CHAPTER – 5

Results and Discussion

5.1 Data Reduction

The evaporation heat transfer coefficient and pressure drop for refrigerant R134a flowing in the plate heat exchanger were explored experimentally. The effects of mean vapor quality, fins and pressure on the evaporation heat transfer were investigated. Initially the experiments were carried out using water on both channels for both bubble finned plate heat exchanger and plate heat exchanger without fin arrangements. The heat transfer coefficient for both single phase and two phase flow were well correlated.

A data reduction analysis is needed in the present measurement to deduce the heat transfer rate from the water flow to the refrigerant flow in the test section.

The equivalent diameter \((D_e)\) can be calculated as

\[
D_e = 4 \times \frac{\text{Volume between the plates}}{\text{Wetted surface area between the plates}}
\]

The Reynolds number is also based on the equivalent diameter \((D_e)\). Convection heat transfer coefficient and friction factor were calculated from reducing the measured raw data. The reduction procedures are given in the following.
In the initial single phase water-to-water heat transfer test for the present plate heat exchangers, the fluid properties were calculated at the average of the inlet and outlet bulk fluid temperatures. The total heat transfer rate for counter flow plate heat exchanger from macroscopic energy balance is

\[ m_h C_{p_h} (T_{h_i} - T_{h_o}) = m_c C_{p_c} (T_{c_i} - T_{c_o}) = Q \]  

(5.1)

After accounting for all heat losses, the average of \( Q_h \) and \( Q_c \) were taken as the total heat load.

\[ Q = \frac{Q_h + Q_c}{2} \]  

(5.2)

where

\[ Q_h = m_h C_{p_h} (T_{h_i} - T_{h_o}) \]

and

\[ Q_c = m_c C_{p_c} (T_{c_i} - T_{c_o}) \]
The overall heat transfer coefficient between the two sides can be expressed as

\[ U = \frac{Q}{A \text{ LMTD}} \]  \hspace{1cm} (5.3)

where 'A' is the heat transfer surface area.

The log mean temperature difference (LMTD) is determined from the inlet and exit temperatures of two counter flow channels.

\[ \text{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \]  \hspace{1cm} (5.4)

with

\[ \Delta T_1 = T_{h_1} - T_{c_2} \]

\[ \Delta T_2 = T_{h_2} - T_{c_1} \]

In view of the same heat transfer area in the hot and cold water sides, the relation between the overall heat transfer coefficient and convective heat transfer coefficient on both sides can be expressed as
\[ \frac{1}{U} = \frac{1}{h_c} + \frac{t}{k_{\text{plate}}} + \frac{1}{h_h} \]  

The heat transfer co-efficient \( h_c \) and \( h_h \) were determined from calibration equations, that were devised on the basis of extended experiments employing the modified Wilson plot technique as described in the chapter 3.4.

For the two phase evaporation heat transfer analysis, first the total heat transfer rate between the counter flows in the plate heat exchanger is calculated from the hot water side.

\[ Q_h = m_h C_{p_h} (T_{h_i} - T_{h_o}) \]  

The thermal resistance summation is used to determine the heat transfer co-efficient of refrigerant side \( (h_r) \)

\[ \frac{1}{U} = \frac{1}{h_h} + \frac{t}{k_{\text{plate}}} + \frac{1}{h_r} \]  

The equivalent mass flux is based on the mean vapor quality \( (x_m) \), which is used to determine equivalent Reynolds number \( (Re_{eq}) \) and equivalent Boiling numbers \( (Bo_{eq}) \)
\[ \text{Re}_{eq} = \frac{G_{eq} D_e}{\mu_i} \]  

(5.8)

and

\[ \text{Bo}_{eq} = \frac{q''}{G_{eq} i_{fg}} \]  

(5.9)

where

\[ G_{eq} = G \left[ (1-x_m) + x_m \left( \frac{\rho_s}{\rho_a} \right)^{0.5} \right] \]  

(5.10)

and \( i_{fg} \) - enthalpy of vaporization

To evaluate the friction factor associated with the refrigerant R134a evaporation, the frictional pressure drop \( \Delta P_f \) was calculated as follows

\[ \Delta P_f = \Delta P_{\text{exp}} - \Delta P_a - \Delta P_{\text{man}} - \Delta P_{\text{ele}} \]  

(5.11)

The acceleration and elevation pressure drop were estimated as

\[ \Delta P_a = G^2 u_{fg} \Delta x \]  

(5.12)
\[ \Delta P_{\text{elc}} = \frac{gL}{u_m} \]  \hspace{1cm} (5.13)

where \( u_m = \left[ x_m v_g + (1 - x_m) v_i \right] \)

The pressure drop in the inlet and outlet manifolds and ports was empirically suggested by Shah and Focke (1988). It is approximately 1.5 times the head due to flow expansion at the inlet

\[ \Delta P_{\text{man}} = 1.5 \frac{u_m^2}{2v_m} \]  \hspace{1cm} (5.14)

where \( u_m \) is the mean flow velocity and is also equal to

\[ u_m = Gv_m \]  \hspace{1cm} (5.15)

From all the above, the friction factor can be expressed as

\[ f_p = \frac{\Delta P_{i}De}{2G^2v_mL} \]  \hspace{1cm} (5.16)
5.2 Single phase water to water Heat transfer and Pressure drop Correlations

In the single phase heat transfer analysis, the mass flux was varied from 2.1 to 5.278 kg/m$^2$s. During the single phase heat transfer in the bubble finned plate heat exchanger, 1600 to 3100 W heat energy was transferred. The overall heat transfer coefficient between the two fluids as based on the log mean temperature difference is 400 - 6800 W/m$^2$K. The effectiveness of the bubble finned plate heat exchanger is calculated using the equation

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

$$
\varepsilon = \frac{Q}{C_{\text{min}} (T_{h_i} - T_{c_i})}
$$

(5.17)

is 0.4.

The heat transfer co-efficient for the cold water side ($h_c$) was correlated in terms of Nusselt number using modified Wilson plot technique as given in the figure 5.1. The variables were plotted as below and the constant ($C_e$) as mentioned in chapter 3, equation 3.5 was determined.
By imposing the exponent of the Prandtl number and dynamic viscosity ratio, exponent of Reynolds number was determined. By using the available data, the Nusselt number was well correlated as

$$\text{Nu} = 1.983 \text{Re}^{0.876} \text{Pr}^{0.33} \left( \frac{\mu}{\mu_w} \right)^{0.14}$$

(5.18)

for $800 < \text{Re} > 2000$

The functional form of the variables were fitted in linear form, which is best suited to $R^2 = 0.9972$, and the co-efficient of the correlation is 0.998 as shown in the figure 5.2.
The Nusselt number correlation for single phase water to water heat transfer coefficient is appropriate for the Reynolds number between 800 to 2000. The experimental results are presented in the figure 5.2 and 5.3.
To determine the heat transfer enhancement in the bubble finned plate heat exchanger, the results of single phase water-to-water flow heat transfer in bubble finned plate heat exchanger (BFPHE) were compared to single phase water to water flow heat transfer in the PHE without fin arrangements (PPHE). The heat transfer co-efficient for cold water side in the plain plate of the PHE was also well correlated using the same procedure and modified Wilson plot technique as
\[ \text{Nu} = 2.158 \text{Re}^{0.85} \text{Pr}^{0.333} \left( \frac{\mu}{\mu_w} \right)^{0.14} \]  

(5.19)

for \( 750 < \text{Re} < 2000 \)

The functional form of the variable was fitted in linear form which is best suited to \( R^2 = 0.9979 \). The co-efficient of the correlation is 0.9963. The correlation is appropriate for the Reynolds number 750-2000. The experimental results are presented in the figure 5.4.

Figure 5.4 The experimental results of the single phase flow heat transfer of PPHE
For the same heat flux and mass flux, the heat transfer was 1.2 to 1.5 times more enhanced for the bubble finned plate heat exchanger than for the plate heat exchanger without fin arrangements (PPHE). The mechanism of heat transfer enhancement is analyzed in the numerical studies (chapter 4). The fluid is accelerated and decelerated in the convergent and divergent section formed by the bubbles respectively. The velocity boundary layer is also decreased. Fluid in the concave bubble is not retained for a long period, it is easily rejoined in the main flow. The figures 5.5 and 5.6 show the variation of heat transfer co-efficient with Reynolds number and mass flow rate respectively. Some of the sample values are also tabulated in the Tables 5.1 and 5.2.

![Figure 5.5 Variations of heat transfer coefficient with Reynolds No](image_url)
Figure 5.6 Variations of heat transfer coefficient with mass flow rate
Table 5.1

Comparison of heat transfer coefficients of BFPHE and PPHE with mass flow rate

<table>
<thead>
<tr>
<th>Mass flow rate (m/kg/s)</th>
<th>Heat transfer coefficient W/m²K (bubble finned plate PHE)</th>
<th>Heat transfer coefficient W/m²K (plain plate PHE)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0487</td>
<td>7515.946</td>
<td>7008.283</td>
</tr>
<tr>
<td>0.0615</td>
<td>9218.984</td>
<td>8793.359</td>
</tr>
<tr>
<td>0.07956</td>
<td>11548.63</td>
<td>11311.99</td>
</tr>
<tr>
<td>0.0812</td>
<td>11756.91</td>
<td>11540.7</td>
</tr>
<tr>
<td>0.083245</td>
<td>12015.88</td>
<td>11825.89</td>
</tr>
<tr>
<td>0.08548</td>
<td>12298.02</td>
<td>12137.59</td>
</tr>
<tr>
<td>0.08654</td>
<td>12431.51</td>
<td>12285.41</td>
</tr>
<tr>
<td>0.0877</td>
<td>12729.94</td>
<td>12447.18</td>
</tr>
<tr>
<td>0.0917</td>
<td>13204.98</td>
<td>13005.02</td>
</tr>
<tr>
<td>0.0986</td>
<td>13708.17</td>
<td>13967.29</td>
</tr>
<tr>
<td>0.099</td>
<td>13919.13</td>
<td>14023.07</td>
</tr>
<tr>
<td>0.125</td>
<td>17406.27</td>
<td>16248.125</td>
</tr>
<tr>
<td>0.131</td>
<td>18048.12</td>
<td>17015.125</td>
</tr>
</tbody>
</table>

Table 5.2

Comparison of heat transfer coefficients of BFPHE and PPHE with Reynolds number

<table>
<thead>
<tr>
<th>Reynolds No.</th>
<th>Heat transfer coefficient W/m²K (bubble finned plate PHE)</th>
<th>Heat transfer coefficient W/m²K (plain plate PHE)</th>
</tr>
</thead>
<tbody>
<tr>
<td>775.7938</td>
<td>6782.287</td>
<td>6731.325</td>
</tr>
<tr>
<td>938.2803</td>
<td>8202.807</td>
<td>8020.16</td>
</tr>
<tr>
<td>979.6985</td>
<td>8564.9</td>
<td>8317.843</td>
</tr>
<tr>
<td>1272.812</td>
<td>11127.41</td>
<td>10373.27</td>
</tr>
<tr>
<td>1293.521</td>
<td>11308.45</td>
<td>10515.48</td>
</tr>
<tr>
<td>1326.098</td>
<td>11593.25</td>
<td>10738.48</td>
</tr>
<tr>
<td>1361.701</td>
<td>11904.51</td>
<td>10981.23</td>
</tr>
<tr>
<td>1397.066</td>
<td>12213.69</td>
<td>11221.36</td>
</tr>
<tr>
<td>1413.793</td>
<td>12359.92</td>
<td>11334.61</td>
</tr>
<tr>
<td>1460.786</td>
<td>12770.75</td>
<td>11651.66</td>
</tr>
<tr>
<td>1577.076</td>
<td>13787.4</td>
<td>12429.52</td>
</tr>
<tr>
<td>1991.257</td>
<td>17408.33</td>
<td>14965.258</td>
</tr>
<tr>
<td>2090.023</td>
<td>18271.79</td>
<td>15623.487</td>
</tr>
</tbody>
</table>
It was also found that, by providing the bubble fins in the plate heat exchanger, the rigidity of the exchanger has increased, so that it is enabled to withstand high pressure. Turbulent flow for low Reynolds number even 700 was also induced so as to enhance the heat transfer. For the same mass flow rate, the bubble finned plate exhibits high Reynolds number; as shown in the figure 5.7 and the values are also given in the table 5.3.

![Figure 5.7 Variations of Reynolds No with mass flow rate](image)
### Table 5.3

**Comparison of Reynolds numbers of BFPHE and PPHE with mass flow rate**

<table>
<thead>
<tr>
<th>Mass flow rate (m) kg/s</th>
<th>Reynolds No (bubble finned plate PHE)</th>
<th>Reynolds No (plain plate PHE)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.062827</td>
<td>1026.78</td>
<td>775.7938</td>
</tr>
<tr>
<td>0.07165</td>
<td>1170.962</td>
<td>979.6985</td>
</tr>
<tr>
<td>0.086561</td>
<td>1414.654</td>
<td>1267.395</td>
</tr>
<tr>
<td>0.0917</td>
<td>1460.786</td>
<td>1293.521</td>
</tr>
<tr>
<td>0.095823</td>
<td>1566.022</td>
<td>1326.974</td>
</tr>
<tr>
<td>0.0986</td>
<td>1570.704</td>
<td>1360.427</td>
</tr>
<tr>
<td>0.09965</td>
<td>1628.562</td>
<td>1378.587</td>
</tr>
<tr>
<td>0.10025</td>
<td>1596.988</td>
<td>1397.066</td>
</tr>
<tr>
<td>0.108411</td>
<td>1771.75</td>
<td>1570.704</td>
</tr>
<tr>
<td>0.113813</td>
<td>1860.025</td>
<td>1596.988</td>
</tr>
<tr>
<td>0.12078</td>
<td>1973.895</td>
<td>1991.257</td>
</tr>
<tr>
<td>0.124917</td>
<td>2041.494</td>
<td>2090.023</td>
</tr>
</tbody>
</table>

The correlation obtained for single phase water-to-water flow heat transfer in the bubble finned plate heat exchanger was compared to Li (1999) correlation for dimple plate as shown in figure 5.8.

Figure 5.8 Dimple plate of Li et al.
The well known Li (1999) correlation is

\[ \text{Nu} = 1.726 \text{Re}^{0.414} \text{Pr}^{0.4} \]  

(5.20)

for \( \text{Re} \leq 2000 \)

The results show that the present data for bubble finned plate heat exchanger generally offer higher heat transfer co-efficient, which are about 10 to 28 percent higher than Li (1999) correlation. At higher mass flux, it exhibits more heat transfer than dimple plate. The variations of heat transfer co-efficient with mass flow rate for the present data and Li correlation are shown in figure 5.9.

![Figure 5.9 variations of heat transfer co-efficient with mass flow rate](image)

Figure 5.9 variations of heat transfer co-efficient with mass flow rate
The Nusselt number Correlations predicted earlier for the conventional heat exchangers were also analyzed and compared.

The correlations from literature for single tubes are:

Rohsenow and Choi (1961)

$$Nu = 0.0223 \, Re^{0.8} \, Pr^{0.4}$$  \hspace{1cm} (5.21)

Rohsenow and Hartnett (1973)

$$Nu = 0.022 \, Re^{0.8} \, Pr^{0.6}$$  \hspace{1cm} (5.22)

Kays and Leung (1962)

$$Nu = 0.0422 \, Re^{0.74} \, Pr^{0.56}$$  \hspace{1cm} (5.23)

Incropera and Dewitt (1981)

$$Nu = 0.027 \, Re^{0.8} \, Pr^{0.14} \left( \frac{\mu}{\mu_w} \right)^{0.14}$$  \hspace{1cm} (5.24)

and it was found that the heat transfer co-efficient in the bubble finned plate is much higher than shell and tube heat exchanger. The variations of heat transfer co-efficient with various parameters for both bubble finned plate of the plate heat exchanger and plain plate are represented graphically and the experimental values are also given.
Figure 5.10 Variations of Nusselt Number with mass flow rate

The variations of heat transfer co-efficient with Reynolds number, mass flow rate and Nusselt number in the bubble finned plate heat exchanger is illustrated in the figures 5.11 to 5.13. They show the increase of Reynolds number and then the turbulence increases the heat transfer and the heat transfer enhancement is more in the high mass flux region. The variations of Reynolds number and Nusselt number with mass flux are also given in figure 5.14.
Figure 5.11 Variations of heat transfer co-efficient with Re of Single phase water flow in bubble finned PHE.
Figure 5.12 Variations of heat transfer coefficient with mass flow rate of single phase water flow in bubble finned PHE

Figure 5.13 Variations of heat transfer coefficient with Nu of Single phase water flow in bubble finned PHE
A reasonable objective in compact heat exchanger design is the production of the smallest unit, which will satisfy the required duty within the specified heat transfer constraints. Smaller exchanger volumes are obtained by using surfaces that exhibit high performance. A simple concept that is to be used for surface selection is one referred to as volume performance index (Picon Nunez et al 1999). Here, the surfaces are compared on the basis of VPI and envelopes for best performance are produced.

The energy balance for the heat transfer process is

\[ m \, C_p \, \Delta T = h \, A \, \text{LMTD} \]  

(5.25)
The Stanton number

\[ St = \frac{h}{mCp} \text{ } A_f \] (5.26)

where \( A_f \) - flow area

we know that the equivalent diameter

\[ De = 4 \frac{A_f L}{A} = \frac{4V}{A} \] (5.27)

where

\[ V = \frac{De \Delta T A_f}{4 \text{ St LMTD}} \] (5.28)

and

\[ A = \left[ \frac{2m^2V}{\delta De \Delta P} \right] \] (5.29)

where \( \delta \) = plate spacing

Substituting \( A \) in \( V \)

\[ V = \left[ \frac{\Delta t}{\text{LMTD}} \right] \left[ \frac{2m^2}{\delta \Delta P} \right] \left[ De \left( \frac{f}{St^3} \right) \right] \] (5.30)

The first two of the three terms on the right hand side of the above equation are associated with the exchanger duty, while the last term is associated with the performance of a specific surface. The inverse of this term is called volume performance index (VPI).
\[
VPI = \frac{\left(\frac{St^3}{f}\right)^\frac{1}{2}}{De}
\] (5.31)

The sizing of the plate heat exchangers require, the specification of the surface type on each of the streams that will take part in the heat transfer process. A reasonable design objective is to achieve the smallest exchanger volume. This can be achieved by ensuring that the allowable stream pressure drop is fully utilized and that the heat transfer surface selected is highly thermally efficient, that is, it will produce high heat transfer coefficients at low Reynolds number. Here, the VPI of the bubble finned plate and plain plate are compared to Reynolds number. The result shows that the bubble finned plate has higher performance index, which is illustrated in the figure 5.15.

![Figure 5.15 Variations of VPI with Reynolds number](image-url)
The pressure drop between the inlet and outlet of the heat exchanger is considered here as the function of pitch ($p_h$) and height ($h_b$) of the bubble, and the Reynolds number ($Re$)

For a particular pitch-height ratio, the pressure drop is the function of Reynolds number

$$\Delta P = f(Re)$$

(5.32)

Using the principle of least squares and assuming a variation for $\Delta P$, the normal equations are

$$\Sigma Y = a \Sigma A + n \Sigma X$$

(5.33)

$$\Sigma XY = A \Sigma X + n \Sigma X^2$$

(5.34)

where $X = \log Re$; $Y = \log \Delta P$; and $A = \log C$.

The pressure drop can be correlated as

$$\Delta P = 0.347 \, Re^{0.055}$$

(5.35)
In the analysis of single phase water to water flow heat transfer in the plate heat exchanger, it was found that the corrugation provided in the plates of PHE induces turbulent flow at low Reynolds number. The heat transfer rate is increased in the Chevron plates (Yan and Lin 1999), with the augmentation in heat transfer increasing with the increase of Chevron angle with flow direction. But at the same time, the pressure drop has also increased. By introducing the bubble shaped fins in the plate, which has similar enhancement in heat transfer to chevron plates, lesser penalty to pressure drop has been achieved.
5.3 Two Phase Evaporating Heat Transfer and Pressure Drop Correlations

The effects of the mass flux, heat flux and system pressure on the evaporation heat transfer of refrigerant R134a in the bubble finned plate heat exchanger were examined as follows. Selected measured data for the average pressure 0.17 MPa and an average imposed heat flux 150 kW/m² are presented in figure 5.16 to illustrate the changes of the heat transfer co-efficient with mass flux.

Here also, the heat transfer co-efficient in the refrigerant side increases with the mass flux.

![Graph showing variations of heat transfer coefficient with mass flux](image)

*Figure 5.16 Variations of heat transfer Co-efficient with mass flux of refrigerant*
Table 5.4

Values of major parameters for various cases

<table>
<thead>
<tr>
<th>Sl No</th>
<th>G in kg/m^2s</th>
<th>q'' in W/m^2</th>
<th>h_r in W/m^2K</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>24.29</td>
<td>85073.39</td>
<td>2153.075</td>
</tr>
<tr>
<td>2</td>
<td>34.78</td>
<td>131439.234</td>
<td>3425.544</td>
</tr>
<tr>
<td>3</td>
<td>36.38</td>
<td>131867.9</td>
<td>3437.669</td>
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<tr>
<td>4</td>
<td>41.25</td>
<td>143481.11</td>
<td>3768.52</td>
</tr>
<tr>
<td>5</td>
<td>51.235</td>
<td>183482.64</td>
<td>4947.207</td>
</tr>
<tr>
<td>6</td>
<td>56.214</td>
<td>181256.848</td>
<td>4879.979</td>
</tr>
<tr>
<td>7</td>
<td>65.341</td>
<td>224368.3</td>
<td>5364.23</td>
</tr>
<tr>
<td>8</td>
<td>72.365</td>
<td>239546.2</td>
<td>6235.12</td>
</tr>
<tr>
<td>9</td>
<td>79.56</td>
<td>241536.2</td>
<td>6289.25</td>
</tr>
</tbody>
</table>

The mass flux and heat flux were varied from 10 to 80 kg/m^2s and 40 to 250 kW/m^2 respectively. Figure 5.17 shows the effects of the heat flux on the heat transfer co-efficient for 0.17 MPa and 60 kg/m^2s. It is noted that the heat flux exhibits a significant influence on the heat transfer in the refrigerant flow.

Data for three different mass fluxes of 10 kg/m^2s, 30 kg/m^2s and 60 kg/m^2s are compared at the same heat flux and system pressure. It is noted that at a low vapor quality, the mass flux exhibits an insignificant influence on the heat transfer. As the vapor quality is high, the difference in the heat transfer co-efficient for the three mass fluxes begins to grow. There values are given in Table 5.5.
Figure 5.17 Variations of heat transfer coefficient with heat flux of R134a.
Table 5.5

Heat transfer coefficients for different mass fluxes of R134a

<table>
<thead>
<tr>
<th>Sl No</th>
<th>$x_m$</th>
<th>for $G = 10$ kg/m$^2$s $'h_r'$</th>
<th>for $G = 30$ kg/m$^2$s $'h_r'$</th>
<th>for $G = 60$ kg/m$^2$s $'h_r'$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.223</td>
<td>2468.29</td>
<td>2492.38</td>
<td>2563.82</td>
</tr>
<tr>
<td>2</td>
<td>0.362</td>
<td>3312.463</td>
<td>3313.396</td>
<td>3326.843</td>
</tr>
<tr>
<td>3</td>
<td>0.524</td>
<td>3584.724</td>
<td>3641.28</td>
<td>3643.242</td>
</tr>
<tr>
<td>4</td>
<td>0.653</td>
<td>3596.256</td>
<td>3702.56</td>
<td>3893.268</td>
</tr>
<tr>
<td>6</td>
<td>0.766</td>
<td>3843.521</td>
<td>4256.83</td>
<td>5439.652</td>
</tr>
<tr>
<td>7</td>
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<td>4302.31</td>
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</tr>
<tr>
<td>8</td>
<td>0.793</td>
<td>4232.51</td>
<td>4623.79</td>
<td>6253.49</td>
</tr>
</tbody>
</table>

The heat transfer co-efficient for the higher mass fluxes rise more quickly than that for the lower mass fluxes. It is attributed to the fact that, at 0.17 MPa the liquid density of R134a is about 154 times higher than that of corresponding vapor density. Thus a great increase in the vapor volume during the evaporation process causes the vapor flow to move in a high speed, which in turn breaks the adjacent liquid film into a large number of tiny liquid droplets in the channel. This highly turbulent mist flow results in a substantial rise in the heat transfer co-efficient. The variations of the frictional pressure drop in the plate heat exchanger increase with vapor quality. It is further noted that higher mass flux results in high pressure drop for the entire quality range.
The effects of the system pressure on the evaporation heat transfer with three cases of 0.08 MPa, 0.13 MPa and 0.17 MPa were examined. In the low quality regime, the pressure effects are small. As the height of the bubble fins are small, it was further noted that the pressure drop is smaller than that of plate heat exchanger of corrugated plate.

Correlation equations for the heat transfer co-efficient and friction factor associated with R34a evaporation in the plate heat exchanger considered here are important in the practical thermal design of the evaporator for various air conditioning and refrigeration systems. Based on the present data the heat transfer co-efficient can be correlated in the form of

\[
Nu_e = 5.751 \Pr^{0.33} \Boeq^{0.644} \Re^{0.876} [(1-x_m) + x_m (\rho / \rho_g)^{0.5}] \\
\text{for } 350 < \Re < 1450
\]

(5.36)

where, \( \Boeq \) is the equivalent boiling number in which an equivalent mass flux is used in its definition, and it was first proposed by Akers et al (1958).

The functional form of the variables was fitted in linear form, which is best suited to \( R^2 = 0.9851 \). The co-efficient of the correlation is 0.9925. The above correlation can satisfactorily correlate the present data with an average deviation of \( \pm 11.52 \) percent. The results are illustrated in the figure 5.18.
Figure 5.18 The experimental results of the two phase flow heat transfer

Figure 5.19 Comparison of the present heat transfer data for the bubble finned plate heat exchanger with those for chevron plates from Wang et al (2003), Muley et al (1999) and Yan and Lin (1999).
The correlation obtained is compared with Wang and Sunden (2003) correlation, Muley and Manglik (1999), and Yan and Lin (1999) correlation for single phase water-to-water flow heat transfer in the Chevron plate. The results show that the bubble finned plate give heat transfer augmentation. It is illustrated in figure 5.19.

The frictional pressure drop ($\Delta P_f$) associated with the refrigerant R134a evaporation was calculated with considering experimental pressure drop, pressure drop due to elevation, acceleration and manifolds. Based on the present data, the friction factor can be correlated as

$$f_t \cdot Re^{0.876} = 1.332 \times 10^4 Re_{eq}^{-0.6427} \quad (5.37)$$

The functional form of the variable was fitted in linear form which is best suited to $R^2 = 0.882$. The results of the experiments are illustrated in figure 5.20.

![Figure 5.20 Experimental results of the two phase flow pressure drop.](image)
The frictional pressure drop of bubble finned plate heat exchanger was compared with the results of Yan and Lin (1999) and Han et al. (2003), and illustrated in the figure 5.21.

![Figure 5.21 Comparison of present data for friction factor with Yan and Lin and Han et al.](image)

It is of interest to compare the evaporation heat transfer co-efficient for the bubble finned plate heat exchanger and for the circular pipes. Data from T.Y. Choi et al (2000) and Eckels and Pate (1991) were compared with present data and illustrated in figure 5.22. Though the experimental conditions including the tube size, mass flow rate, heat flux level and pressure for these data are somewhat different, the comparison does clearly show that, the heat transfer co-efficient for the bubble finned plate heat exchanger is much higher than that for the circular pipes.
The data of the evaporation heat transfer in the super heated regions are illustrated in figures 5.23 and 5.24. In super heated vapor region, the bubble finned plate also exhibits higher heat transfer rate. The heat transfer co-efficient is 2.4 times higher than saturated flow for the particular mass flux.
Figure 5.23 Variations of heat transfer co-efficient with mass flow rate of R134a.

Figure 5.24 Variations of heat transfer co-efficient with mean vapor quality.
5.4 Saturated and Super heated flow Boiling Curves

The effects of the boiling heat flux, refrigerant mass flux, system pressure and inlet condition for super heated flow of refrigerant R134a in the bubble finned plate heat exchanger are explored in detail. The results are presented in terms of the boiling curves and heat transfer co-efficient.

The variations of the heat transfer co-efficient with equivalent mass flux and Reynolds number for the average pressure 0.17 MPa are illustrated in figure 5.25 and 5.26. It is clearly noted that the heat transfer co-efficient increases with the Reynolds number and equivalent mass flux. It is found that, the Nusselt number for the evaporative fluid flow increases with equivalent Reynolds numbers, which is also shown in figure 5.27.

![Figure 5.25 Variations of heat transfer co-efficient with equivalent mass flux of R134a.](image)
Figure 5.26 Variations of heat transfer co-efficient with Reynolds number of R134a.

Figure 5.27 Variations of Nusselt number with equivalent Reynolds number of R134a.
Figure 5.28 shows the effect of the refrigerant mass flux on the measured R134a evaporation heat transfer co-efficient at an average pressure of 0.17 MPa and an average imposed heat flux of 150 kW/m², for the mass flux 10, 30, 60 kg/m²s and the mean vapor quality varying from 0.2 to 0.8.

![Figure 5.28 Variations of heat transfer co-efficient with mean vapor quality.](image)

These data indicate that, at a given mass flux the evaporation heat transfer increases with mean vapor quality of R134a in the plate heat exchanger. For instance at 60 kg/m²s the evaporative heat transfer co-efficient at the quality of 0.8 is about 2.39 times larger than at 0.18. This obviously results from the simple fact that, at a higher mean vapor quality, the liquid film on the surface is thinner and the evaporation rate is thus higher.
At mass flux 70 kg/m²s and 80 kg/m²s, the heat transfer co-efficient values differ only slightly from mass flux 60 kg/m²s.

Figure 5.29 shows the refrigerant R134a evaporation heat transfer co-efficient at three different heat fluxes \(q'' = 100 \text{ kW/m}^2, 150 \text{ kW/m}^2\) and 200 kW/m²) at 0.17 MPa and 60 kg/m²s. It is noted that the quality averaged evaporation heat transfer co-efficients at 150 and 200 kW/m² are respectively about 6% and 10% larger than that at 100 kW/m². Compared with the mass flux effects, the heat flux has a smaller effect on the evaporative heat transfer co-efficient in the high vapor quality region.

Figure 5.29 Variation of heat transfer co-efficients with mean vapor quality for three different heat fluxes.
Figure 5.30 presents the data for the evaporation heat transfer co-efficient at the heat flux of 15 kW/m² and mass flux of 60 kg/m²s, for three system pressure of 0.08, 0.13 and 0.17 MPa. The results indicate that an increase in system pressure leads to a slight reduction in the evaporation heat transfer.

Figure 5.30 Variations of heat transfer co-efficient with mean vapor quality for three different inlet pressures.
CHAPTER - 6

Conclusion

6.1 Conclusion from the Research Work

Heat transfer characteristics and pressure drop in a bubble finned plate heat exchanger have been explored experimentally. Experiments on the evaporation heat transfer and pressure drop in the bubble finned plate heat exchangers also have been performed with refrigerant R134a. Experimental results and numerical simulations for the performance of fluid flow and heat transfer in such compact heat exchanger are reported. These results have been analyzed and also compared with those of plate heat exchanger without fin arrangements.

- The heat transfer rate is enhanced to 1.2 to 1.5 times in the bubble finned plate of a plate heat exchanger over the plate heat exchanger without fin arrangements, for the single phase convective heat transfer in water flow.

- The Volume Performance Index in the bubble finned plate heat exchanger is higher, which means it can transfer high heat transfer at low Reynolds number.

- The evaporation heat transfer coefficient for R134a flow in the bubble finned plate heat exchanger is quite higher, particularly in the regime of high vapor quality, and the mass flux also exhibits significant effects on the heat transfer
coefficient. The variations of the frictional pressure drop in the plate heat exchanger increase with vapor quality. It is further noted that, higher mass flux results in high pressure drop for the entire quality range.

- The Nusselt number is well correlated for single and also two phase flows. The correlations are appropriate for the Reynolds number between 750 to 2000. The correlation obtained for single phase water-to-water flow in bubble finned plate heat exchanger was compared to Li (1999) correlation for dimple plate. The results show that the present data for bubble finned plate heat exchanger offer higher heat transfer coefficient.

- CFD simulation has been carried out to predict the flow in bubble finned plate heat exchanger. It is found that the convergent – divergent channels formed between bubble fins forced a fluid to become strong three dimensional flow to augment heat transfer. As the fluid flows over and around the bubbles, the velocity boundary layers are decreased evidently and restarted, thus they enhance heat transfer.
6.2 Limitation of this Research Work

- The studies on evaporation heat transfer in bubble finned plate heat exchanger were done with R134a only.
- The experiments were carried out with constant inlet quality of the refrigerant. The heat transfer co-efficient for different dryness fractions is not reported.
- The overall heat transfer co-efficient of the bubble finned plate is only meant for copper plate.
- The heat transfer co-efficient for the different pitches, heights and numbers of the bubble fins are not investigated.
6.3 Scope for the Future Work

In the present work, a new plate heat exchanger, which gives an augmentation in the heat transfer coefficient with a lesser penalty to pressure drop, has been taken up. A design equation for predicting the average heat transfer coefficient with single phase and with vaporization for the bubble finned plate heat exchanger is formulated. The effect of mean vapor quality, fins and pressure on the evaporation heat transfer are investigated.

This study consists the evaporation heat transfer in the bubble finned plate heat exchanger with R134a only. According to Han et al (2003), the constants in the equation for heat transfer coefficients of R410A in a brazed plate heat exchanger is different from that of R22. So the experiment can be carried out, in the bubble finned plate heat exchanger with different proposed future eco friendly refrigerants like R290, R404A, R407C, R410A, R507A, R600 etc for evaporation heat transfer.

In this study, the correlation gives the average heat transfer coefficient of the evaporation heat transfer as a function of mean vapor quality in the heat exchanger. The heat transfer coefficient for a particular dryness fraction is not studied. The study in this regard to find the heat transfer coefficient as a function of dryness fraction is needed.

Hsieh and Weng (1997) suggested that, the pitted coating surfaces give better performance in R134a and plasma spraying surface performs well in R407C. The studies on enhancement of heat transfer with coated surface on bubble finned plate need to be made.
Further experiments are needed to study the effect of size of the fins, pitch of the fins, number of passes etc. in order to find out the effect of these parameters on the performance of the Plate Heat Exchanger.