j	1077.742419		1077.742419	
g)	Critical Reynold's number Re*f	832.8747012		832.8747012	
h)	J	0.01265		0.01161	
i)	F	0.05341		0.05039	
j)	Pressure drop per length, Δp/L	4931.722799		6042.142505	
k)	Convective Heat transfer Coefficient, h	344.7043956		374.129065	
l)	The fin parameter,M	151.592523		157.9301671	
m)	ML_f	0.720064484		0.750168294	
n)	The fin effectiveness, ηf	0.856797214		0.846809108	
	The surface effectiveness, ηo	0.871434637		0.862467461	
5)	**Overall heat transfer coefficient,W/m2-K**				
a)	Total area/seperating surface (wall) area,Ao/Aw	8.385784167	m^2/m^2	8.385784167	m^2/m^2
b)	Ovearall thermal resistance,1/Uo	0.005807054	m^{2}K/W		
c)	Overall heat transfer coefficient,Uo	172.2043597	W/m^2-K		
6)	**Heat transfer surface area,m2**				
	UA for heat exchanger	1088	W/K		
	required heat transfer area, A	6.318074651	m^2		
	Required heat transfer area per length , A /L	7.714857143	m^2/m		
	The required length of the heat exchanger, L	0.818949014	m		
7)	**Pressure drop,Δp**				
a)	pressure drop,Δp	4038.829524	Pa	4948.206647	Pa
		0.040388295	bar	0.049482066	bar
8)	**Final Dimension**				
	Core Length	819	mm		
	Core Width	115	mm		
	Total Width	127	mm		
	No .of Layers HP side	5			
	No .of Layers LP side	4			
	Core Height	92.7	mm		
	Total Height	104.7	mm		
	Remarks	**Design is OK**			
9)	**Effect of logitudnal heat conduction**				
	Aw	0.005165143	m2		
	Cmin	81.26013			
	FOS	2.47			
	UA	2687.36	W/K		

NTU	33.0710768	
Λ	0.011641587	
Cr	0.9478	
Y	0.364902822	
Γ	0.019634706	
Φ	0.00380512	
Ψ	1.007639308	
r1	1.264786167	
1-ε	0.019911533	
E	0.980088467	
heat exchanger area	15.60564439	m2
length of heat exchanger	2.022804065	m2

PLATE FIN HEAT EXCHANGER DESIGN USING WIETING CORRELTION

1) Heat Exchanger Design Specifications

a)	Type of heat exchanger :	Plain plate fin heat exchanger							
b)	Fin type	Offset Serrated							
c)	Fin thickness, t	0.0002	m	0.2	mm	0.0002	m	0.2	mm
d)	Fin frequency, f	714.25	fpm			714.25	fpm		
e)	Fin length, Lf	0.0015	m	1.5	mm	0.0015	m	1.5	mm
f)	Fin height, h	0.0093	m	9.3	mm	0.0093	m	9.3	mm
g)	Plate thickness, a	0.0008	m	0.8	mm	0.0008	m	0.8	mm
h)	Fin spacing, s	0.0012	m	1.2	mm	0.0012	m	1.2	mm
i)	Plate Spacing, b	0.0095	m	9.5	mm	0.0095	m	9.5	mm
j)	Free flow area per fin , a_{ff}	0.0000093	m^2			0.0000093	m^2		
k)	Frontal area per fin, a_{fr}	0.0000133	m^2			0.0000133	m^2		
l)	Heat transfer area, A_s	0.00003522	m^2			0.00003522	m^2		
m)	Fin area , A_f	0.00003162	m^2			0.00003162	m^2		
n)	Equivalent diameter,De	0.002125714	m	2.12571429	mm	0.002125714	m	2.126	mm
o)	Fin area /total surface area, A_f /A_s	0.897785349	m^2/m^2			0.897785349	m^2/m^2		
p)	Frontal area ratio,σ =Aff / Afr	0.69924812	m^2/m^2			0.69924812	m^2/m^2		
q)	α = s/h	0.129032258				0.129032258			
r)	δ = t/l	0.133333333				0.133333333			
s)	γ=t/s	0.166666667				0.166666667			
t)	Conductivity of fin material, K_f	150	W/m-K			150	W/m-K		
u)	End Plate Thickness	6	Mm						
v)	End Bars/ End Bars Thickness	6	Mm						

2) Data input:

hot fluid:			**cold fluid:**	
Inlet temperature,	310	K	99.716	K
Outlet temperature,	124.26	K	301.54	K
Mass flow rate,	0.0822	kg/s	0.07791	kg/s
Pressure at inlet,	8	Bar	1.15	bar
Allowable pressure drop	0.05	Bar	0.05	bar
Density at average temperature,	1.583	Kg/m^3	1.711	Kg/m^3

3) Assumptions:

a)	Average wall temperature, Tw =	200	K				
b)	Width, W	0.115	M	115	mm	0.115	m
c)	Number of Layers	7				6	
a)	Area between plates	0.0076475	m2			0.006555	
b)	Free flow area , A_{ff}	0.0053475	m^2			0.004583571	

4) Convective heat transfer coefficients

hot fluid side	cold fluid

				side	
a)	Average or bulk temperature, Tavg	217.13	K	200.628	K
b)	The mean film temperature, Tm	208.565	K	200.314	K
c)	Properties at the mean film temperature				
	Specific heat, Cp	1043	J/Kg-K	1043	J/Kg-K
	Viscocity, μ	0.0000134	N/m^2-s	0.00001295	N/m2-sec
	Prandtl number, Pr	0.74767		0.75	
d)	Core mass velocity, G	15.371669	kg/s-m^2	16.99766246	kg/s-m2
e)	The Reynold's number, Re	2438.490776		2790.12926	
f)	Critical Reynold's number Re1	1000		1000	
g)	Critical Reynold's number Re2	2000		2000	
h)	j	0.01244		0.01184	
i)	f	0.09012		0.08775	
j)	Pressure drop per length, Δp/L	3164.2519		3485.41642	
k)	Convective Heat transfer Coefficient, h	242.0763791		254.2087757	
l)	The fin parameter,M	127.037102		130.1816105	
m)	ML_f	0.603426234		0.61836265	
n)	The fin effectiveness, ηf	0.894033241		0.889424648	
	The surface effectiveness, ηo	0.904864596		0.900727069	
5)	**Overall heat transfer coefficient,W/m2-K**				
a)	Total area/seperating surface (wall) area,Ao/Aw	8.385784167	m^2/m^2	8.385784167	m^2/m^2
b)	Ovearall thermal resistance,1/Uo	0.008325123	m^2K/W		
c)	Overall heat transfer coefficient,Uo	120.1183481	W/m^2-K		
6)	**Heat transfer surface area,m2**				
	UA for heat exchanger	1088	W/K		
	required heat transfer area, A	9.057733619	m^2		
	Required heat transfer area per length , A /L	8.625	m^2/m		
	The required length of the heat exchanger, L	1.050172014	m		
7)	**Pressure drop,Δp**				
a)	pressure drop,Δp	3323.00879	Pa	3660.286781	Pa
		0.033230088	bar	0.036602868	bar
8)	**Final Dimension**				
	Core Length	1051	mm		
	Core Width	115	mm		
	Total Width	127	mm		
	No .of Layers HP side	7			
	No .of Layers LP side	6			
	Core Height	133.58	mm		
	Total Height	145.58	mm		

Remarks	**Design is OK**	

9) **Effect of logitudnal heat conduction**

Aw	0.006810629	m2
Cmin	81.26013	
FOS	2.5	
UA	2720	W/K
NTU	33.4727498	
λ	0.011961847	
Cr	0.9478	
y	0.37949523	
γ	0.019427009	
Φ	0.003971804	
Ψ	1.007975284	
r1	1.266606438	
1-ε	0.019853256	
E	0.980146744	
heat transfer area	22.64433405	m2
length of heat exchanger	2.625430034	m

PLATE FIN HEAT EXCHANGER DESIGN USING JOSHI &WEBB CORRELATION

1)	**Heat Exchanger Design Specifications**									
a)	Type of heat exchanger :	Plain plate fin heat exchanger								
b)	Fin type	Offset Serrated								
c)	Fin thickness, t	0.0002	m	0.2	mm	0.0002	m	0.2	mm
d)	Fin frequency, f	714.25	fpm			714.25	fpm		
e)	Fin length, Lf	0.0015	m	1.5	mm	0.0015	m	1.5	Mm
f)	Fin height, h	0.0093	m	9.3	mm	0.0093	m	9.3	Mm
g)	Plate thickness, a	0.0008	m	0.8	mm	0.0008	m	0.8	Mm
h)	Fin spacing, s	0.0012	m	1.2	mm	0.0012	m	1.2	Mm
i)	Plate Spacing, b	0.0095	m	9.5	mm	0.0095	m	9.5	Mm
j)	Free flow area per fin, a_{ff}	0.0000093	m^2			0.0000093	m^2		
k)	Frontal area per fin, a_{fr}	0.0000133	m^2			0.0000133	m^2		
l)	Heat transfer area, A_s	0.00003522	m^2			0.00003522	m^2		
m)	Fin area, A_f	0.00003162	m^2			0.00003162	m^2		
n)	Equivalent diameter,De	0.001584327	m	1.58432709	mm	0.001584327	m	1.584	Mm
o)	Fin area /total surface area, A_f / A_s	0.897785349	m^2/m^2			0.897785349	m^2/m^2		
p)	Frontal area ratio, σ = Aff / Afr	0.69924812	m^2/m^2			0.69924812	m^2/m^2		
q)	$\alpha = s/h$	0.129032258				0.129032258			
r)	$\delta = t/l$	0.133333333				0.133333333			
s)	$\gamma = t/s$	0.166666667				0.166666667			
t)	Conductivity of fin material, K_f	150	W/m-K			150	W/m-K		
u)	End Plate Thickness	6	mm						
v)	End Bars/ End Bars Thickness	6	mm						

2)	**Data input:**						
	hot fluid:				**cold fluid:**		
	Inlet temperature,	310	K			99.716	K
	Outlet temperature,	124.26	K			301.54	K
	Mass flow rate,	0.0822	kg/s			0.07791	kg/s
	Pressure at inlet,	8	bar			1.15	bar
	Allowable pressure drop	0.05	bar			0.05	bar
	Density at average temperature,	1.583	Kg/m^3			1.711	Kg/m^3
3)	**Assumptions:**						
a)	Average wall temperature, Tw =	200	K				
b)	Width, W	0.115	m	115	mm	0.115	m
c)	Number of Layers	6				5	
a)	Area between plates	0.006555	m^2			0.0054625	
b)	Free flow area, A_{ff}	0.004583571	m^2			0.003819643	

4)	**Convective heat transfer coefficients**				
		hot fluid side		cold fluid side	
a)	Average or bulk temperature,Tavg	217.13	K	200.628	K
b)	The mean film temperature,Tm	208.565	K	200.314	K

c)	Properties at the mean film temperature				
	Specific heat, Cp	1043	J/Kg-K	1043	J/Kg-K
	Viscocity, μ	0.0000134	N/m^2-s	0.00001295	N/m2-sec
	Prandtl number, Pr	0.74767		0.75	
d)	Core mass velocity, G	17.93361384	kg/s-m^2	20.39719495	kg/s-m2
e)	The Reynold's number, Re	2120.351505		2495.430769	
f)	Critical Reynold's number Re1	681.1382467		690.9666918	
g)	Critical Reynold's number Re2	2000		2000	
h)	J	0.00954		0.00893	
i)	F	0.05180		0.04885	
j)	Pressure drop per length, Δp/L	3321.485567		3748.892456	
k)	Convective Heat transfer Coefficient, h	216.5340502		230.267632	
l)	The fin parameter,M	120.1482557		123.8998606	
m)	ML_f	0.570704215		0.588524338	
n)	The fin effectiveness, ηf	0.903929313		0.898575355	
	The surface effectiveness, ηo	0.913749145		0.90894244	
5)	**Overall heat transfer coefficient,W/m2-K**				
a)	Total area/seperating surface (wall) area,Ao/Aw	8.385784167	m^2/m^2	8.385784167	m^2/m^2
b)	Ovearall thermal resistance,1/Uo	0.009034333	m^2K/W		
c)	Overall heat transfer coefficient,Uo	110.6888519	W/m^2-K		
6)	**Heat transfer surface area,m2**				
	UA for heat exchanger	1088	W/K		
	required heat transfer area, A	9.829354825	m^2		
	Required heat transfer area per length , A /L	9.643571429	m^2/m		
	The required length of the heat exchanger, L	1.019264999	m		
7)	**Pressure drop,Δp**				
a)	pressure drop,Δp	3385.473984	Pa	3821.114866	Pa
		0.03385474	bar	0.038211149	bar
8)	**Final Dimension**				
	Core Length	1020	mm		
	Core Width	115	mm		
	Total Width	127	mm		
	No .of Layers HP side	6			
	No .of Layers LP side	5			
	Core Height	113.14	mm		
	Total Height	125.14	mm		

Remarks **Design is OK**

effect of logitudnal heat conduction

Aw	0.005987886
Cmin	81.26013
FOS	2.41
UA	2622.08
NTU	32.2677308
Λ	0.010836451
Cr	0.9478
Y	0.331415035

Γ	0.020128559
Φ	0.003418493
Ψ	1.006860439
r1	1.265101793
1-ε	0.019923945
E	0.980076055
Area	23.68874513
Length	2.456428648

PLATE FIN HEAT EXCHANGER 1 DESIGN (Deepak corelation)

1) Fluid Data input:

Working Fluid	Pres	Temp	Dens	H	Cp	Viscosity	Prandtl#
Nitrogen	bar	K	kg/m3	KJ/kg	KJ/kg K	Pa-s	
HP Inlet	8	310.00	8.70041	320.436	1.0507	0.0000185	0.71675
HP Exit	7.95	120.45	24.59257	114.67011	1.21522	0.0000084	0.7680636
HP Mean	7.975	215.225	12.64345	220.14748	1.0706	0.0000139	0.7181032
LP Inlet	1.15	100.74	3.93202	102.49171	1.0705	0.0000068	0.7254834
LP Exit	1.1	305.80	1.21202	317.47858	1.04036	0.0000182	0.7162658
LP Mean	1.125	203.27	1.86879	210.59691	1.04589	0.0000131	0.7136091
	1.1	310	1.19555	321.84803	1.04033	0.0000184	0.7163585

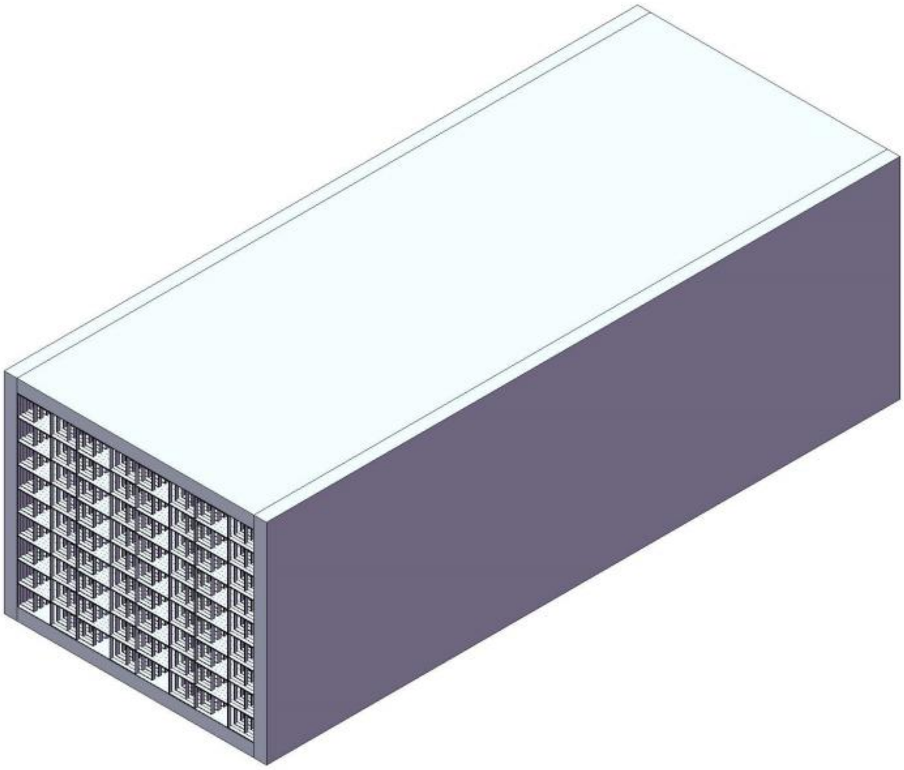
Mass Flow rate kg/s (HP Side)	Mass Flow rate kg/s (LP side)	Ch W/K	Cc W/K	Cmin W/K	Cmax W/K	Cr
0.0822	0.0787	88.00332	82.311543	82.311543	88.00332	0.93532316

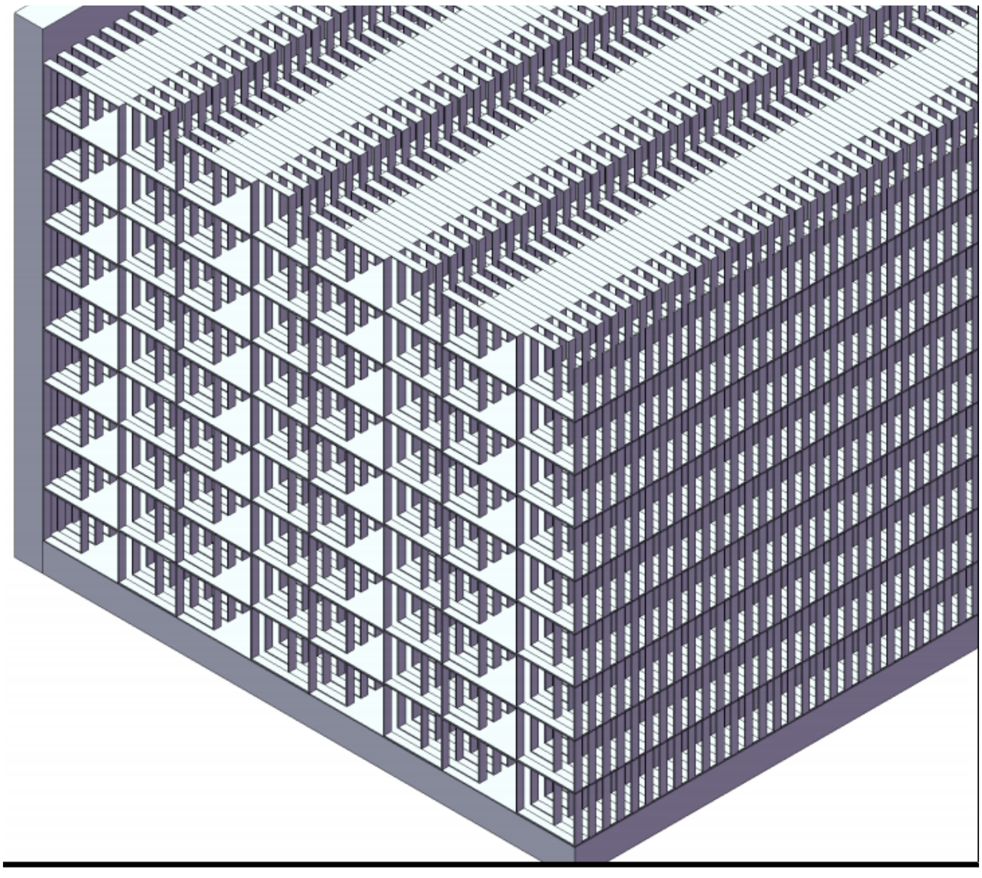
ε	NTU, Req	UA, W/K	LMTD	Q, Watts	UA, LMTD
0.9801	22.1348	1821.94954	10.032072	16913.95616	1685.9884

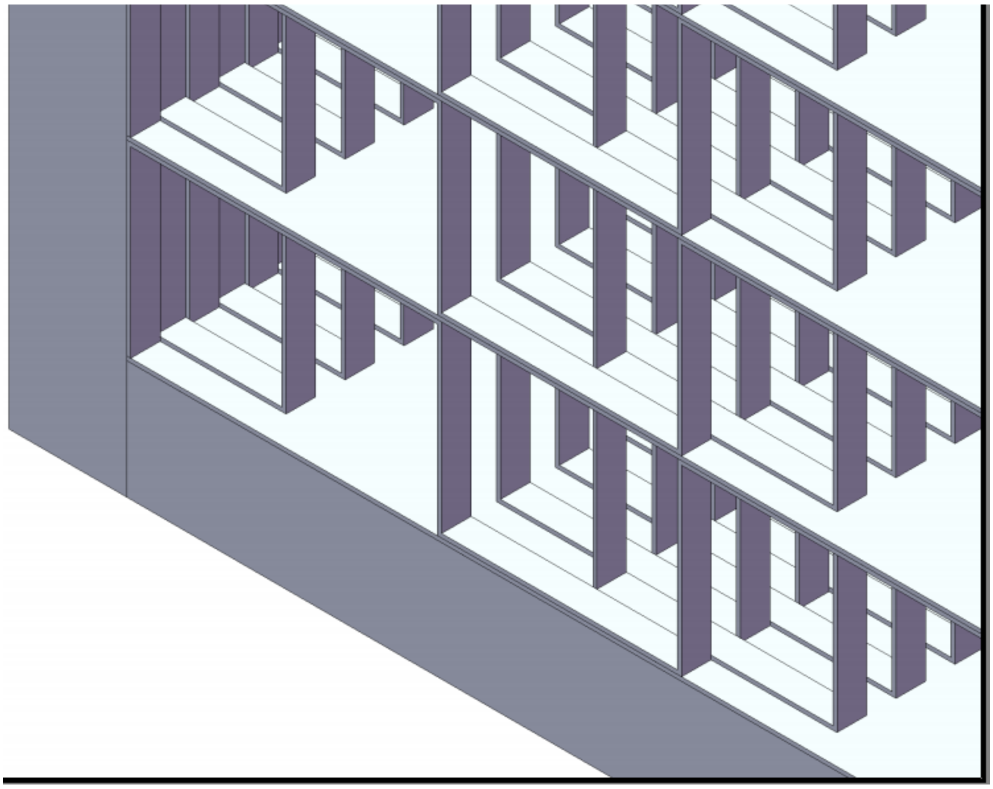
2)	**Heat Exchanger Fin Specifications**	**H.P Side**				**L.P. Side**		
a)	Type of heat exchanger :	Plain plate fin heat exchanger						
b)	Fin type	Offset Serrated						
c)	Fin thickness, t	0.0002	m	0.2	mm	0.0002	M	0.2 mm
d)	Fin frequency, f	714.25	fpm			714.25	Fpm	
e)	Fin length, L_f	0.0015	m	1.5	mm	0.0015	M	1.5 mm
f)	Fin height, h	0.0093	m	9.3	mm	0.0093	M	9.3 mm
g)	Plate thickness, a	0.0008	m	0.8	mm	0.0008	M	0.8 mm
h)	Fin spacing, s= (1-ft)/f	0.0012	m	1.2	mm	0.0012	M	1.2 mm
i)	Plate Spacing, b=h+t	0.0095	m	9.5	mm	0.0095	M	9.5 mm
j)	Free flow area per fin, $a_{ff}=(s-t)h$	0.0000093	m^2			0.0000093	m^2	
k)	Frontal area per fin, $a_{fr}=(s+t)(h+t)$	0.0000133	m^2			0.0000133	m^2	
l)	Heat transfer area, $a_s=2hl+2ht+2sl$	0.00003522	m^2			0.00003522	m^2	
m)	Fin area, $a_f=2hl+2ht$	0.00003162	m^2			0.00003162	m^2	
n)	Equivalent diameter, $D_e=2(s-t)hl/(2hl+2ht+2sl)$	0.001584327	m	1.58432709	mm	0.001584327	M	1.58432709 mm
o)	Fin area /total surface area, a_f/a_s	0.897785349	m^2/m^2			0.897785349	m^2/m^2	
p)	Frontal area ratio, $\sigma = a_{ff}/a_{fr}$	0.69924812	m^2/m^2			0.69924812	m^2/m^2	
q)	$\alpha = h/s$	7.75				7.75		

r)	$\delta = l/s$	1.25				1.25	
s)	$\gamma = t/s$	0.166666667				0.166666667	
t)	Conductivity of fin material, K_f	150	W/m-K			150	W/m-K
u)	End Plate Thickness	6	mm				
v)	End Bars/ End Bars Thickness	6	mm				
3)	**Assumptions:**						
a)	Width, W	0.2	m	200	mm	0.2	M
b)	Number of Layers, n	10				9	
a)	Total Area between plates A_{fr}= b*n*W	0.0190	m^2			0.0171	m^2
b)	Total Free flow area aff =σ * A_{fr}	0.013285714	m^2			0.011957143	m^2
4)	**Convective heat transfer coefficients**						
d)	Core mass velocity, G= m_f/A_{ff}	6.187096774	kg/s-m^2			6.581839904	kg/s-m2
e)	The Reynold's number, Re=G*D_e /μ	705.2075546				796.0142933	
f)	Critical Reynold's number Re*j	1077.742419				1077.742419	
g)	j factor	0.02350				0.02209	
h)	Convective h t c ,h = (j Cp G)/$pr^{(2/3)}$	194.1181841				190.4474043	
i)	The fin parameter,M =sqrt(2 h/K_f t)	113.7594492				112.6787186	
j)	ML_f	0.528981439				0.523956041	
k)	The fin effectiveness, ηf=tanhML /ML	0.91610419				0.917534073	
l)	The surface effectiveness, ηo	0.924679571				0.925963299	
5)	**Overall heat transfer coefficient,W/m2-K**						
a)	Total area/seperating surface (wall) area,Ao/Aw	8.385784167	m^2/m^2			8.385784167	m^2/m^2
b)	Ovearall thermal resistance,1/Uo	0.01072936	m^2K/W				
c)	Overall heat transfer coefficient,Uo	93.20220081	W/m^2-K				
6)	**Heat transfer surface area,m2**					**Without Longitudinal Conduction**	
	Factor of Safety	1.91					
	Design N_{tu} for heat exchanger	42.277468				22.1348	
	Design UA for heat exchanger	3479.923625	W/K			1821.949542	W/K
	Required heat transfer area, A	37.33735464	m^2			19.54835322	m^2

	Required heat transfer area per length , A /L	30.18857143	m^2/m	30.18857143	m^2/m
	The required length of the heat exchanger, L	1.23680429	m	0.647541513	m
7)	**Pressure drop,Δp**				
	Critical Reynold's number				
a)	Re*f	832.8747012		832.8747012	
	f factor	0.08188		0.07523	
	Pressure drop per length, $\Delta p/L=4f\,G^2/(2\,\rho\,D_e)$	312.9525781		2201.312508	
	pressure drop,Δp	387.0610911	Pa	2722.592753	Pa
		0.003870611	bar	0.027225928	bar
8)	**Final Dimension**			**Without Longitudinal Conduction**	
	Core Length	1237	mm	648	mm
	Core Width	200	mm		
	Total Width	212	mm		
	No .of Layers HP side	10			
	No .of Layers LP side	9			
	Core Height	195	mm		
	Total Height	207	mm		
9)	**Longitudinal heat conduction**				
	Conduction area	0.016157143			
	λ=KwAw/LCmin	0.02380642			
	Y=λNtuCr	0.941379538			
	γ=(1-Cr)/((1+Cr)*(1+Y)	0.017214122			
	Φ=γ(Y/(1+Y))^0.5*Y*(1+γ)/(1-γ(1+γ))	0.011670991			
	Ψ=(1+Φ)/(1-Φ)	1.023617623			
	r1=(1-Cr)Ntu/(1+λ*Ntu*Cr)	1.408469158			
	1-ε=(1-Cr)/(Ψexp(r1)-Cr)	0.019894718			
	ε	0.980105282			







CONCLUSION

Dimension of heat exchanger is given by

(1) By Mangahic correlation

Core length =2022 mm
Core width =115 mm
Total width =115+2*6=127 mm
No. of HP side =5
No. of LP side =4
Core height =92.7 mm
Total height =92.7+2*6=104.7 mm

(2) By Wieting correlation

Core length =2625 mm
Core width =115 mm
Total width =115+2*6=127 mm
No. of HP side =7
No. of LP side =6
Core height =133.58mm
Total height =133.58+2*6=145.58 mm

(3) By Joshi & Webb correlation

Core length =2456 mm
Core width =115 mm
Total width =115+2*6=127 mm
No. of HP side =6
No. of LP side =5
Core height =113.14mm
Total height =113.14+2*6=125.14 mm

(4) By Deepak & Maity correlation

Core length =1237 mm
Core width =200 mm
Total width =200+2*6=212 mm
No. of HP side =10
No.of LP side =9
Core height =195 mm
Total height =195+2*6=207 mm

• Contribution of this Design

• Possibilities of Future Work

Contribution of this Design

Plate Fin heat exchangers have already made a mark on the technology of the twentieth century. A variety of equipment – from automobiles to aircrafts, considers them essential, while others are adopting then for their superior performance. Still, the technology has remained largely proprietary. Driven by industrial needs and international sanctions, our country has initiated a multi–pronged research programme on this challenging subject. This design constitutes a small component of this effort.

Issues related to materials, manufacturing techniques and design approaches remain crucial to widespread application of plate fin heat exchangers. Heat transfer and flow friction characteristics of plate fin surfaces , however, will play the most vital role in its success. There is a shortage of experimental data and all existing correlations essentially represent the same basic information.

The primary contribution of this design is to developing design of compact heat exchanger by combining computational and experimental data.

Experiments on heat transfer over plate fin surfaces are expensive and difficult Direct numerical simulation (DNS) and comparable numerical techniques need computing resources beyond the affordability of most heat exchanger designers. Under these circumstances the approach taken in this design provides a workable solution.

4.2 Possibilities of Future Work

With physical constraints on time and resources, we have not been able to address to some aspects of the problem which have a strong symbiotic relationship with the material covered in this design. Among the most obvious topics are:

1. Plain fins of non-rectangular geometry – triangular, trapezoidal and comparable shapes,
2. Offset strip fin in hard way configuration –
3. Herringbone fins
4. Other fin types such as perforated plain fins and louver fins. The louver fin, particularly, can offer substantial computational challenges.

Availability of better heat transfer and flow friction correlations and increased confidence in the results are expected to stimulate the application of these fin geometries .

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