CHAPTER 2

LITERATURE REVIEW

There are several promising developments going on in the field of compact plate fin heat exchangers for several applications. The major focus of the review is the research work carried out in the field of off set strip fins and wavy fins. The studies on the effect of flow maldistribution, which arises especially in aerospace applications due to sharp changes in flow direction, are also highlighted. The review is summarized under the following broad sections:

- Inlet fluid flow non-uniformity.
- Wavy fins characteristics.
- Offset fins characteristics.

2.1 FLOW NON-UNIFORMITY ANALYSIS IN THE COMPACT HEAT EXCHANGERS

baffle plates design were placed at the inlet of the core for improvement in flow distribution based on the CFD study. Finally, they proved that flow non-uniformity has been drastically reduced due to the implementation of baffle plates. To the end, \( j \) and \( f \) data for some of the fins were presented.

Wen et al (2006) have investigated flow characteristics of flow field in the entrance of a plate-fin exchanger by means of Particle Image Velocimetry (PIV). Based on experiments, they suggested that punched baffle could effectively improve fluid flow distribution in the header.

Zhang and Yanzhong (2003) have investigated the flow non-uniformity in a plate-fin heat exchanger by CFD software. Based on the investigation, two modified headers with a two stage distributing structure were proposed to reduce the flow non-uniformity. They proved that the fluid flow distribution in plate-fin heat exchangers (PFHE) is more uniform if the ratios of outlet and inlet equivalent diameters for both headers are equal.

Anjun et al (2003) introduced the concept of second header installation. By experimentation, they proved that the performance of flow distribution in PFHE was effectively improved by the optimum design of the both header configurations.

Ranganayakulu and Panigrahi (2001) analyzed a cross flow two pass plate-fin compact heat exchanger, accounting for the effects of non-uniform inlet fluid flow due to headers using a finite element method. Using inlet flow non-uniformity models, the exchanger effectiveness and pressure drops and its deterioration due to the effects of flow non-uniformity were calculated for design case of heat exchanger. Based on the flow non-uniformity effects, they modified the exchanger hot side inlet and outlet headers to improve the exchanger effectiveness and reduce the pressure drop.
Lalot et al (1999) investigated the effect of flow non-uniformity on the performance of plate heat exchangers. They found the optimum location of perforated grid in the inlet header and observed reverse flow with poor header configuration. Most of previous research mainly investigated the effect of flow non-uniformity on the heat exchanger performance deterioration based on their specific flow mal-distribution cases.

Ranganayakulu et al (1997) and Ranganayakulu and Seetharamu (1999) investigated the effect of two-dimensional non-uniform flow distribution at inlet on both hot and cold fluid sides of cross flow plate-fin heat exchangers using a finite element model. It was found that the performance deteriorations and variations in pressure drops are quite significant in some typical applications due to fluid flow non-uniformity.

Table 2.1 Comparison of available methods to reduce the non-uniformity of flow

<table>
<thead>
<tr>
<th>Sl.No.</th>
<th>Authors</th>
<th>Method used to reduce flow non-uniformity</th>
<th>Type of heat exchanger</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Ranganayakulu et al (2007, 2006)</td>
<td>Baffle plate and inlet pipe orientation</td>
<td>PFHE</td>
</tr>
<tr>
<td>3</td>
<td>Zhang and Yanzhong (2003)</td>
<td>Two stage distribution structure</td>
<td>&quot;</td>
</tr>
<tr>
<td>6</td>
<td>Ranganayakulu et al (1997), Ranganayakulu and Seetharamu (1999)</td>
<td>Improved design of existing header</td>
<td>PFHE</td>
</tr>
</tbody>
</table>
The methods adopted by the researchers to reduce the non-uniformity of fluid flow are compared in Table 2.1. However, the efficiency of the each method cannot be compared, because each researcher has used different geometry of the heat exchanger and the analysis was carried out at different Reynolds number. In order to choose the best method, the flow non-uniformity analysis is to be carried out on a typical heat exchanger at a particular Reynolds number by using all the available methods or combination of any two or three methods. By this way, it is definitely possible to improve the flow uniformity further at the inlet of the compact heat exchangers.

2.2 HEAT TRANSFER SURFACE - WAVY FINS

The various research and developmental activities involved in the area of wavy fins as shown in Figure 2.1 are highlighted in this section.

Figure 2.1 Wavy fin (a) Photographic view (b) & (c) dimensional notations
2.2.1 Comparison of wavy and parallel plate channel

Comparison of heat transfer enhancement in the wavy channel with respect to a conventional parallel-plate channel has been reported by Leonardo Goldstein and Sparrow (1977), O’Brien and Sparrow (1982), Wang and vanka (1995), Gradeck et al (2005), and Sang Dong Hwanget et al (2006). O’Brien and Sparrow (1982) reported that over the range of Reynolds number (1500 - 25,000), the enhancement of heat transfer as compared to a conventional parallel-plate channel was about a factor of 2.5. Friction factors obtained from axial pressure distributions were virtually independent of the Reynolds number and equal to 0.57, a value appreciably greater than that for unidirectional duct flows.

Analog conclusions were reported in the works of Gradeck et al (2005). They showed that the heat transfer enhancement of about 2 times for the range of Reynolds number (0-7500). However, Wang and vanka (1995) and Leonardo Goldstein and Sparrow (1977) suggested that in the steady-flow regime, the average Nusselt numbers for the wavy wall channel were only slightly larger than those for a parallel-plate channel. On the other hand, in the transitional and turbulent flow regimes, the enhancement of heat transfer was about a factor of 2.5. Friction factors for the wavy channel were about twice those for the parallel-plate channel in the steady-flow region, and remained almost constant in the transitional regime. These authors’ statement is giving contradiction with O’Brien and Sparrow (1982), Gradeck et al (2005) at very low Reynolds number. According to Sang Dong Hwanget et al (2006), the results showed that for the wavy duct, high performance factors of 2.2 were obtained at a low Reynolds number of 1000 due to relatively higher mass transfer enhancement than increase in pressure loss. But the performance factor was decreased gradually at \( Re \geq 1000 \), because the secondary vortices disappear and flow separation/reattachment flow characteristics occurred.
These authors statement is more or less matching with O’Brien and Sparrow (1982) and Gradeck et al (2005) at the low Reynolds number, but gives a contradiction at high Reynolds numbers with all above authors. Hence, the performance study of wavy fin needs further research to eradicate these contradictions.

2.2.2 Turbulence models

Accuracy of the various turbulence models for the numerical simulation of wavy fin have been discussed in the works of Manabu Horiuchi et al (1982), Amano (1985, 2000), Amano et al (1987) and Patel et al (1991). According to Patel et al (1991), the two layer turbulence model of Chen and Patel (1988) appears to capture most of the important physical features of such flows. Another study (Amano 1985) revealed that the slope of the Nusselt number on the Reynolds number obtained using standard k-ε model with a special three-layer near-wall agrees with the experimental data of Izumi et al (1982). However, the computed Nusselt numbers in the turbulent flow regime were 10-20 percent lower than the measured values. Their further studies (Amano et al 1987 and Amano 2000) showed that the average Nusselt numbers computed by using the Reynolds Stress Model (RSM) were about 20 percent higher than those obtained by using the k-ε model, and consequently agreed well with experimental data. This is because the turbulence level predicted in the re-circulating region by the k-ε model was much lower than the results of Reynolds-stress model, thus resulting in lower heat transfer rates in the channel.

In addition, the RSM can take non-isotropic effects into account, which were strong in the recirculation region. Later, Manabu Horiuchi et al (2006) made a numerical simulation of turbulent flow with the RANS turbulent models. The models were the two-layer, the low Reynolds number
The simulation results were compared with those of the DNS and the prediction accuracy was discussed. The above comparisons reveal that sophisticated models such as the V2F model and the non-linear low $k - \varepsilon$ model predicted more precisely the influence of flow in the recirculation region on the wavy wall.

2.2.3 Geometrical parameters

The effects of different non-dimensional geometrical parameters such as $\alpha = S/H$, $\gamma = 4a/\lambda$, $\eta = S/2A$ and $2A/H$ on heat transfer and pressure drop have been reported in Reza Motamed Ektesabi et al (1987), Wang and Chen (2002), Metwally and Manglik (2004), Jiehai Zhang et al (2004) and Manglik et al (2005). The ranges of studies for these parameters are summarized in the Table 2.2.

Table 2.2 Range of parametric studies carried out on geometrical parameters of wavy fins

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Authors</th>
<th>Range of study</th>
<th>Reynolds/ Prandtl number</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Metwally and Manglik (2004)</td>
<td>$0 &lt; \gamma &lt; 1.0$ for $\eta = 1.0$</td>
<td>$10 &lt; \text{Re} &lt; 1000$ $\text{Pr} = 5$, $35$, and $150$</td>
</tr>
<tr>
<td>2</td>
<td>Wang and Chen (2002)</td>
<td>$0 &lt; a/\lambda &lt; 0.5$</td>
<td>$100 &lt; \text{Re} &lt; 700$ $\text{Pr} = 0.71$ and $6.93$</td>
</tr>
<tr>
<td>3</td>
<td>Jiehai Zhang et al (2004)</td>
<td>$0.125 \leq \gamma \leq 0.5$ and $0.1 \leq \eta \leq 3.0$</td>
<td>$10 &lt; \text{Re} &lt; 1000$ $\text{Pr} = 0.7$</td>
</tr>
<tr>
<td>4</td>
<td>Manglik et al (2005)</td>
<td>$0.240 \leq \alpha \leq 0.968$, $0.303 \leq \eta \leq 1.22$ for constant $\gamma = 0.2667$</td>
<td>$10 &lt; \text{Re} &lt; 1000$ $\text{Pr} = 0.7$</td>
</tr>
<tr>
<td>5</td>
<td>Reza Motamed Ektesabi et al (1987)</td>
<td>Fin height (H) = 3, 5, 10, 15 &amp; 20 mm and Wave amplitude (2A) = 10, 15, &amp; 20 mm</td>
<td>$300 &lt; \text{Re} &lt; 40,000$ $\text{Pr} = 6.93$</td>
</tr>
</tbody>
</table>
The above survey reveals the following conclusions.

- The optimum \((\gamma / \psi)\) performance is obtained for corrugation geometries in the range \(0.3 \leq \gamma (4a/\lambda) \leq 0.6\). In the non-swirl flow regime, the enlarged surface area of the corrugated-plate is solely responsible for the enhancement and the thermal benefits are significantly less.

- The amplitude of the Nusselt number and skin friction factor are increased with an increase on the amplitude wavelength ratio. However, at a sufficiently larger value of amplitude

## Table 2.2  (Continued)

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Authors</th>
<th>Range of study</th>
<th>Reynolds/ Prandtl number</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>Gschwind et al (1995)</td>
<td>1.8 mm (\leq s \leq 6.4) mm and (s/a = 1, 1.5, 2.5, 2.9, 3.2) and 3.5 for constant (\lambda = 26) mm, amplitude (a = 1.825) mm, the dimensionless wavelength (\lambda/a = 14.25), the duct length and width was 400mm and 150 mm.</td>
<td>50 (\leq \text{Re} \leq 10,000) with air and in a smaller range with water as working fluid. The Reynolds number (related to the duct height)</td>
</tr>
<tr>
<td>7</td>
<td>Patel et al (1991)</td>
<td>2(a/\lambda) = 0.02, 0.10, and 0.40. Where 2(a) and (\lambda) indicated the height and length of the cavity respectively.</td>
<td>(\text{Re} = 8610) and (\text{Re} = 12, 800)</td>
</tr>
<tr>
<td>8</td>
<td>Sparrow et al (1983)</td>
<td>--</td>
<td>2000 (&lt;\text{Re}\ &lt; 27,000. 4 &lt; \text{Pr}\ &lt; 12.</td>
</tr>
</tbody>
</table>
| 9       | Hang Scok Choi et al (2005) | wave amplitude was changed in three steps 0.01, 0.05 and 0.1 | \(\text{Re} = 6760\)  
\(\text{Pr} = 0.7\) |
| 10      | Gong-Nan et al (2006) | Wave length = 4, 4.5 and 6mm, wavy pitches = 9, 13 and 15 mm and channel width = 1.5, 2 and 3mm. | 100 \(<\text{Re}\ < 1100\.  \text{Pr} = 0.7\) |
wavelength ratio, the corrugated channel acts as an effective heat transfer device, especially, at higher Reynolds number.

- The heat transfer performance in the swirl flow regime (Re ≈ 600) is found to be enhanced considerably for all γ and η > 0.25, when compared with the (j/f) performance of flat fins, and a peak performance is obtained with 1.0 < η < 1.2. In low flow rates (Re ≈ 10), on the other hand, a much larger fin waviness severity (γ > 0.5) may be required in order to achieve any significant enhancement.

- The cross-section aspect ratio (α) and fin separation (η) appear to have competing effects on the thermal-hydraulic performance, as measured by the surface area goodness factor (j/f) or core compactness, and the optimum dependent upon the flow regime. Nevertheless, increasing fin density (or decreasing η) tends to promote a relatively better (j/f) performance under swirl flow conditions and thus provide for a more compact wavy plate-fin heat exchanger core.

- At 2A/H < 1, the friction factor is increased somewhat due to the transition from laminar to turbulent flow. While at 2A/H > 1, it is decreased monotonically with increasing Reynolds number throughout the flow regime. Further, the formation of a large-scale vortex is the cause of the transition from laminar to turbulent flow.

The above authors studied about the net effects of dimensionless geometrical parameters on heat transfer and pressure drop. The individual effects of geometrical parameters such as fin height (H), fin amplitude (A), fin wavelength (λ) and fin spacing have been reported in Sparrow and Comb

- It is possible to introduce longitudinal vortices in the wavy duct flows by centrifugal flow instability. The flow instability exists in a small range of Goertler numbers and depends strongly on the duct heights. Unstable duct flow over the whole duct length is observed only at small duct heights. The above mentioned results proved that the pressure losses in the channels with longitudinal vortices are small in comparison with those of turbulent duct flows.

- With Sparrow and Comb (1983) duct configuration, the increase in the inter wall spacing gives rise to a 30% increase in the fully developed Nusselt number relative to that of Reza Motamed et al (1987), however the friction factor is increased more than twice.

- Finally, it is concluded that intensity of heat transfer may be improved with the increase of wave amplitude, or the decrease of wavelength, fin spacing and fin height.

### 2.2.4 Flow and heat transfer correlations

Development of correlations for wavy fins in the form of Nusselt number and fanning friction factor have been studied in Sparrow and Hossfeld (1984), Stasiek (1998) and Yar-Tsai Lin et al (2002). The first two authors gave correlations under single phase flow conditions, whereas the last author gave correlation under two phase flow (humid) conditions. Their correlations are compared in Table 2.3.
Table 2.3 Comparison of available correlations for wavy fins

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Authors</th>
<th>Correlations</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Sparrow and Hossfeld (1984)</td>
<td>Nu = 0.491 * Re^{0.632} * Pr^{0.3}</td>
</tr>
<tr>
<td>2</td>
<td>Stasiek (1998)</td>
<td>Nu ∝ Re^{2/3}</td>
</tr>
<tr>
<td>3</td>
<td>Yar-Tsai Lin et al (2002)</td>
<td>Nu = 0.02656Re_{Dh}^{0.92333} (F_s/D_h)^{0.25906}Re_{Dh}^{-0.15453} (F_s/D_h)^{-0.096529} \theta^{0.47028} \theta^{1.3385} \theta^{0.07773}RH^{0.13035}</td>
</tr>
<tr>
<td></td>
<td>(For two phase flow i.e., under humid condition)</td>
<td>f = 0.02403 Re_{Dh}^{0.41543} (F_s/D_h)^{-0.096529} \theta^{0.47028} \theta^{1.3385} \theta^{0.07773}RH^{0.13035}</td>
</tr>
</tbody>
</table>

Where:
- F_s = Fin spacing (m),
- \theta = Corrugation angle (°),
- RH = Inlet relative humidity (%)

The Nusselt number correlations of first two authors have been compared with the experimental results of Gradeck et al (2005) and found too much variation among the results. This may be due to the difference in geometrical parameters considered in their analysis. Based on the review, it is found that very few correlations are available for wavy fins, when compared to offset fins. However, no correlations have been reported considering the effect of variations in geometrical parameters as considered in offset fin case. So, there is a need to develop the correlations by considering these parameters.

Further, Sparrow and Hossfeld (1984) has also studied the effects of the rounding of the corrugation on the heat transfer enhancement for a wide range of Reynolds number (2000 < Re < 33,000) and for Prandtl number (4 < Pr < 11). They used water as working fluid. For the given flow parameters, they showed that the rounding induced a decrease in Nusselt numbers due to the smoothing of flow. However, the effect of different values
of wavy tip radius on flow and heat transfer is not highlighted anywhere. But, variations in wavy tip radius are having significant effects on flow and heat transfer.

### 2.3 HEAT TRANSFER SURFACE - OFFSET AND STRIP FINS

The research works carried out on various aspects of offset fins as shown in Figure 2.2 are summarized in this section.

![Figure 2.2 Offset and strip-fin (a) Photographic view (b) dimensional notations](image)

**Figure 2.2 Offset and strip-fin (a) Photographic view (b) dimensional notations**

#### 2.3.1 Fin thickness

The effect of larger fin thickness in offset fin heat exchangers has been reported in Kays (1960), Cur and Sparrow (1979) and Patankar and Prakash (1981). Kays (1960) observed 25% increase in pressure drop for the fin thickness of 0.01 inch as compared to the 0.006 inch. However, he has not reported the increase in heat transfer. According to Patankar and Prakash (1981), for the Reynolds number range of 100 - 2000, ‘f’ for the thick plate (t/H = 0.3) was 10 – 16 times the corresponding values for the zero-thickness plate and the heat transfer was found to increase with t/H, but not as much as one would expect from the increased average velocity based on the minimum free flow area and the increased surface area for the thick plates. The Stanton
number values for the case of t/H = 0.3 were only about 24 times the corresponding values for the zero-thickness plate. But, according to Cur and Sparrow (1979), at higher Reynolds number, the increase in fully developed Nusselt number was about 40% over the range from t/L = 0.04 to 0.12. Even larger increase were encountered at lower Reynolds number owing to the triggering transition from laminar to turbulent flow. They have not mentioned the percentage of increase in pressure drop with respect to thickness.

2.3.2 Two and three dimensional computations

Differences between 2D and 3D numerical results have been reported in Suzuki et al (1985) and Guannan and Shah (1999). According to Guannan and Shah (1999), it was found that the effect of 3D geometry was similar in the laminar flow region (Re_{3D} \leq 1600), when compared to the 2D results. In the transition flow region, the use of the 2D computation was inaccurate to predict the flow and heat transfer performance for the OSF surfaces and hence the effect of the 3D geometry was significant. But, Suzuki et al (1985) compared the 2D results with in-house experimental results and found that the 2D computations were in quantitatively good agreement with the performance study for the offset strip fin heat exchanger in the range of Re < 800. The range of Reynolds number quoted by both the authors is different. This may be due to the differences in geometrical dimensions considered in their analysis. Unlike the flow through the parallel rectangular duct, in the case of offset fin, the transition Reynolds number varies with geometrical dimensions of the fin.

2.3.3 Other working fluids

Studies on offset fins using liquid as working fluids have been reported in Tinaut et al (1992) and Sen Hu and Keith Herrold (1995a). Sen Hu and Keith Herrold (1995b) compared the experimental data for liquids against
air (Pr = 0.7) correlations from Webb and Joshi (1987) and Wieting (1975). The results of both models and experiments were obtained for inlet fluid temperature of 10°C. It was observed that at the same Reynolds number, the Colburn factor for liquids predicted using air models was approximately twice the Colburn factor obtained from the experimental data for the liquids. Air model over-predicted the heat transfer coefficient for liquids. This comparison between the air models and the experimental results demonstrated that the results for air cannot be accurately applied to liquid applications. Similarly, for a certain fin geometry and Reynolds number, the friction factor for different Prandtl numbers should be the same. But, the measured friction factor from liquid experiments was found to be higher than that from the air models. This difference might be due to burrs on the fins, created during the fin manufacturing process.

2.3.4 Heat transfer and flow friction Correlations

Development of heat transfer and pressure drop correlations in the form of $j$ and $f$ for the offset strip fins have been reported in Manson (1950), London and Shah (1968), Wieting (1975), Mochizuki and Yagi (1975), Webb and Joshi (1987), Manglik and Bergles (1995), Muzychka and Yovanovich (1999), Maiti (2002) and Ranganayakulu et al (2007). These correlations are compared in Table 2.4.
Table 2.4 Comparison of available $j$ and $f$ correlations for offset fins

<table>
<thead>
<tr>
<th>S. No</th>
<th>Authors</th>
<th>Correlations and Hydraulic diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Wieting (1975)</td>
<td>For $Re_{D} \leq 1000$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$f = 7.661 \left( \frac{l}{D} \right)^{0.384} \alpha^{-0.092} Re_{D}^{-0.712}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j = 0.483 \left( \frac{l}{D} \right)^{0.162} \alpha^{-0.184} Re_{D}^{-0.536}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>For $Re_{D} \geq 2000$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$f = 1.136 \left( \frac{l}{D} \right)^{-0.781} \left( \frac{h}{D} \right)^{0.534} Re_{D}^{-0.198}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j = 0.242 \left( \frac{l}{D} \right)^{-0.322} \left( \frac{h}{D} \right)^{0.089} Re_{D}^{-0.368}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Where $D = 2sl/(s+h)$</td>
</tr>
<tr>
<td>2</td>
<td>Webb and Joshi (1987)</td>
<td>$Re \leq Re^\theta$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j = 0.53(Re_{D})^{-0.50} \left( \frac{l}{D_h} \right)^{-0.15} \alpha^{-0.14}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$f = 8.12(Re_{D})^{-0.74} \left( \frac{l}{D_h} \right)^{-0.41} \alpha^{-0.02}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$Re \geq (Re^\theta + 1000)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$j = 0.21(Re_{D})^{-0.40} \left( \frac{l}{D_h} \right)^{-0.24} \left( \frac{h}{D_h} \right)^{0.02}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$f = 1.12(Re_{D})^{-0.36} \left( \frac{l}{D_h} \right)^{-0.65} \left( \frac{h}{D_h} \right)^{0.17}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Where</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$Re^\theta = 257 \left( \frac{l}{s} \right)^{0.21} \left( \frac{l}{l} \right)^{0.59} D_h \left[ t + 1.328 \left( \frac{Re}{lD_h} \right)^{0.5} \right]^{-1}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$D_h = 2(s-t)h/[s+h+th/l]$</td>
</tr>
<tr>
<td>S. No</td>
<td>Authors</td>
<td>Correlations and Hydraulic diameter</td>
</tr>
<tr>
<td>-------</td>
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<td>---------------------------------------------------------------------------</td>
</tr>
</tbody>
</table>
| 3     | Mochizuki and Yagi (1975)        | $Re < 2000$  
$j = 1.37 \left( \frac{1}{D_h} \right)^{0.25} \alpha^{-0.184} Re^{-0.67}$  
$f = 5.55 \left( \frac{1}{D_h} \right)^{0.32} \alpha^{-0.092} Re^{-0.67}$  
$Re \geq 2000$  
$j = 1.17 \left( \frac{1}{D_h} + 3.75 \right)^{-1} \left( \frac{l}{D_h} \right)^{0.089} Re^{-0.36}$  
$f = 0.83 \left( \frac{1}{D_h} + 0.33 \right)^{-0.5} \left( \frac{l}{D_h} \right)^{0.534} Re^{-0.20}$  
where $D_h = \frac{2sh}{s+h}$. |
| 4     | Manson (1950)                     | $j = \begin{cases} 
0.6 \left( \frac{l}{D_h} \right)^{0.5} Re^{0.6} & \text{if } D_h \leq 3.5 \\
0.321 Re^{0.5} & \text{if } D_h > 3.5
\end{cases}$  
For $Re \leq 3500$  
$f = \begin{cases} 
11.8 \left( \frac{l}{D_h} \right) Re^{0.67} & \text{if } D_h \leq 3.5 \\
3.371 Re^{0.67} & \text{if } D_h > 3.5
\end{cases}$  
For $Re > 3500$  
$f = \begin{cases} 
0.38 \left( \frac{l}{D_h} \right) Re^{0.24} & \text{if } D_h \leq 3.5 \\
0.1086 Re^{0.24} & \text{if } D_h > 3.5
\end{cases}$  
where hydraulic diameter is defined by $D_h = \frac{2sh}{s+h}$. |
Table 2.4 (Continued)

<table>
<thead>
<tr>
<th>S. No</th>
<th>Authors</th>
<th>Correlations and Hydraulic diameter</th>
</tr>
</thead>
</table>
| 5     | Manglik and Bergles (1995) | \( j = 0.6522 \text{Re}^{0.5403} \alpha^{0.1541} \beta^{0.1499} \gamma^{0.0678} \)  
\( f = 9.6243 \text{Re}^{0.7422} \alpha^{0.1856} \beta^{0.3053} \gamma^{0.2659} \)  
\( \left[ 1 + 5.269 \times 10^{-5} \text{Re}^{1.340} \alpha^{0.504} \beta^{0.456} \gamma^{1.055} \right]^{0.1} \)  
where \( D_h = \frac{4shl}{2(s + hl + th) + ts} \), \( \beta = t/s \) and \( \gamma = v/l \) |
| 6     | Maiti (2002) | Re < \text{Re}^*  
\( f = 4.67(\text{Re})^{0.7} (h/s)^{0.196} (l/s)^{0.181} (t / s)^{0.104} \)  
\( j = 0.36(\text{Re})^{0.51} (h/s)^{0.275} (l/s)^{0.27} (t / s)^{0.063} \)  
Re > \text{Re}^*  
\( f = 0.32(\text{Re})^{0.286} (h/s)^{0.221} (l/s)^{0.185} (t / s)^{0.023} \)  
\( j = 0.18(\text{Re})^{0.42} (h/s)^{0.288} (l/s)^{0.184} (t / s)^{0.05} \)  
\( \text{Re}_f^* = 648.3(h/s)^{0.06} (l/s)^{0.1} (t / s)^{0.196} \)  
\( j = 1568.58(h/s)^{0.217} (l/s)^{1.433} (t / s)^{0.217} \)  
\( \text{Hydraulic diameter} = D_h = \frac{2lh(s-t)}{ls + hl + ht} \)  
100 ≤ \text{Re} ≤ 10,000 |
Based on the above survey, it is understood that there are limited experimental data available in the literature for different geometries of offset strip fins. Since, very few experimental data are available, the designers are forced to use available correlations or numerical tool for optimum design of compact heat exchangers. Now-a-days plenty of numerical tools are available in the market and different correlations are available in the literatures. This reduces the experimental set up and testing cost to some extent.

### Table 2.4 (Continued)

<table>
<thead>
<tr>
<th>S. No</th>
<th>Authors</th>
<th>Correlations and Hydraulic diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>7</td>
<td>Muzychka and Yovanovich (1999)</td>
<td></td>
</tr>
</tbody>
</table>
\[
\begin{align*}
    f &= \left[ \frac{f \, \text{Re}_{d_h} \, \frac{d_f}{D_h}}{\text{Re}_{d_h}} + 1.328 \left( \text{Re}_{d_h} \frac{L_s}{d_h} \right)^{1/2} \right] + 0.074 \left( \text{Re}_{d_h} \frac{L_s}{d_h} \right)^{1/2} + \left( \frac{H_t + \frac{s_f}{2}}{2L_s (H + s)} \frac{C_D}{100} \right)^{1/2} \\
    j &= \left( \frac{\text{Nu}_{d_h} \frac{d_f}{D_h}}{\text{Re}_{d_h} \, \text{Pr}^{1/3}} \right)^{1/5} + 0.641 \left( \frac{f \, \text{Re}_{d_h}}{\text{Re}_{d_h}^{2/3}} \right) \left( \frac{d_f^2}{D_h L_s} \right)^{1/3} + 0.037 \left( \frac{\text{Re}_{d_h} \frac{L_s}{d_h}}{d_f^2} \right)^{1/3} \left( \frac{d_f^2}{D_h L_s} \right)^{2/3} \\
\end{align*}
\]

Where $C_D = \text{Form drag} = 0.88$ and $1.3 < n < 5$.  
\[
\begin{align*}
    D_h &= \text{Hydraulic diameter of the sub channel,} = 4A/P. \\
    d_h &= \text{hydraulic diameter of OSF array} = 4V \, \text{free} \, A_wet. \\
    A &= \text{Area,} \, m^2, \, V = \text{Volume,} \, m^3 \\
    2 < m < 5, \text{Correlation is valid for all regimes.}
\end{align*}
\]
extent. Even though plenty of correlations are available in the literature for these type of fins, the designers are not able to select the best correlation, because of too much deviations observed among the correlations. These variations are shown in Figure 2.3 (a) and 2.3 (b). It is evident that remarkable variations are present in the estimation of \( j \) and \( f \) factors of Offset strip surfaces. In the laminar region of \( f \) factors, all correlations are predicting well with experimental data of Kays and London experimental data (1984) except Manson (1950) correlation as given by Manglik and Bergles (1995), which over predicted by twice to thrice. In turbulent region of \( f \) factors, Manglik and Bergles (1995) and Maiti (2002) predicted well. But Mochizuki and Yagi (1975) and Webb and Joshi (1987) under predicted more than 15%. Weiting (1975) over predicted by 30% and Manson (1950) over predicted twice. In \( j \) factor is concerned for both regions, except Weiting (1975) and Mochizuki and Yagi (1975), all others under predicted by more than 20%. Manson (1950) over predicted by almost twice with experimental data of Kays and London (1984). Giving exact reasons for variation of these factors may not be possible due to involvement of so many parameters such as manufacturing aspects and testing conditions.
Figure 2.3 Comparison of literature correlations with Kays and London (1984) experimental results for Fin: 1/8 - 16.00 (D) (a) $f$ vs Re and (b) $j$ vs Re
2.4 CONCLUSIONS FROM LITERATURE SURVEY AND MOTIVATION

Many authors have attempted to develop correlations for $f$ and $j$ as these parameters, have major role in the design of compact plate fin heat exchanger. The importance of fluid flow non-uniformity in the compact plate fin heat exchanger is also understood from the literature. Several authors have contributed significantly to develop the correlation for offset and strip fins. However all their results are contradicting. In the case of wavy fins, only very few correlations are available in the literature and also the effect of geometrical parameters are not considered by the researchers.

After carrying out the literature survey and towards achieving the objectives of present research work, the following tasks have been performed.

Detailed pressure drop analysis has been carried out on four typical compact plate fin heat exchangers using commercial CFD Software-Fluent. This analysis involves preparation of CFD model, meshing, solving and post processing the results. Keeping the fluid flow non-uniformity in the mind, the inlet headers of these three heat exchangers are modified by placing a baffle plate. The above pressure drop analysis is repeated and the results for all the cases are discussed. In addition, the flow non-uniformity is estimated for one of these heat exchangers, and its effects are highlighted. In order to validate the pressure drop analysis, an experiment has been conducted on three of these heat exchangers. The total pressure drop measured in these heat exchangers are compared with CFD data.

Further, this study is extended for two types of fins, such as wavy and offset fins. These fins are widely used in compact plate heat exchangers. In general, offset fins are used in hot air side of the heat exchangers and wavy fins are used in the cold side of the heat exchangers for aerospace applications.
In the case of wavy fin, a parametric study is carried out to understand the effect of fin parameters such as fin height, fin spacing, fin wave length, fin amplitude and wavy tip radius on fanning friction ($f$) and Colburn ($j$) factors. Then generalized correlations have been developed for laminar and turbulent regions, in the form $f$ and $j$ by considering all geometrical and flow parameters. The results of these correlations are compared with the experimental data available in the literature.

In the case of offset fins, it is understood from the literature that, plenty of correlations are available in the form of $f$ and $j$, for both laminar and turbulent regions. Hence the results of all the correlations with fluent results are compared with in-house experimental data for a particular offset fin. Based on this comparison, the deviations among the correlation results are estimated and the uncertainties for the use of these correlations are highlighted. Finally, keeping the uncertainty in the mind, a new design data is generated using Fluent for some of the offset fins used for selected applications.