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Thermal and hydraulic optimization of plate heat exchanger using multi objective genetic algorithm

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ABSTRACT

In this paper thermal and hydraulic optimization of water to water chevron type plate heat exchanger is presented. The optimization is performed using the multi objective genetic algorithm in MATLAB optimization environment. Constrain matrix is a set of different geometrical parameters of plate heat exchanger within the logical bounds. The two objective functions are pressure drop of hot side and heat transfer. Due to conflicting nature of these objective functions, no single solution can satisfy both of the objective function simultaneously. The increase in heat transfer will results in increase in pressure drop, therefore, optimization results are presented as Pareto Front. Multi objective genetic algorithm tool was employed to find a set of optimum solution which was trade-off between pressure drop and heat transfer. At the end, sensitivity analysis was performed to analyse the effect of geometrical parameters of heat exchanger on thermal and hydraulic performance. The sensitivity results show that the heat transfer and pressure drop are greatly affected by the vertical port centre distance, plate spacing and number of thermal plates.

1. Introduction

Heat exchanger is an essential component of the almost every thermal system. Their design should be well suited to the application in term of thermal, economical and hydraulic performance. The performance of heat exchanger is strongly correlated with its geometrical parameters and operating conditions. If the operating conditions are fixed, then geometry of the heat exchanger play vital role in the hydraulic and thermal performance of heat exchanger. Thus, the optimum performance can be ensured at a specific geometry of heat exchanger.

Genetic Algorithms (GAs) is an evolutionary algorithm based optimization technique for the optimization of non-linear and complex systems. The optimum geometrical parameters of the plate heat exchanger can be estimated by using genetic algorithm. Genetic Algorithms concept was introduced in the 1970s but the first main works related to GAs are introduced by Holland and De Jong in 1975 [1,2]. However, the application of GAs in the field of the heat transfer is more recent. The major area of applications of GAs in the field of heat transfer is the sub research area related to design, shape, network, placing and ordering of thermal systems. About 74% publication of GAS in the field transfer are from the area of heat exchanger, heat exchanger network, HVAC, power generation, conduction heat transfer and radiation heat transfer [3]. A lot of studies have been conducted to improve the heat transfer and reduce the pressure drop in heat exchanger [4]. Among other methods of optimization, GAs had been widely used to optimize the

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Table 1
Review of optimization studies of the heat exchanger using Genetic Algorithms.

Ref.	System description	Objective function	Genetic algorithm	
			Type	Variables
[5]	Heat exchanger networks	Minimize: total annual cost Maximize: system reliability	Multi	7
[6]	Plate-fin heat exchanger	Minimize: total annual cost Maximize: heat transfer rate	Multi	6
[7]	Shell-and-tube heat exchanger	Minimize: total cost Maximize: effectiveness	Multi	8
[8]	Plate-fin heat exchanger	Minimize: friction factor f Maximize: Colburn factor j	Multi	4
[9]	Fin-and-tube heat exchanger	Minimize: total weight Minimize: total annual cost	Multi	7
[10]	Shell and tube heat exchanger	Maximize: Effectiveness Minimize: Total Cost, pressure drop and No. of entropy generation units	Multi	6
[11]	Plate-fin heat exchanger	Minimize: friction factor f Maximize: Colburn factor j	Multi	4
[12]	Plate heat exchanger	Minimize: cost of heat exchanger Minimize: pressure drop	Multi	4
[13]	Plate heat exchanger	Maximize: heat transfer coefficient Minimize: pressure drop	Multi	6
[14]	Plate-fin heat exchanger	Minimize: Total Volume Minimize: Total Cost	Multi	3
[15]	Shell-and-tube heat exchanger	Minimize: Total Cost	Single	6
[16]	Plate-fin heat exchanger	Minimize: heat transfer units (NTU)	Single	4
[17]	Plate heat exchanger	Minimize: Heat transfer area	Single	4
[18]	Shell-and-tube heat exchanger	Minimize: total annual cost	Single	3
[19]	Organic Rankine Cycle	Minimize: Total Investment Cost Minimize: Thermal Efficiency	Multi	7

design of heat exchangers. In recent years, extensive research has been conducted on heat exchanger optimization. But, these studies primarily focusing on shell & tube and plate fin heat exchanger at system level. A comprehensive review of the application of GAs in the design and optimization of heat exchange is presented in Table 1.

It can be seen that most of these studies deal with the optimization of the shell and tube heat exchanger, plate and fin heat exchanger, and fin and tube heat exchanger. These optimization studies primarily involve the reduction of cost and maximum heat transfer or effectiveness of heat exchanger. However, there are only few studies that deal with the optimization of plate heat exchanger.

The cost of heat exchanger, annual operating cost, weight and volume depends on the heat transfer and pressure drop characteristics of the heat exchanger for a particular application. Therefore, it is of crucial importance to investigate the effect of geometrical parameters on heat transfer and pressure drop which are the prerequisite for selection of heat exchanger. There are limited number of studies related to plate heat exchanger optimization and little known about the effect of primary geometrical parameters on the thermal and hydraulic performance of plate heat exchanger. The main focus of the present study is

- Investigate the effect of geometrical parameters of heat exchanger on thermal and hydraulic performance of heat exchanger.
- Optimize the geometrical parameters for maximum heat transfer and minimum pressure drop by employing multi objective genetic algorithm.

Structure of the paper is as follows; Section 1 deals with the introduction and literature review, Section 2 deals with the plate heat exchanger design, the optimization methodology is discussed in Section 3, while the results are discussed ins Section 4 of the paper.

2. Plate heat exchanger

Plate heat exchangers are most widely used in various applications including heating, cooling, heat recovery, condensation and evaporation. The complex geometry profile of plate heat exchanger enables high degree of turbulence which results in high heat transfer.

2.1. Geometry

The geometrical configuration of chevron type plate heat exchanger is shown in Fig. 1.

The geometrical parameters can be calculated from basic geometry of the thermal plate. The effective length and width of plate heat exchanger [20] is given by

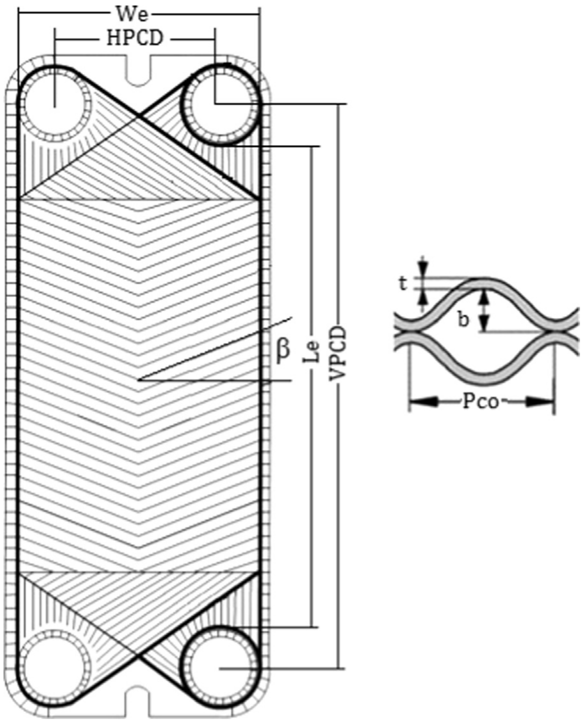

Fig. 1. Geometry of plate heat exchanger (PHE).

$$Le = VPCD - D \tag{1}$$

$$We = HPCD + D + 0.015 \tag{2}$$

Whereas D is port diameter, $VPCD$ & $HPCD$ is vertical port centre distance and horizontal distance respectively. The area of thermal plate and total area of heat exchanger [20] is given by

$$A_P = We \times Le \tag{3}$$

$$A = n \times A_P \tag{4}$$

Whereas n represent number of thermal plates. The hydraulic diameter [21] of the plate heat exchanger

$$D_h = \frac{4 \times (b \times We)}{2 \times (b + We \times \Phi)} \cong \frac{2b}{\Phi} \text{ whereas } We \ll b \tag{5}$$

Whereas b is plate thickness and Φ is enlargement factor.

2.2. Heat transfer

Heat transfer depends on operating conditions and geometry of the heat exchanger. The mass velocity of each side of heat exchanger and Reynolds Number [22] is calculated by

$$G = \frac{\dot{m}}{(N_c \times b \times We)} \tag{6}$$

$$R_e = \frac{G \times D_h}{\mu} \tag{7}$$

Whereas N_c is number of channels and μ is the viscosity of the fluid. For water to water chevron type plate heat exchanger, the heat transfer correlations [22] are given by

$$N_U = 0.2267 \times (R_e)^{0.631} \times (P_r)^{0.33} for \ \ 15 \le R_e \ \le 15000 \tag{8}$$

Whereas P_r is the Prandtl Number. The convective heat transfer coefficient for both sides is given by

$$h = \frac{N_U \times k}{D_h} \tag{9}$$

Whereas k is thermal conductivity of the fluid. The overall heat transfer coefficient & number of transfer units are given by

$$\frac{1}{U} = \frac{1}{h_c} + \frac{t}{k_p} + \frac{1}{h_h} \tag{10}$$

$$NTU = \frac{UA}{C_{min}} \text{ whereas } C_{min} = (\dot{m})_c \times (C_P)_c or C_{min} = (\dot{m})_h \times (C_P)_h \tag{11}$$

Whereas the subscript c & h denotes the hot and the cold side while k_p & t denotes thermal conductivity and thickness of the plate. For the counter flow heat exchanger the effectiveness [20] and heat transfer is given by

$$\varepsilon = \frac{[1-e^{\{-1\times NTU\times(1-Cr)\}}]}{[1+Cr \times e^{\{-1\times NTU\times(1-Cr)\}}]}; C_r = C_{min}/C_{max} \tag{12}$$

$$Q = \varepsilon \times C_{min} \times (T_{h,i} - T_{c,i}) \tag{13}$$

2.3. Pressure drop

The pressure drop across the plate heat exchanger consists of four components; pressure drop due to acceleration of the fluid, due to change in elevation, due to inlet/exit manifolds, and due to friction inside the corrugated plate heat exchanger. Previous research shows that the magnitude of pressure drop due to acceleration of the fluid & due to change in elevation is very small as compared to frictional and port pressure loss. In this study only frictional and port pressure loss is considered. The friction factor [22] for both sides is given by

$$f = 0.572 \times (R_e)^{-0.217} \text{ if } R_e > 550 \tag{14}$$

$$f = 26.34 \times (R_e)^{-0.830} \text{ if } R_e \leq 550 \tag{15}$$

The frictional pressure loss [22] will be

$$(\Delta P)_f = \frac{4\times f \times Le \times G^2}{2\times\rho \times D_h} \tag{16}$$

Where the port velocity and manifold pressure loss is given by

$$V = \frac{\dot{m}}{(\pi/4) \times D^2} \tag{17}$$

$$(\Delta P)_P = \frac{1.5 \times (V)^2}{2 \times \rho} \tag{18}$$

Total pressure loss is given by

$$\Delta P = (\Delta P)_f + (\Delta P)_P \tag{19}$$

The fluid properties are calculated by using REFPRP Version 9. The properties are calculated at mean temperature of the fluid.

3. Optimization

Depending on application of heat exchanger there is a choice of pressure drop and heat transfer, depends on which factor is most important. If we increase the heat transfer the pressure drop will also increase and vice versa. Therefore an appropriate optimization tool is necessary to find a set of optimum solution to this problem. The set of optimum solution for such case is called Pareto Front Solution.

3.1. Genetic algorithm

Genetic algorithm (GA) is a search heuristic which is based on techniques inspired by natural evolution, such as inheritance, mutation, selection, and crossover. The Genetic algorithm process modifies the given population within the given constrain bounds and limits. In each step, the selected population is used as parents to produce children for the next generations. The population which has higher fitness value is selected for next iteration. This process continues until a set of population is obtained which satisfy the objective function as well as constrains. For single objective function, GA easily yields the maximum and minimum value if the constrain range is not large enough.

3.2. Multi objective genetic algorithm

For more than one objective function multi objective genetic algorithm is applied. Mathematically the multi objective genetic algorithm can be represented as

$$min\,[f_1(x), f_2(x), f_3(x)\ldots\ \ldots\ \ldots f_n(x)]; x \epsilon X \tag{20}$$

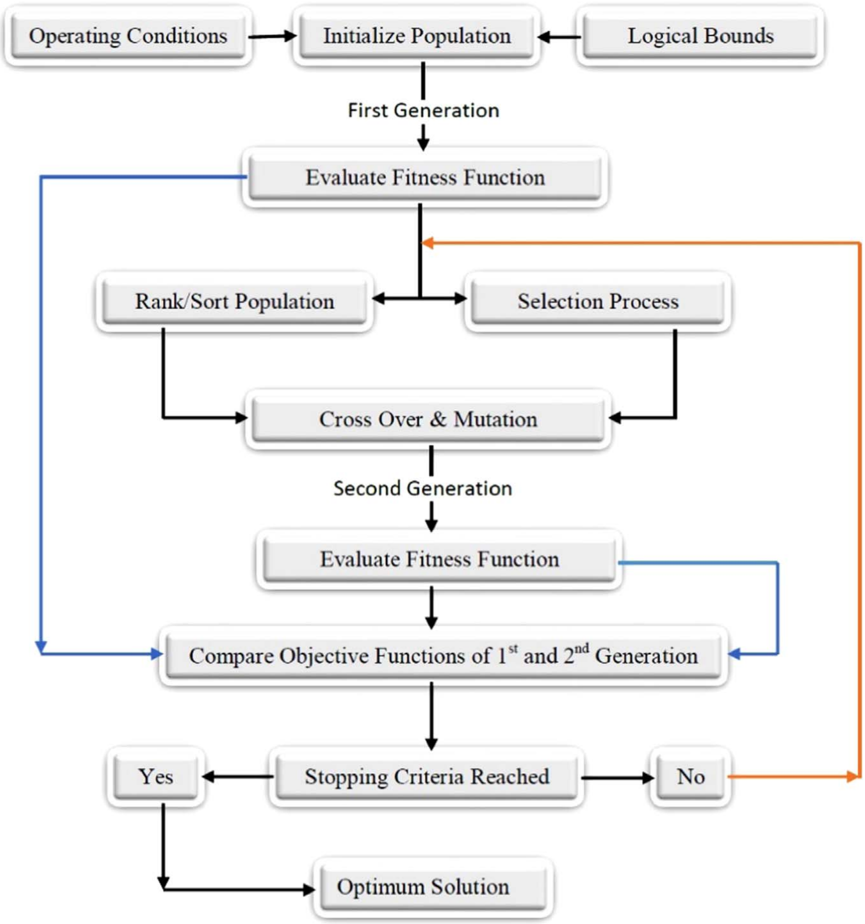

Fig. 2. Main Steps of multi objective genetic algorithm.

Whereas

n = no. of objective function & n ≥ 2

X = feasible set of decision vectors defined by constraint functions

In this article, the constrain matrix consist of horizontal port centre distance, vertical port centre distance, enlargement factor, port diameter, plate thickness, number of thermal plates and plate spacing. First objective function is maximum heat transfer and second objective function is minimum pressure drop. In the MATLAB program, negative sign is added to heat transfer to get the maximum value. The main steps of multi objective genetic algorithm are shown in Fig. 2.

4. Result and discussion

The chevron angel of plate heat exchanger is 60 degree and is made of SS316. The operating conditions are listed in Table 2. The constrain matrix and constant geometrical parameters for optimization & sensitivity analysis are presented in Table 3.

4.1. Pareto front solution

Due to inverse relationship between pressure drop and heat transfer no single solution can satisfy both objective function at single

Table 2
Operating Conditions of Plate Heat Exchanger.

Fluid name	Flow rate (Kg/s)	Inlet temperature (C)	Pressure (bar)
Hot (Water)	22	85	3
Cold (Water)	20	25	3

Table 3
Geometrical bounds and constant value of geometrical parameters.

Range type	HPCD (m)	VPCD (m)	D (m)	Φ	b (m)	t (m)	n
Lower bound	0.1	0.4	0.1	1.11	0.0015	0.0003	10
Upper bound	0.8	1.5	0.3	1.25	0.005	0.003	200
Constant value	0.6	1.3	0.2	1.17	0.002	0.0005	150

value. The set of optimum solution is presented in Table 4.

For multi objective genetic algorithm, the population type is considered double vector and the population size is chosen to be 105. The selection type was tournament which means that the fittest individual have more chances of selection. The reproduction cross over fraction was set at 0.8. The Pareto front solution is obtained after 71 iterations and result are shown in Fig. 3. The terminating criteria is based on the change in optimum solution which is selected to be 10^{-6}. When the average change in optimum solution reached to this value the program is terminated. The graphical representation of Pareto front solution is shown in Fig. 3.

4.2. Sensitivity analysis

The effect of horizontal port centre distance, vertical port centre distance, plate spacing, plate thickness and number of thermal plates on thermal and hydraulic performance of PHE is discussed in detail. Fig. 4 shows the effect of horizontal port centre distance on heat transfer and pressure drop.

With the increase of port centre distance the amount of heat transfer increases due to the increase in heat transfer area while pressure decreases due to decrease of effective width which results in low Reynolds Number & mass velocity. Therefore frictional pressure drop decreases with increase in horizontal port centre distance. This result shows that horizontal port centre distance can be increased to the allowable cost to decrease pressure and increase heat transfer.

In Fig. 5 the effect of vertical port centre is shown. The heat transfer increases with the increase of the vertical port centre distance as the area of thermal plate is increased. But due to the increase of the vertical port centre distance with constant width of plate, the frictional pressure drop will also increase due to increase in the flow length. Therefore, the overall pressure drop will be increased. In this case, there is a compromise between heat transfer and pressure drop. Heat transfer can be maximized under the allowable limits of pressure drop.

Fig. 6 show the effect of plate spacing on the heat transfer and pressure drop. Heat transfer decreases with the increase in plate spacing while pressure across the plate heat exchanger is also decreased.

This is because with the decrease of plate spacing, the hydraulic diameter will increase and ultimately the pressure drop will decrease. At low plate spacing, the decrease in pressure is very sharp. Fig. 7 shows the variation of pressure drop and heat transfer with plate thickness. Since plate corrugated depth is kept constant in this i.e. 0.008 m so the change in plate thickness does not affect any other geometrical parameter. The pressure drop will be constant because all other geometrical parameters are constant but heat transfer increases with decrease of plate thickness as the thermal resistance will decrease. Therefore the plate thickness can be

Table 4
Optimum Solution of Plate Heat Exchanger.

HPCD (m)	VPCD (m)	D (m)	Φ -	b (m)	t (m)	n -	ΔP (kPa)	Q kW
0.797	1.494	0.20	1.23	0.00151	0.00044	183	3.937	1673
0.788	0.463	0.30	1.19	0.00479	0.00130	182	0.070	72
0.784	0.433	0.30	1.17	0.00500	0.00059	180	0.066	57
0.797	1.492	0.24	1.23	0.00156	0.00042	183	3.132	1608
0.789	1.474	0.28	1.21	0.00403	0.00043	182	0.232	644
0.796	1.492	0.22	1.23	0.00152	0.00044	183	3.547	1648
0.797	1.494	0.26	1.22	0.00178	0.00045	183	1.967	1412
0.796	1.489	0.27	1.23	0.00277	0.00048	183	0.570	936
0.796	1.490	0.28	1.22	0.00166	0.00042	183	2.276	1489
0.796	1.492	0.24	1.23	0.00159	0.00047	183	2.888	1562
0.789	1.257	0.30	1.18	0.00470	0.00059	180	0.134	440
0.797	1.494	0.20	1.23	0.00151	0.00044	183	3.937	1673
0.797	1.489	0.26	1.23	0.00197	0.00041	183	1.503	1296
0.790	1.121	0.30	1.19	0.00463	0.00110	181	0.126	375
0.789	1.358	0.29	1.21	0.00421	0.00054	183	0.181	552
0.796	1.493	0.21	1.23	0.00150	0.00044	183	3.821	1670
0.796	1.487	0.28	1.20	0.00261	0.00045	183	0.627	978
0.797	1.491	0.27	1.22	0.00204	0.00040	183	1.279	1245
0.784	0.696	0.30	1.18	0.00498	0.00059	181	0.083	173
0.789	1.474	0.28	1.21	0.00391	0.00049	182	0.247	661
0.792	0.921	0.30	1.19	0.00456	0.00100	181	0.113	291

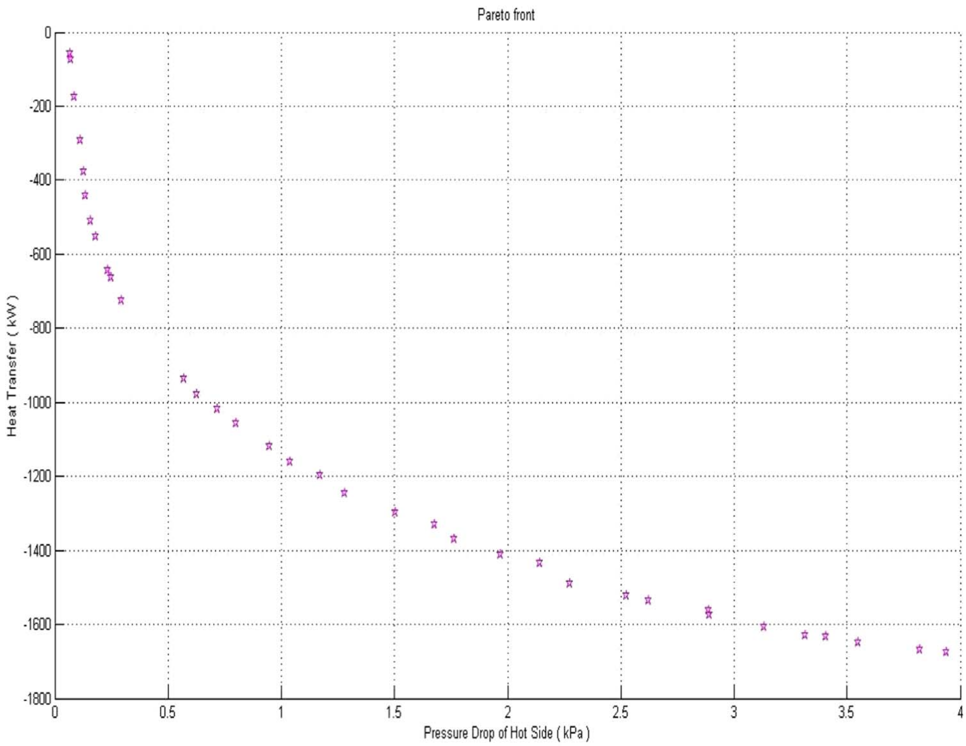

Fig. 3. Pareto front solution for minimum pressure drop and maximum heat transfer.

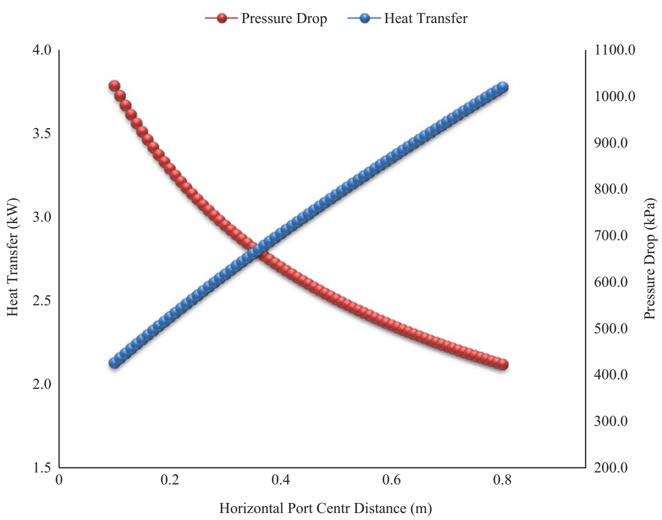

Fig. 4. Effect of horizontal port centre distance on heat transfer & pressure drop.

decreased depending on the maximum operating pressure. The plate strength should be enough to withstand maximum pressure.

In Fig. 8, the effect of number of thermal plates is presented. It is known that heat transfer increases with increase in number of thermal plate as heat transfer is increased. The pressure drop decreases with increase in number of thermal plates as the frictional pressure drop is significantly decreased. The decrease in pressure is very sharp initially. The maximum number of thermal plates can be selected depending on cost and required amount of heat transfer as well as allowable pressure drop.

5. Conclusion

In the article the thermal and hydraulic optimization of the plate heat exchanger is presented. For a logical bound of geometrical parameters of plate heat exchanger the heat transfer and pressure drop are optimized. Due to inverse relationship between pressure

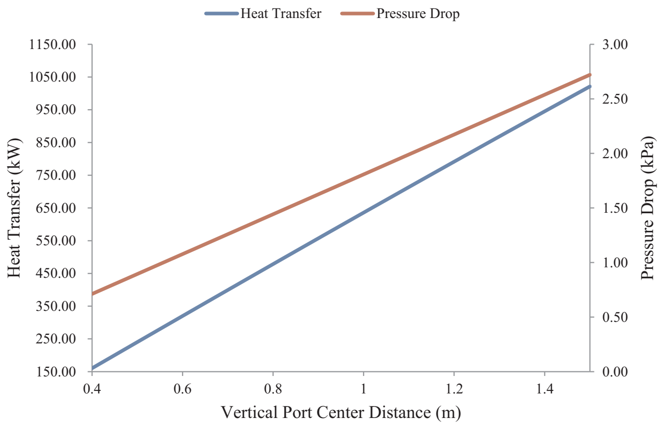

Fig. 5. Effect of vertical port centre distance on heat transfer & pressure drop.

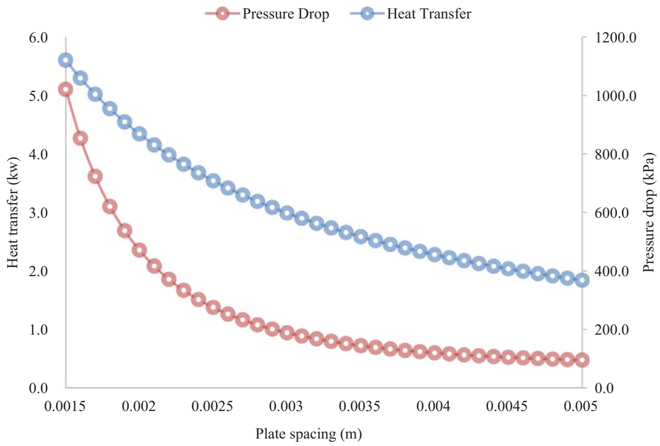

Fig. 6. Effect of plate spacing on heat transfer & pressure drop.

Fig. 7. Effect of plate thickness on heat transfer & pressure drop.

drop and heat transfer no single value can optimize both objective function. Multi objective genetic algorithm is used in MATLAB program to find the optimum solution which will satisfy both objective functions to some extent. It is quite helpful for design engineer to choose the optimum solution from Pareto Front according to their choice of pressure drop and heat transfer. The sensitivity

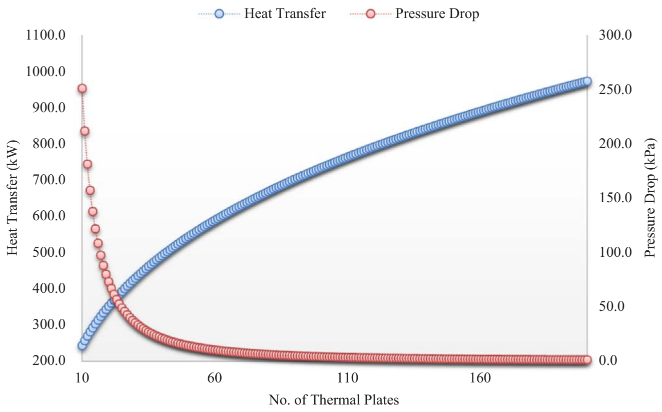

Fig. 8. Effect of number of thermal plates on heat transfer & pressure drop.

analysis is also performed to study the effect of the individual geometrical parameter on heat transfer and pressure drop. Results show that horizontal port centre distance and number of thermal plates shows same trend for pressure drop and heat transfer. The heat transfer & pressure drop increases with increase in horizontal port centre distance & number of thermal plates. The optimum solution then will be towards maximum width and maximum number of plates which ultimately depends on economic analysis. For vertical port centre distance and plate spacing there is a trade-off between pressure drop and heat transfer. The optimization and sensitivity analysis can play major role in deciding the optimum design for design engineers.

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