Plate heat exchangers (PHE) are categorized as compact heat exchangers. In PHEs, heat transfer between two fluids take place through series of metal plates. In plate heat exchangers, series of large rectangular thin metal plates clamped together to form narrow parallel plate channels (passages). Hot and cold fluid flows through alternate channel in PHE and carryout the heat transfer. Fig. 6.1 shows the schematic of PHE with fluid flow path. In PHEs, a narrow channel is formed between every two consecutive plates with the help of gasket. So, the fluid will flow in these channels along the plane of the plates. Gaskets are placed and designed in such a way that it allows the same fluid to flow in alternate channels. Also, both the fluids are separated by thin metal plates which provide turbulence to the fluid flow. As shown in Fig. 6.1, both the fluids enter and exit from first plate (i.e. cover plate) and directed to next plates. Here, gasket allowed hot fluid to flow through first channel (formed between cover plate and first thin metal plate) while cold fluid moved to second channel (formed between first and second metal plate). In this way, same fluid flow in alternate channels and hence heat transfer takes place between them through thin metal plates.

Fig. 6.1 Schematic of Plate heat exchanger with fluid flow path

PHEs are widely used in Petroleum, chemical processing, food & beverages, cryogenics, and pharmaceutical industries. The distinguishing features of PHEs are its high surface area density and thermal effectiveness, resulting in reduced size, weight, and
space compared to other types of heat exchanger (Kays and London, 1984). On the other end, high hydraulic losses (i.e. pressure drop) involved in PHEs. Thus, the trade-off between thermal and hydraulic behaviour is always required to reach at optimum design of PHEs. Further, large number of design parameters is involved in the design of PHEs that should satisfy the geometric/operating constraints and heat duty requirements. As a result, metaheuristic algorithms are more suitable to obtain the optimized design of PHEs as compared to conventional optimization methods. Generally, objectives involved in the design optimization of PHE are thermodynamics (i.e. maximum effectiveness, minimum entropy generation rate, minimum pressure drop, etc.) and economics (i.e. minimum cost, minimum weight, etc.).


Najafi and Najafi (2010) performed a multi-objective optimization of PHE with pressure drop and heat transfer coefficient of a heat exchanger as objective functions. The authors used NSGA-II as an optimization tool. Arsenyeva et al. (2011) proposed mathematical model based area optimization of a multi-pass plate-and-frame heat exchanger. Hajabdollahi et al. (2013) obtained optimized geometric parameters of gasket plate heat exchanger for maximum effectiveness and minimum total cost by adapting NSGA-II. Lee and Lee (2015) carried out a thermodynamic optimization of PHE. The authors considered two conflicting objectives namely, Colburn factor and friction factor for optimization and used GA as an optimization tool. Further, authors also developed the correlation for Colburn factor and friction factor. Hajabdollahi et al. (2016) presented the comparative study of gasket plate and shell and tube heat exchangers from the economic point of view by using a GA.
Main contributions of this chapter are (i) To develop multi-objective thermal-hydraulic optimization problem of plate heat exchanger to maximize overall heat transfer coefficient and minimize total pressure drop. (ii) To employed multi-objective variant of the heat transfer search (MOHTS) algorithm to solve the thermal-hydraulic optimization problem of PHE. (iii) Select a final optimal solution from the Pareto optimal set with the help of LINMAP decision-making approach. (iv) Identify the underlying relationship of decision variables during thermal-hydraulic optimization and (v) Investigate sensitivity of design variable on the optimized value of thermal-hydraulic objective functions (vi) Validate the optimization results by experimental investigation.

6.1 Thermal modeling of plate heat exchanger

In the present work a chevron plate heat exchanger is investigated for the optimization. Geometry of PHE is shown in Fig. 6.1 while detail of chevron plates and corrugation dimensions is shown in Fig. 6.2. In this work, $\varepsilon$-NTU approach is utilized to predict the performance of PHE (Shah and Sekulic 2003). The PHE is assumed to running under a steady state, with negligible heat loss and uniform velocities. Further, heat transfer coefficients are assumed to be uniform and constant.

![Fig. 6.2 Schematic of chevron plate and corrugation dimensions](image-url)
### 6.1.1 Heat transfer

The effectiveness of counter flow heat exchanger is obtained using the following correlation,

\[
\varepsilon = \frac{1 - e^{-NTU(1-C^*)}}{1 - C^* e^{-NTU(1-C^*)}}
\]  

(6.1)

Where, \( C^* \) represents the heat capacity rate ratio and \( NTU \) represents the number of transfer unit of the exchanger and calculated as

\[
C^* = \frac{(mC_p)_{\text{min}}}{(mC_p)_{\text{max}}}
\]  

(6.2)

\[
NTU = \frac{UA_e}{(mC_p)_{\text{min}}}
\]  

(6.3)

Where, \( m \) is the mass flow rate of fluid; \( C_p \) is the specific heat of fluid; \( A_e \) is the effective heat transfer area; \( U \) is the overall heat transfer coefficient of the exchanger and given by,

\[
U = \frac{1}{\left(\frac{1}{h_h} + R_{f,h} + \left(\frac{1}{h_c}\right) + R_{f,c} + \left(\frac{t}{k}\right)_w\right)}
\]  

(6.4)

Where, subscript \( h \) and \( c \) stand for hot side and cold side respectively while \( w \) stand for wall condition; \( R_f \) is the fouling resistance; \( k \) is the thermal conductivity of plate material; \( t \) is the plate thickness; and \( h \) is the heat transfer coefficient and obtained using the following equation,

\[
h = n(Re)^{n_1}(Pr)^{\frac{1}{3}}\left(\frac{H_b}{\mu_w}\right)^{0.17}\left(\frac{k}{d_h}\right)
\]  

(6.5)

Where, the value of coefficient \( n \) and \( n_1 \) depends in the flow characteristic and chevron angle. The value of \( n \) and \( n_1 \) is given in references (Shah and Sekulic 2003) for different chevron angle. Also, \( \mu \) is the fluid viscosity; \( Pr \) is the Prandtl number; \( Re \) is the Reynolds number; \( d_h \) is the hydraulic diameter.

In Plate heat exchanger, the length of the compact plate is given by,

\[
L_p = L_V - D_p
\]  

(6.6)

Where, \( D_p \) is the port diameter; \( L_v \) is the vertical distance between ports and obtained using the below equation,

\[
L_V = L_H + D_p
\]  

(6.7)

Where, \( L_H \) is the horizontal distance between ports.

The mean flow channel gap \( (b) \) of the PHEs can be determine as below,

\[
b = p - t
\]  

(6.8)
Where, \( t \) is the plate thickness; and \( p \) is the plate pitch and obtained using the following equation,
\[
P = \frac{L_c}{N}
\]
(6.9)
Where, \( L_c \) is the vertical distance between ports; \( N \) is the number of plates.

The enlargement factor of PHE is obtained using the below equation,
\[
\varphi = \frac{A_1}{L_p L_W}
\]
(6.10)
Where, \( A_1 \) is the heat area; \( A_{lp} \) is the projected area of the plate and obtained using the below equation,
\[
A_{1p} = L_p L_W
\]
(6.11)
Where, \( L_w \) is the plate width.

Effective heat transfer area \( (A_e) \) of the PHEs can be express as,
\[
A_e = A_1 N_e
\]
(6.12)
Where, \( N_e \) is the effective number of plates in PHE.

The number of channels per pass \( (N_{cp}) \) of the plate heat exchanger is obtained using the below equation,
\[
N_{cp} = \frac{N - 1}{2N_p}
\]
(6.13)
Where, \( N_p \) is the number of pass.

The Reynolds number of the fluid flow is obtained using the following equation,
\[
Re = \frac{G d_h}{\mu}
\]
(6.14)
Where, \( G \) and \( d_h \) is the mass velocity in channel and hydraulic diameter respectively and obtained using the following equations,
\[
G = \frac{m}{N_{cp} b L_W}
\]
(6.15)
\[
d_h = \frac{4b L_W}{2(b + L_W \phi)} \approx \frac{2b}{\phi}
\]
(6.16)

6.1.2 Pressure drop

The pressure drop through the channels can be given as
\[
\Delta P_{channel} = \frac{4f \varphi N_p G^2}{2d_h \rho}
\]
(6.17)
Where, $\rho$ is the density of fluid; and $f$ is the friction factor and obtained using the following equation,

$$ f = \frac{n_2}{Re^{n_3}} \tag{6.18} $$

Where, the value of coefficient $n_2$ and $n_3$ depends in the flow characteristic and chevron angle. The value of $n_2$ and $n_3$ is given references (Shah and Sekulic 2003) for different chevron angle.

The pressure drop in inlet and outlet ports may be given as,

$$ \Delta P_{port} = 1.4N_p \frac{G_p^2}{2\rho} \tag{6.19} $$

Where, $G_p$ is the mass velocity in the port and obtained using the following correlation,

$$ G_p = \frac{4m}{\pi D_p^2} \tag{6.20} $$

The total pressure drop is the summation of the channel pressure drop and the port pressure drop and given by,

$$ \Delta P_{total} = \Delta P_{channel} + \Delta P_{port} \tag{6.21} $$

Based on the above thermal model, objective functions for the present work are defined in the next section.

### 6.2 Objective function, design variables and constraints

In this work, a multi-objective optimization is carried out between conflicting objectives. Maximization of overall heat transfer coefficient and minimization of the total pressure drop of PHE are considered as objectives. For counter flow PHE, overall heat transfer coefficient is calculated using following equation,

$$ U = \frac{1}{\left(\frac{1}{h_b} + R_{f,h} + \frac{1}{h_c} + R_{f,c} + \frac{t}{k}\right)} \tag{6.22} $$

Where, $t$ and $k$ are wall thickness and wall thermal conductivity respectively.

Similarly, the total pressure drop of PHE is summation of the channel pressure drop and the port pressure drop which are given by,

$$ \Delta P_{total} = \Delta P_{channel} + \Delta P_{port} \tag{6.23} $$

$$ \Delta P_{total} = \left(\frac{4fL_c N_p G^2}{2D_h \rho}\right) + \left(1.4N_p \frac{G_p^2}{2\rho}\right) \tag{6.24} $$

Where, $f$ is friction factor.
In this work, multi-objective heat transfer search algorithm is used for the multi-objective optimization of a PHE. The multi-objective problem can be described as follows,

Maximise/Minimise \( f(X) = f_1(X), f_2(X) \)

\[ X = [x_1, x_2, \ldots, x_k] \]  

\text{(6.25)}

Where, \( f_1(X) \) and \( f_2(X) \) are overall heat transfer coefficient and total pressure drop of PHE respectively. Also, the constraints are stated as,

\[ g_i(X) \leq 0, \quad i = 1, 2, \ldots, nc \]  

\text{(6.26)}

\[ x_{j,\min} \leq x_j \leq x_{j,\max}, \quad j = 1, 2, \ldots, nd \]  

\text{(6.27)}

Where, \( nc \) and \( nd \) are numbers of constraints and decision variables respectively.

6.2.1 Design variables and constraints

In this work, eight design variables which affect the performance of PHE are considered for optimization. These variables include: (i) horizontal distance of ports (ii) vertical distance of ports (iii) total length of compact plates (iv) port diameter (v) plate thickness (vi) chevron angle (vii) enlargement factor (vii) number of plates. The design parameters variation ranges are shown in Table 6.1. Moreover, the objective functions should satisfy the following constraints.

\[ \Delta P_{\text{cold}} \leq \Delta P_{\text{cold},\max} \]  

\text{(6.28)}

\[ \Delta P_{\text{hot}} \leq \Delta P_{\text{hot},\max} \]  

\text{(6.29)}

Table 6.1 Ranges of design variables

<table>
<thead>
<tr>
<th>Design variables</th>
<th>Lower bound</th>
<th>Upper bound</th>
</tr>
</thead>
<tbody>
<tr>
<td>Port Diameter, ( D_p ) (mm)</td>
<td>100</td>
<td>400</td>
</tr>
<tr>
<td>Vertical length of plats, ( L_v ) (mm)</td>
<td>1100</td>
<td>2000</td>
</tr>
<tr>
<td>Horizontal length of plats, ( L_h ) (mm)</td>
<td>300</td>
<td>700</td>
</tr>
<tr>
<td>Total length of compact plates, ( L_c ) (mm)</td>
<td>300</td>
<td>600</td>
</tr>
<tr>
<td>Enlargement factor, ( \phi )</td>
<td>1.15</td>
<td>1.25</td>
</tr>
<tr>
<td>Plate thickness, ( t ) (mm)</td>
<td>0.3</td>
<td>0.9</td>
</tr>
<tr>
<td>Number of plates, ( N )</td>
<td>100</td>
<td>1900</td>
</tr>
</tbody>
</table>
6.3 Application example of plate heat exchanger

It is intended to design and optimized PHE (as shown in Fig. 6.1 and 6.2) constructed from SS304 and used for water to water heat exchange. The hot water at 338 K is entering into PHE with the mass flow rate of 140 kg/s. The cold water having the mass flow rate of 140 kg/s is supplied to PHE at a temperature of 295 K. The desired outlet temperature of the hot fluid is 318 K. Temperature dependent thermo-physical properties values of both the fluids are considered during the optimization procedure. So, the objectives are to find out the design parameter of PHE (i.e. a horizontal distance of ports, a vertical distance of ports, length of compact plates, port diameter, plate thickness, chevron angle, enlargement factor, and number of plates) for maximum overall heat transfer coefficient and minimum total pressure drop.

6.4 Results- discussion

Initially, single objective optimization of both objective functions is carried out to identify its behaviour with respect to each other. The control parameters of HTS and MOHTS algorithm used in the present investigation are listed in Table 6.2.

Table 6.2 Control parameters of HTS and MOHTS algorithm

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Selection probability of conduction phase</td>
<td>0-0.3333</td>
</tr>
<tr>
<td>Selection probability of convection phase</td>
<td>0.3333-0.6666</td>
</tr>
<tr>
<td>Selection probability of radiation phase</td>
<td>0.6666-1</td>
</tr>
<tr>
<td>Conduction factor</td>
<td>2</td>
</tr>
<tr>
<td>Convection factor</td>
<td>10</td>
</tr>
<tr>
<td>Radiation factor</td>
<td>2</td>
</tr>
</tbody>
</table>

The results of the single objective optimization are demonstrated in Table 6.3. It can be observed from the results that maximum overall heat transfer coefficient is obtained when the pressure drop is highest and vice versa. Thus, the results of single objective optimization reveal the conflicting between overall heat transfer coefficient and total pressure drop. So, the multi-objective optimization is carried out between conflicting thermal-hydraulic function with the help of MOHTS algorithm.
Table 6.3 Optimal result with single objective consideration

<table>
<thead>
<tr>
<th>Objectives</th>
<th>Pressure drop (kPa)</th>
<th>Overall heat transfer coefficient (kW/m²K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall heat transfer</td>
<td>1170.16</td>
<td>15.18</td>
</tr>
<tr>
<td>Pressure drop consideration</td>
<td>21.5</td>
<td>4.24</td>
</tr>
</tbody>
</table>

For the considered example of PHE, 200 design points are generated as Pareto optimal points during multi-objective optimization. Fig. 6.3 shows the distribution of Pareto optimal points in two dimensions objective space of overall heat transfer coefficient and total pressure drop. Design variables value of some selected Pareto points of Fig. 6.3 are displayed in Table 6.4 along with objective function value.

![Variation of overall heat transfer coefficient and total pressure drop during thermal-hydraulic optimization](image)

It can be observed from the results that overall heat transfer coefficient and total pressure drop reduced simultaneously from design point A to F. This behaviour is observed due to rises in length of compact plates, port diameter, a horizontal and vertical
distance of port, and enlargement factor from design point A to F. Overall, 98.2% reduction in heat transfer coefficient is observed with 72.1% reduction in total pressure drop between point A and F of Pareto solutions. All the solutions of Fig. 6.3 are non-dominated in nature because an improvement in the value of one objective function degrading other objective function value. Further, decision-making method namely LINMAP is utilized to select the best solution from Pareto optimal points. Detail description related to working of LINMAP method is available in the literature (Hwang and Yoon, 2012). The final solutions selected by the LINMAP method is shown in Fig. 6.3 and listed in Table 6.4.

Table 6.4 Optimal parameters for sample design point (A-F) during multi-objective consideration

<table>
<thead>
<tr>
<th>Design points</th>
<th>Port Diameter (mm)</th>
<th>L_v (mm)</th>
<th>L_h (mm)</th>
<th>L_c (mm)</th>
<th>Enlargement factor</th>
<th>Plate thickness (mm)</th>
<th>Number of plates</th>
<th>Chevron angle (D)</th>
<th>U (kW/m²K)</th>
<th>Pressure drop (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>103.1</td>
<td>1100</td>
<td>300</td>
<td>300</td>
<td>1.15</td>
<td>0.3</td>
<td>100</td>
<td>45</td>
<td>15.18</td>
<td>1170.16</td>
</tr>
<tr>
<td>B</td>
<td>145.2</td>
<td>1100</td>
<td>300</td>
<td>304.3</td>
<td>1.15</td>
<td>0.3</td>
<td>226</td>
<td>60</td>
<td>14.43</td>
<td>748.42</td>
</tr>
<tr>
<td>C</td>
<td>250.7</td>
<td>1100</td>
<td>300</td>
<td>375.3</td>
<td>1.15</td>
<td>0.3</td>
<td>100</td>
<td>60</td>
<td>12.81</td>
<td>441.43</td>
</tr>
<tr>
<td>D</td>
<td>320.6</td>
<td>1100</td>
<td>300</td>
<td>532.9</td>
<td>1.15</td>
<td>0.3</td>
<td>100</td>
<td>50</td>
<td>10.51</td>
<td>225.14</td>
</tr>
<tr>
<td>E</td>
<td>335.8</td>
<td>1100</td>
<td>300</td>
<td>600</td>
<td>1.15</td>
<td>0.3</td>
<td>101</td>
<td>65</td>
<td>7.5</td>
<td>75.02</td>
</tr>
<tr>
<td>F</td>
<td>400</td>
<td>2000</td>
<td>304.3</td>
<td>304.4</td>
<td>1.21</td>
<td>0.3</td>
<td>100</td>
<td>65</td>
<td>4.24</td>
<td>21.5</td>
</tr>
<tr>
<td>LINMAP</td>
<td>258.4</td>
<td>1100</td>
<td>300</td>
<td>304.4</td>
<td>1.15</td>
<td>0.3</td>
<td>100</td>
<td>65</td>
<td>12.71</td>
<td>428.83</td>
</tr>
</tbody>
</table>

Distribution of design variables corresponding to Pareto optimal points of Fig. 6.3 is shown in Fig. 6.4. It can be observed from the figure that optimum value of design variables namely plate thickness, vertical port distance, and enlargement factor remain invariable for majority Pareto optimal points. Thus, the effect of these design variables is not significant in obtaining Pareto optimal solutions between overall heat transfer coefficient and total pressure drop. Likewise, variation in the distribution of horizontal port distance is observed for few data points only. However, scatter distribution of port diameter, chevron angle, number of plates and length of compact plates are observed for
Pareto solutions. Variation in these design variables either increased or reduced value of both the objective functions simultaneously. Thus, these design variables produced conflicting behaviour between overall heat transfer coefficient and total pressure drop.

An investigation is carried out to identify the sensitivity of design variables to overall heat transfer coefficient and total pressure drop. Pareto solutions A-F is considered for this examination and results are shown in Figs. 6.5 and 6.6. It can be observed from Fig. 6.5 that overall heat transfer coefficient and total pressure drop is reduced with increasing port diameter, vertical port distance, horizontal port distance, and length of compact plates. Further, port diameter influence more on total pressure drop as compared to overall heat transfer coefficient while vertical port distance influence more on overall heat coefficient as compared to total pressure drop. Moreover, the influence of both the variables is acuter towards the minimum pressure drop design. On the other hand, horizontal port distance and length of compact plates affect equally on thermal-hydraulic function. As a result, uniform reduction in total pressure drop and overall heat transfer coefficient take place with the rise in the value of both the variables.
Fig. 6.5 Sensitivity of design variables to the optimized value of thermal-hydraulic function ((a) port diameter (b) vertical distance between ports (c) horizontal distance between ports (d) length of compact plates)

Fig. 6.6 Sensitivity of design variables to the optimized value of thermal-hydraulic function (a) enlargement factor (b) plate thickness (c) number of plates (d) chevron angle
Figure 6.6 shows the effect of enlargement factor, number of plates, chevron angle and plate thickness on the thermal-hydraulic function. It can be noted from the figure that narrow distribution of both the objective functions is observed with increasing the value of enlargement factor. Thus, enlargement factor is less sensitive to the optimized value of a thermal-hydraulic function. On the other hand, very few solution points of a thermal-hydraulic function are observed by changing plate number and chevron angle from its optimized value. This is due to the constraint violation with the change in the value of both the design variables. Thus, the optimized value of a thermal-hydraulic function is sensitive to both the design variables. At last, reduction in overall heat transfer coefficient with the simultaneous rise in total pressure drop is observed by increasing plate thickness from its optimized value. Here, very few solution points are observed towards minimum pressure drop and maximum heat transfer coefficient design due to constraint violation.

Convergence behaviour of the MOHTS algorithm is identified for the thermal-hydraulic objective of PHE. Results obtained through 30 individual run of MOHTS algorithm is used to plot the convergence. Pareto points obtained from external achieve is plotted on thermal-hydraulic coordinates. Fig. 6.7 demonstrates the convergence behaviour of MOHTS algorithm at the interval of 1000, 2000, 4000, 6000 and 8000 function evaluation respectively. It can be observed from the figure that the number of
solution points increases and its distribution became uniform with the proceeding of function evolutions. A complete Pareto front is obtained between conflicting thermal-hydraulic objectives at the end of 8000 function evaluation.

6.4.1 Experimental validation

In order to carry out experimental validation, a scale model of PHE is developed in the laboratory. A 1:5 scale model of PHE is developed with respect to optimum solution suggested by the LINMAP method (as listed in Table 6.4). Further, due to manufacturing constraints, geometric parameters considered in PHE model is somewhat deviated from the optimization results obtained using the HTS algorithm.

Figure 6.8 shows the complete experimental set-up with the enlarge view of PHE. An experimental set-up of PHE consists of flow measurement, pressure measurement, and temperature measurement. Further, all the measurement devices are calibrated before its application in the experimentation. The hot and cold fluids are supplied to the PHE at a rate of 17.5 kg/s. The supplied temperature of hot fluid is 338 K and that of cold fluid is
A set of experiments is performed to verify the optimized and experimental results.

Table 6.5 shows the comparison of the optimization results obtained using the HTS algorithm with experimental results along with the geometric parameters of PHE model. It can be observed from the results that the optimum value of overall heat transfer co-efficient obtained using the HTS algorithm is 30.65 kW/m\(^2\)K while that obtained during experimentation is 28.06 kW/m\(^2\)K. Similarly, the total pressure drop observed during optimization is 958.2 kPa while that obtained during experimentation is 1064.3 kPa. Thus, 8.87% deviation in overall heat transfer coefficient is observed between optimized value and experimental value. Similarly, 9.96 % deviation is observed between optimized and experimental value of total pressure drop of PHE.

Table 6.5 Comparison of optimized and experimental results of PHE

<table>
<thead>
<tr>
<th>Design/output parameters</th>
<th>Optimized value</th>
<th>Experimental value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Port Diameter (mm)</td>
<td>51.69</td>
<td>51</td>
</tr>
<tr>
<td>Vertical length of plats (mm)</td>
<td>220</td>
<td>220</td>
</tr>
<tr>
<td>Horizontal length of plats (mm)</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>Total length of compact plates (mm)</td>
<td>60.90</td>
<td>60</td>
</tr>
<tr>
<td>Enlargement factor</td>
<td>1.15</td>
<td>1.15</td>
</tr>
<tr>
<td>Plate thickness (mm)</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>Number of plates</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Chevron angle (D)</td>
<td>65</td>
<td>65</td>
</tr>
<tr>
<td>Overall heat transfer coefficient (kW/m(^2)K)</td>
<td>30.65</td>
<td>28.06</td>
</tr>
<tr>
<td>Pressure drop (kPa)</td>
<td>958.20</td>
<td>1064.3</td>
</tr>
</tbody>
</table>

Apart from the human error and measurement error during experimentation, one additional reason behind the deviation of optimized and experimental results are manufacturing constraints involved during the development of PHE. Due to manufacturing constraints the geometric dimension of the developed PHE is deviated from the optimized value which further affects the output values of the PHE obtained during the experimentation. However, the optimized and experimental results of PHE are within good agreement and one can used the proposed approach for the optimization and development of other type of heat exchanger also.