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ORIGNAL ARTICLE

Design, optimization and performance of plate heat exchanger

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Abstract

The plate heat exchanger is an important heat recovery equipment which transfers heat from high temperature fluid to low temperature fluid with help of corrugated plates. The plates of plate heat exchanger provide large surface area for heat transfer per unit volume. They have both the compactness and high thermal performance. The plate heat exchangers are generally used in the many fields of engineering like refrigeration, HVAC, petrochemical, liquid food, beverage, dairy and heath care sector.

In the present study, thermo-hydraulic performance of plate heat exchanger has been presented. A range of geometrical and operating parameters have been considered for numerical simulation of plate heat exchanger. The performance of plate heat exchanger has been obtained in the terms of the effectiveness, heat transfer and pressure drop for two fluids. Further, the plate heat exchanger has been optimized for the single objective as well as multi-objective functions using genetic algorithm. The objective of this study is to minimize two objective functions independently which are entropy generation unit and total annual cost for the specified heat transfer rate. The optimum values of the geometrical parameters have been obtained after optimization for single objective as well for multi objective optimization. The optimum results for unconstrained and constrained conditions have been compared.

1. Introduction

A heat exchanger is device which is used to exchange thermal energy between two or more fluids, at different temperatures and in thermal contact. Heat exchangers are generally used for heating and cooling of a fluid stream of concern and evaporation or condensation of single or multicomponent fluid streams. The thermodynamic model of waste heat recovery in organic Rankine cycle to compare the thermodynamic and thermo-economic performance of the working fluid in plate heat exchanger was developed and It has been observed that thermo-economic optimization tells to use higher evaporative temperature because use of the higher temperature it decreases the cost of the system [1]. the effect of the horizontal port distance and number of plates on pressure drop and heat transfer are same. The increment in horizontal port distance and number of plates will cause increment in pressure drop and heat transfer. For vertical port distance and spacing between two plates, heat transfer and pressure drop both have trade off [2]. The optimal structure and heat transfer coefficient of welded plate heat exchanger by using the grey relation theory and CFD simulation was analyzed. Firstly, grey theory has been used to design the experiment and simulation. CFD simulation results have been verified by optimal results. It has been observed that the factors which has greatest influence on heat transfer coefficient of welded plate heat exchanger are short axis and long axis distance by grey relation theory and simulation. While bundle spacing has least impact on heat transfer coefficient of welded plate heat

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exchanger [3]. The thermo-hydraulic performance of plate heat exchanger for model which will be employed in CO_2 supercritical Rankine cycle to recover waste heat at low temperature was calculated. Conditions for which this model will be used are inlet temperature 100°C and temperature difference at outlet of 5°C and input heat power of 20kW. It has been observed that there are couple of boundary conditions which consist of outlet pressure and temperature difference are 130 bar and 2°C that maximizes thermal efficiency [4]. Multi objective optimization gives the wide variety of optimal solutions which can be used by users according to their application and requirement and it has been found that there is 8.87% deviation in overall heat transfer coefficient and 9.96% deviation in total pressure drop between optimization and experimental results [5]. The new correlations for the two phase pressure drop which is occurring through the channels of plate heat exchanger with small gaps was presented. The gap between two plates varies from 0.4 to 4 mm and channel width is fixed to 20 mm. Test fluids were water and air, superficial velocity of water varies from .03 to 2.39 and for air it varies from .05 to 18.7. Lockchart-Martinelli type correlation was used to express two phase frictional multiplier. Effect of mass flux and gap size were considered in expressing the correlations [6]. The heat flux, saturated flow boiling heat transfer and pressure drop increases almost linearly in R-410 [7]. If water superficial velocity is more than 1.0 m/s and air velocity is low, the entire channel area will be covered by a continuous liquid phase. All superficial velocity has been tested and found that heat transfer coefficient of air/water mixture is more than that of water without air and increase in heat transfer is significant at low water superficial velocity which shows that liquid at bottom of furrow increases the heat transfer [8]. The shear controlled two phase flow can exist in the turbulent flow conditions of non-adiabatic two phase system in furrow of plates. It has been observed that heat transfer is 3-4 times more than heat transfer calculated by nusselt theory of laminar film [9]. The annular-liquid bridge pattern is appeared in both upward and downward flow. Bubbly flow pattern is appeared only in upward flow and slug pattern is appeared only in downward flow and it has been observed that two phase pressure drop has the significant effect of the variation of velocity of water and air [10].

In this work, Performance analysis of plate heat exchanger which is based on geometrical parameters, vertical port distance (L_v), horizontal port distance (L_h) and port diameter (D_p) and operating parameters, mass flow rate of hot fluid (m_h) and mass flow rate of cold fluid (m_c) was analyzed and optimize the plate heat exchanger with single objective for unconstrained and constrained conditions.

2. Numerical methodology

Thermo-hydraulic performance analysis of plate heat exchanger has been considered with geometrical parameters and operating parameters. The performance of plate heat exchanger has been taken in the terms of effectiveness, heat transfer and pressure drop for hot fluid (water) and cold fluid (water) of plate heat exchanger. The sensivity analysis has been done to study the effect of geometrical parameters as well as operating parameters on the performance of plate heat exchanger. Following assumptions have been made as below.

1. The inlet and outlet properties of hot and cold fluids are uniform and constant.
2. The working fluids has been used as water with constant specific heat.
3. Thera is no mixing between hot and cold fluids during the heat exchange.
4. The heat transfer is being done adiabatically, it means that three is no heat gain from surrounding and heat loss to surrounding.
5. Hot and cold fluids are flowing in opposite directions (counter-flow directions).

2.1 Thermo-hydraulic modeling

In this section, the thermo-hydraulic modeling of plate heat exchanger has been mentioned. Thermo- hydraulic modeling is used to find the performance parameters of plate heat exchanger such as effectiveness, heat transfer and pressure drop for hot and cold fluid streams. The geometrical parameters and fluid properties have been discussed in the following sections.

2.2 Core geometrical parameters

The following study has been discussed [11]. In this study geometrical parameters like vertical port distance, horizontal port distance, port diameter, width of plate, thickness of plate and chevron angle of corrugated plate have been shown in the given fig.1. These geometrical parameters are very important to study the performance of plate heat exchanger.

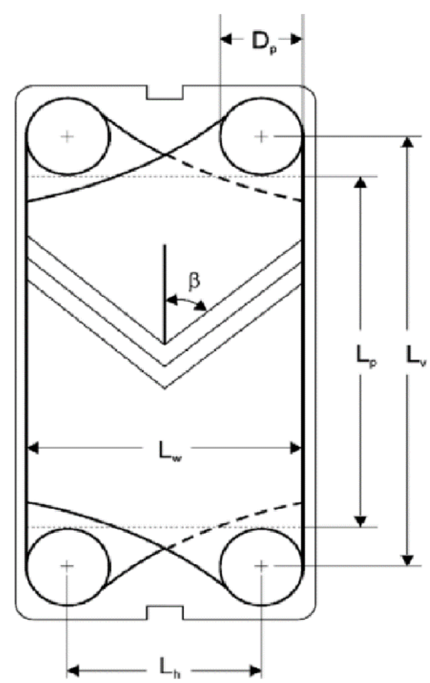

Figure 1: Basic geometry of chevron angle

Fig.1 shows the basic geometry of the plate of plate heat exchanger. The large variety of plate are available but most widely used plates are chevron type plate which is known as corrugated plates. Because of corrugation it has large surface area for heat transfer. Corrugations of plate change the direction and velocity of flowing fluid because of that turbulence is created which enhances heat transfer between two fluids. β angle is shown in the fig. 1 which is known chevron angle. The adjacent plate always has reversed chevron angle because when two plates are clamped together there are many contact points which will support the system and we can use the thin plates in plate heat exchanger.

2.2.1 Heat transfer area

The surface area of corrugated plate is more than that of flat plate. To calculate this increment in surface area of plate there is one factor which is called enlargement factor (ϕ). The enlargement factor can be defined as ratio of developed length and projected length.

$$\Phi = \frac{\text{developed length}}{\text{projected length}} = \frac{A_1}{A_{1p}} \tag{1}$$

Where A_1 is actual heat transfer area and A_{1p} is projected heat transfer area of the plate.

$$A_{1p} = L_p \cdot L_w \tag{2}$$

L_p and L_w can be calculated by vertical port distance (L_v) and horizontal port distance (L_h) and port diameter (Dp)

$$L_p = L_v - D_p \tag{3}$$

$$L_w = L_h + D_p \tag{4}$$

2.2.2 Mean flow channel gap

The two adjacent plates with gasket are clamped together a conduit is formed that is known as channel. Mean channel gap can be defined as

$$b = p - t \tag{5}$$

Where p is pitch of plate and t is thickness of plate. Mean flow channel gap is used to calculate the value of mass flow rate and Reynold number. If manufacturer does not specify the plate pitch it can be determined by the compressed plate pact (L_c) and total number of plates (N_t).

$$p = \frac{L_C}{N_t} \tag{6}$$

2.2.3 Channel hydraulic diameter

Hydraulic diameter D_h can be defined as

$$D_h = \frac{4X channel\ flow\ area}{wetted\ perimeter} = \frac{4A_c}{P_w} \cong \frac{2b}{\Phi} \tag{7}$$

$$D_h = \frac{4\,(b)(Lw)}{2(b+Lw\phi)} \cong \frac{2b}{\Phi} \tag{8}$$

Where $b \ll L_{w.}$

Number of channels per pass

$$N_{cp} = \frac{N_t - 1}{2N_p} \tag{9}$$

Where N_t is total number of plates and N_p is number of pass. Effective number of plates

$$\mathrm{N_e} = \mathrm{N_t} - 2 \tag{10}$$

Mass flow rate per channel

$$\dot{\mathrm{m}}_{\mathrm{ch}} = \frac{\dot{\mathrm{m}}}{\mathrm{N_{cp}}} \tag{11}$$

Area of free flow

$$A_{ff} = L_w\,\mathrm{b} \tag{12}$$

The value of $\dot{\mathrm{m}}_{\mathrm{ch}}$ can be different of hot and cold fluids

2.2.4 Heat transfer coefficients

The channel mass velocity is given by

$$G_{ch} = \frac{\dot{\mathrm{m}}_{\mathrm{ch}}}{N_{cp}\ A_{ff}} \tag{13}$$

The Reynold number of hot fluid can be given by

$$Re_h = \frac{G_h D_h}{\mu_h} \tag{14}$$

The Reynold number of cold fluid can be given by

$$Re_c = \frac{G_c D_c}{\mu_c} \tag{15}$$

The hot fluid heat transfer coefficient can be given by using C_h=.3 and n=.669 [11]

$$Nu_h = \frac{h_h D_h}{k} = C_h (Re_h)^n (Pr)^{1/3} \tag{16}$$

$$Nu_h = \frac{h_h D_h}{k} = 0.3 (Re_h)^{0.663} (Pr)^{1/3} \tag{17}$$

The cold heat transfer coefficient can be given by

$$Nu_c = \frac{h_c D_h}{k} = C_h (Re_c)^n (Pr)^{1/3} \tag{18}$$

Overall heat transfer coefficient

$$\frac{1}{U_f} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{t}{k} + FF \tag{19}$$

Where k is thermal conductivity and FF is fouling factor. The actual heat transfer is

$$Q_f = U_f \cdot A_e \cdot LMTD \tag{20}$$

Where $A_e = N_e . A_1$ and LMTD is Logarithmic Mean

Temperature Difference.

$$LMTD = \frac{(T_{hi}-T_{co})-(T_{ho}-T_{ci})}{ln(\frac{T_{hi}-T_{co}}{T_{ho}-T_{ci}})} \tag{21}$$

Required heat duty

$$Q_r = (\dot{m}C_p)_c(\Delta T_c) = (\dot{m}C_p)_h(\Delta T_h) \tag{22}$$

2.2.5 Pressure drop analysis

Pressure drop through channels

$$\Delta P_c = 4f\frac{L_v N_p}{D_h} \cdot \frac{{G_c}^2}{2\rho} \tag{23}$$

Pressure drop through port of the plate

$$\Delta P_p = 1.4N_p\frac{{G_p}^2}{2\rho} \tag{24}$$

Total pressure drop through both channel and port [11]

$$\Delta P_t = 4f\frac{L_v N_p}{D_h} \cdot \frac{{G_c}^2}{2\rho} + 1.4N_p\frac{{G_p}^2}{2\rho} \tag{25}$$

Friction factor is given by [2]

$$f = \frac{1.441}{(Re)^{0.206}} \tag{26}$$

The Effectiveness is given by

$$\varepsilon = \frac{T_{hi}-T_{ho}}{T_{hi}-T_{ci}} \quad \text{or} \quad \varepsilon = \frac{T_{co}-T_{ci}}{T_{hi}-T_{ci}} \tag{27}$$

2.3 Case study

In the proposed model, plate heat exchanger has been considered for thermo-hydraulic designing. For this purpose, geometrical parameters and operating parameters have been used within range. The range is given below

Vertical port distance (L_v)	=	0.3 - 1.5 (m)
Horizontal port distance (L_h)	=	0.3 -0.7 (m)
Port diameter (D_p)	=	0.1- 0.4 (m)

The operating parameters given below

Mass flow rate of hot fluid (m_h)	=	10- 140 (kg/s)
Mass flow rate of cold fluid (m_c)	=	10- 140 (kg/s)

The analysis has been done for plate heat exchanger by changing geometrical as well as operating parameters within given range of parameters. The effect of geometrical and operating parameters on the effectiveness, heat transfer and pressure drop of plate heat exchanger has been determined by plotting the curves within given range. The operating parameters for hot fluid given below

Inlet temperature (T_h)	338 (K)
Specific heat (C_p)	4183 (J $kg^{-1}K^{-1}$)
Density (ρ)	985 (kg m^{-3})
Dynamic viscosity (μ)	5.09×10^{-4}(N-s m^{-2})
Prandtl number (P_r)	3.31
Thermal conductivity (k)	0.645($Wm^{-1}K^{-1}$)

The operating parameters for cold fluid given below

Inlet temperature (T_h)	295 (K)
Specific heat (C_p)	4178 (J $kg^{-1}K^{-1}$)
Density (ρ)	995 (kg m^{-3})
Dynamic viscosity (μ)	7.09×10^{-4}(N-s m^{-2})
Prandtl number (P_r)	5.19
Thermal conductivity (k)	0.617(kg m^{-3})

The performance of plate heat exchanger has been studied in the terms of effectiveness, heat transfer and pressure drop.

The total pressure drop

$$\Delta P_t = 4f\frac{L_v N_p}{D_h} \cdot \frac{{G_c}^2}{2\rho} + 1.4N_p\frac{{G_p}^2}{2\rho} \tag{28}$$

The effectiveness [2]

$$\varepsilon = \frac{[1-exp\{-NTU(1-Cr\}]}{[1+Crexp\{-NTU(1-Cr)\}]} \tag{29}$$

The heat transfer

$$Q = \varepsilon\, C_{min}\, (T_{h,1} - T_{c,1}) \tag{30}$$

The geometrical and operating parameters have been varied within range to study the effect on pressure drop, effectiveness and heat transfer. The all thermo- physical properties like (specific heat, density, dynamic viscosity and thermal conductivity) and prandtl number remains constant not changing with the temperature. The calculations involved in the simulation of plate heat exchanger have been shown in the flow chart

3. Results and Discussions

The performance of plate heat exchanger has been analyzed by considering geometrical and operating parameters. In this study the performance of plate heat has been considered in the terms of the effectiveness, heat transfer and pressure drop for hot fluid as well as cold fluid by changing geometrical parameters (vertical port distance, horizontal port distance and port diameter) and operating parameters (mass flow rate of hot fluid and mass flow rate of cold fluid) for plate heat exchanger. The main objective of this study is to study the effect of geometrical and operating parameters on the performance of plate heat exchanger.

3.1 Mass flow rate of hot fluid (m_h)

It is observed that the effectiveness increases continuously with the mass flow rate. Mass flow rate has similar effect on heat transfer rate. The slope of the curve of heat transfer and effectiveness is decreasing. It means effectiveness and heat transfer initially increasing at faster rate and after a point both increase at slower rate. Pressure of hot fluid side is increasing as mass flow rate of hot fluid is increasing.

3.2 Mass flow rate of cold fluid (m_c)

As mass flow rate of cold fluid increase, the effectiveness of plate heat exchanger decreases. The heat transfer increases with mass flow rate of cold fluid. The pressure also increases with the mass flow rate of cold fluid but the increment in the pressure of cold side is more than that of hot side.

3.3 Vertical port distance (L_v)

The vertical port distance is a geometrical parameter. As vertical port distance increases effectiveness of plate heat exchanger also increases. The effect of vertical distance on heat transfer is same. As vertical port distance increases heat transfer also increases. The effect of vertical port distance on pressure can be seen from graph that pressure increases with vertical port distance linearly .

3.4 Horizontal port distance (L_h

It can be observed from the graph that as horizontal port distance increases; effectiveness of the plate heat exchanger also increases. The effect of horizontal port distance on heat transfer can be seen from graph, heat transfer increasing with horizontal distance. Pressure drop and horizontal distance both are opposite in nature. As horizontal port distance increases, the pressure drop decreases.

3.5 Port Diameter (D_p)

The port diameter and effectiveness is conflicting in nature. As port diameter increases, effectiveness of the plate heat exchanger decreases and as port diameter decreases, the effectiveness of the plate heat exchanger increases. The effect of port diameter on heat transfer is similar like effectiveness.

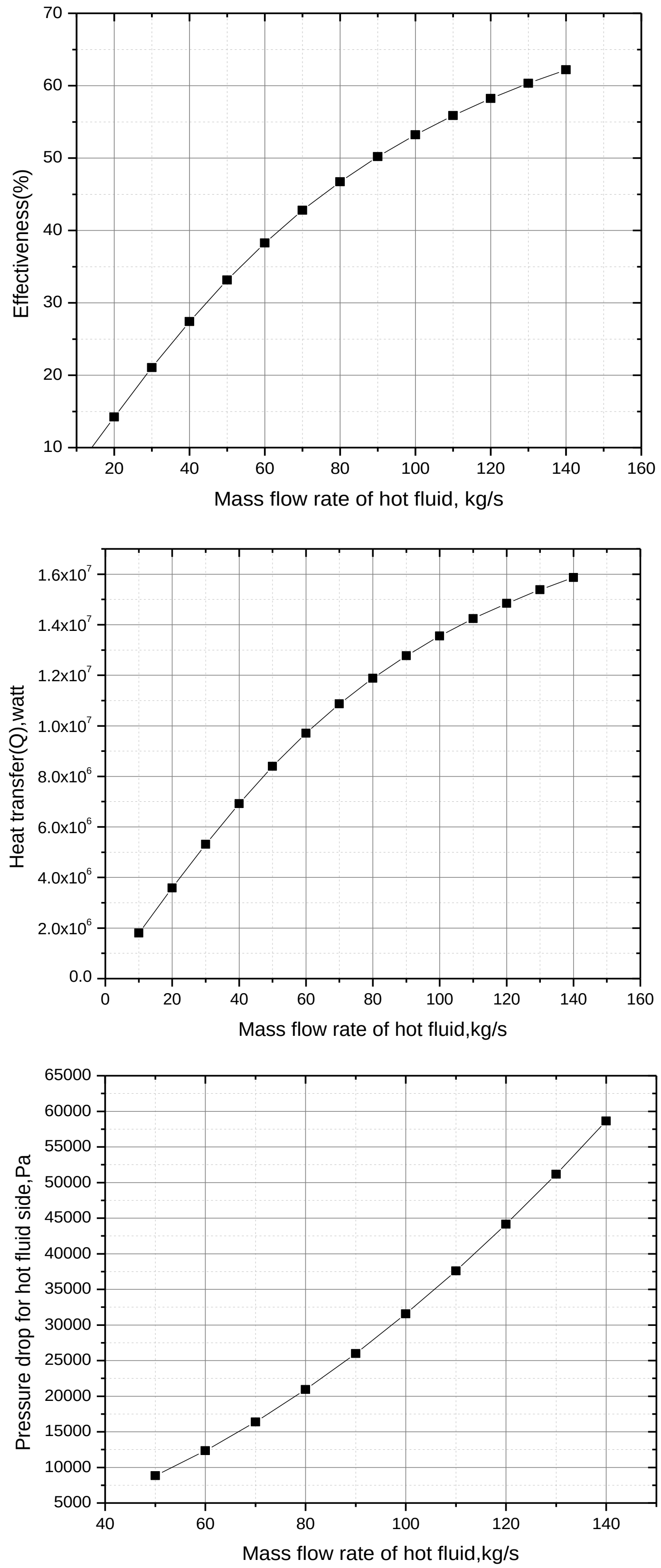

Figure 2: The effect of hot mass flow rate on (a) effectiveness (b) heat transfer (c) pressure drop

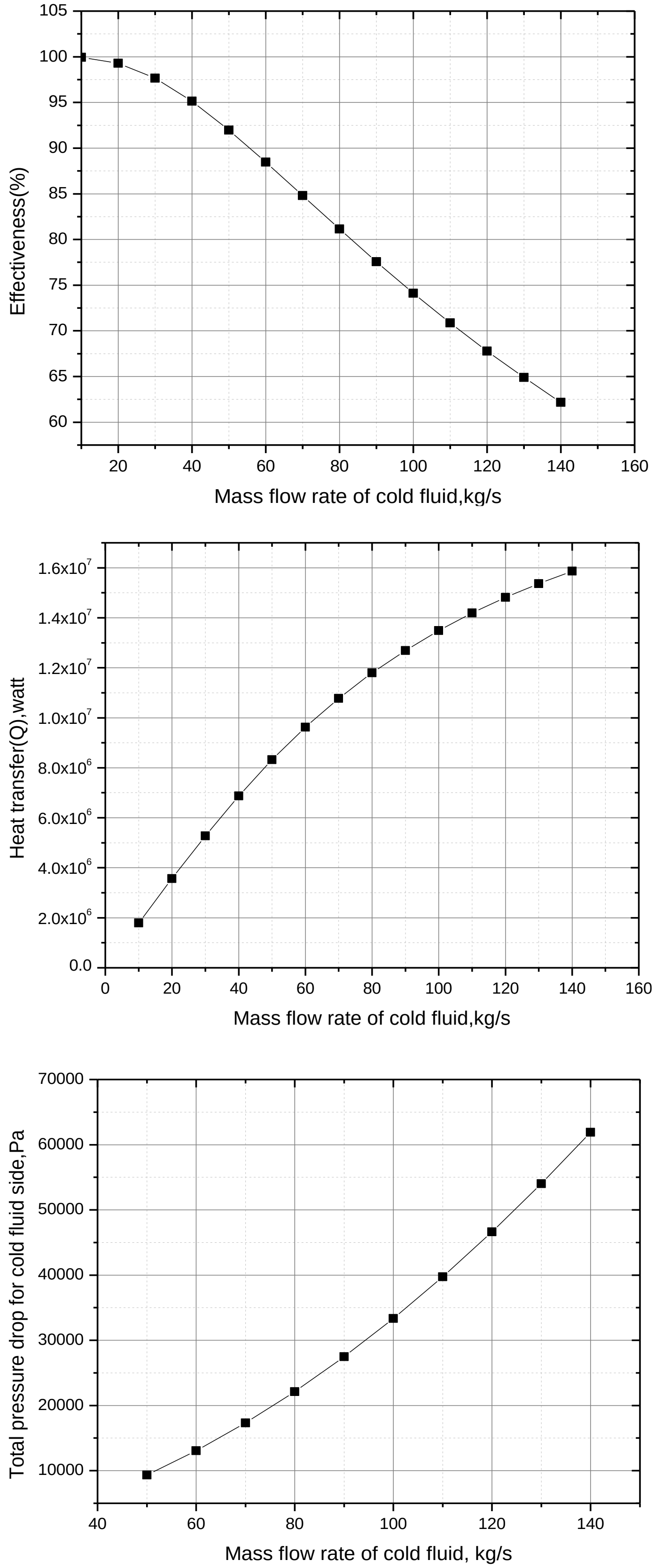

Figure 3: The effect of cold mass flow rate on (a) effectiveness (b) heat transfer (c) pressure drop

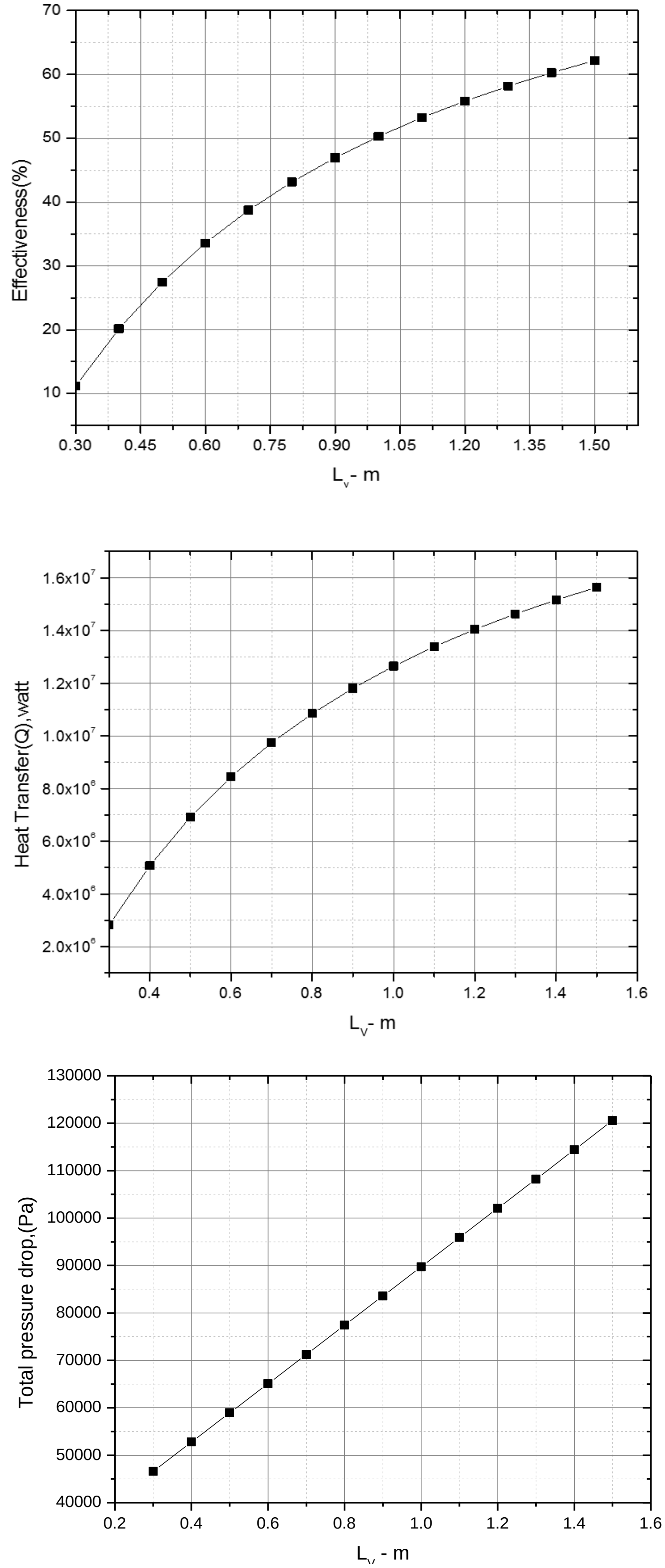

Figure 4: The effect of the vertical port distance on (a) effectiveness (b) heat transfer (c) total pressure drop

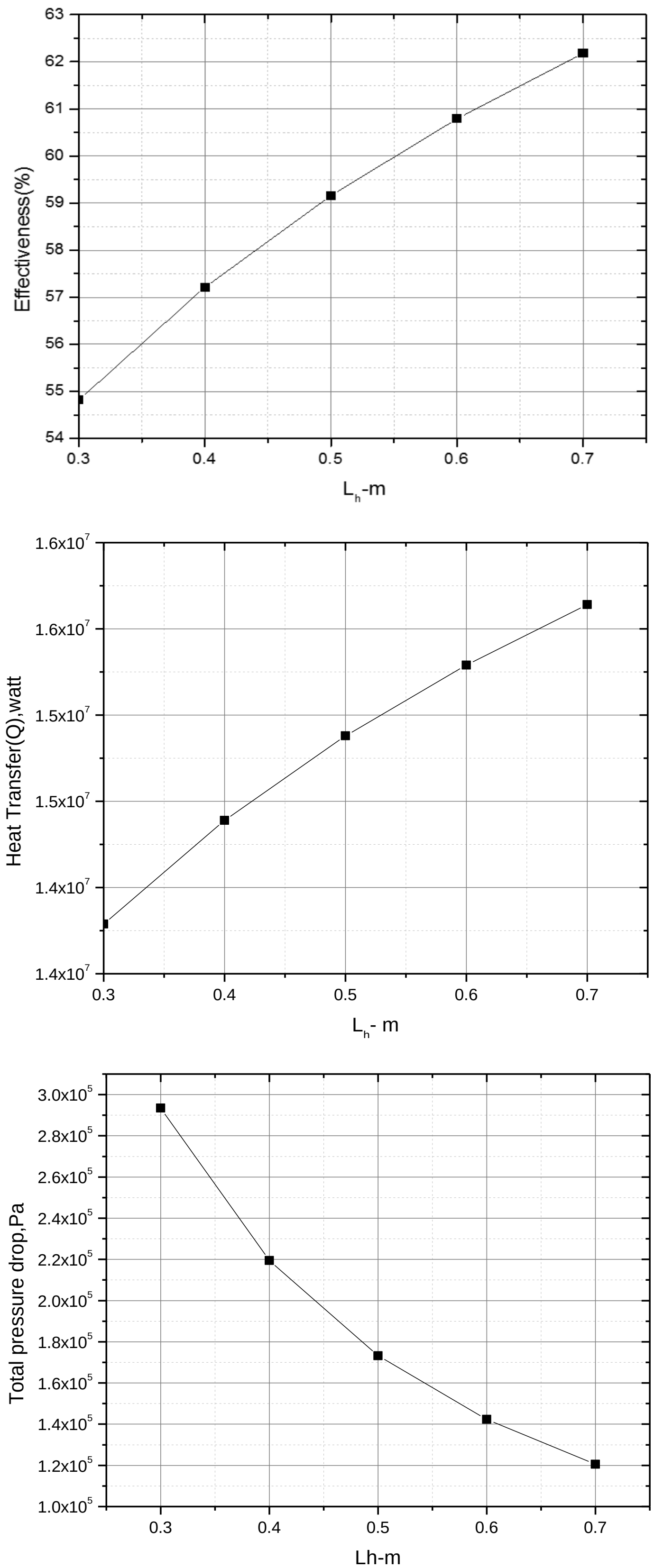

Figure 5: The effect of the horizontal port distance on (a) effectiveness (b) heat transfer (c) total pressure drop

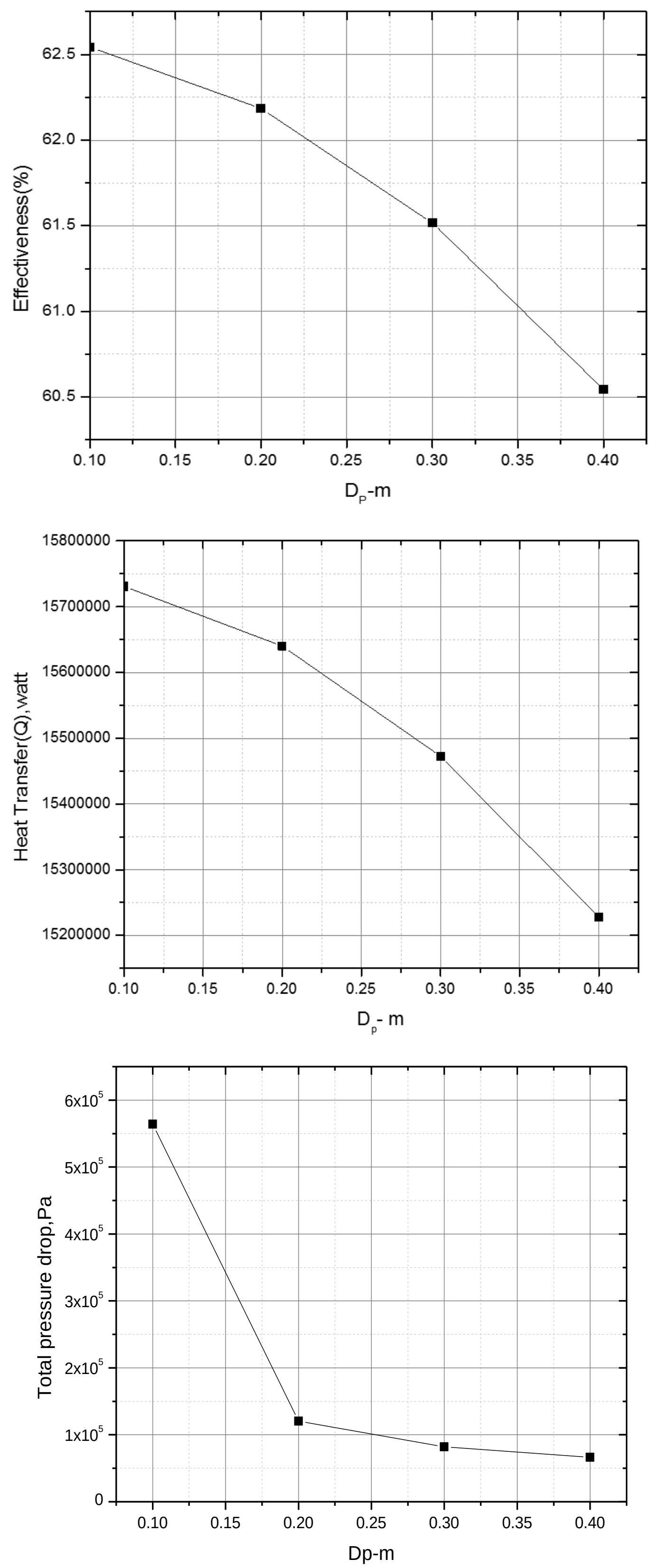

Figure 6: The effect of the vertical port distance on (a) effectiveness (b) heat transfer (c) total pressure drop

3.6 Design optimization using genetic algorithm

In the proposed model, plate heat exchanger has been considered and optimum value of geometrical parameters are obtained by using genetic algorithm. For this purpose, optimization tool box of genetic algorithm in MATLAB is used. For this study, optimization technique based on genetic algorithm has been developed for plate heat exchanger which will maximize the heat transfer and minimize the pressure drop. To achieve this, two objective functions as entropy generation units (N_s) and total annual cost (TAC) have been considered and to get minimum of two objective functions, optimized geometrical parameters have been obtained. For this optimization, geometrical parameters considered are vertical port distance (L_v), horizontal port distance (L_h), port diameter (D_p), thickness of plate (t), gap between two plates (b) and number of plates (N_t). The optimized values of geometrical parameters have been obtained for constrained and unconstrained case

3.6.1 Brief description of solution methodology

A genetic algorithm is a non-traditional optimisation technique which is completely based on the concept of genetics and natural selection. An elaborate description of this technique will be available in the number of references as [12, 13].

The genetic algorithm is initiated with an initial set of the population which are represented by a binary coded string structure from 0's to 1's. The vocabulary used in GA is based on genetics. The value of the objective function for a particular member decides it's merit in comparison with its counterparts. In GA terminology, it is called as the fitness function. A simple GA works with three important operators namely reproduction, crossover and mutation.

Usually, the first operator applied is 'Reproduction' on a population. First of all, reproduction operator has to select some good strings from the population based on their fitness values relative to the fitness of the other members and form a mating pool that's why sometimes this operator is also known as the 'Selection' operator. Strings with high fitness value have the higher chances to being selected. This reproduction operator is an artificial version of natural selection which mimics the Darwinian principal of the survival of the fittest. By exchanging the parts of the strings from the parents, the crossover operator alters the composition of the offspring and creates new strings. With the help of the suitable crossover probability (p_c), better new strings can be created. Different types of crossover techniques are used in the GA toolbox but for the given model, constraint dependent crossover technique has been used. Mutation is a next operator which provides variation to the population by altering a bit of the string from 0 to 1 and vice versa with a small mutation probability (p_m). To maintain diversity in the population and to achieve the local search around the current solution, the mutation operator is used which is also not possible sometimes with crossover and reproduction operator. By altering the string locally, mutation operator tries to create a better string and also to maintain diversity in the population [14].

To illustrate the working principle of GA's, firstly an unconstrained optimisation problem has to be considered and after that it can be converted into the constrained optimisation problem. For the described problem, GA has been used for unconstrained as well as for constrained minimisation objective function. The problem can be stated as

$$Minimise\ f(X)\ , \quad X = x_{i\ ,} \qquad i = 1, 2 \ldots N \tag{31}$$

$$\text{and} \qquad x_{i,min} \leq x_i \leq x_{i,max} \tag{32}$$

The constraints are given as:

$$h_j(X) \leq 0, \qquad j = 1, 2 \ldots M \tag{33}$$

Now the first step is to convert the constraint optimisation problem into an unconstrained optimisation problem by adding a penalty term.

$$Minimise\ f(X) + \sum_{j=1}^{M} \varphi\left(h_j(X)\right), \tag{34}$$

Subjected to:

$$x_{i,min} \leq x_i \leq x_{i,max}, \qquad i = 1, 2 \ldots N \tag{35}$$

Where φ is a penalty function term defined as:

$$\varphi\big(h(X)\big) = R1.\ \langle h(X)\rangle^2 \tag{36}$$

Where $R1$ is the penalty parameter having an arbitrary large value.

The next step is to convert the minimum optimisation problem to a maximisation problem. This is done by changing the objective function such that the optimum point remains unchanged as follows:

$$Maximise\ F(X), \tag{37}$$

Where,

$$F(X) = \frac{1}{1+\left\{f(X)+\sum_{j=1}^{M} \varphi\left(h_j(X)\right)\right\}} \tag{38}$$

The above algorithm will be used for minimising the entropy generation (N_s) and the total annual cost (TAC).

3.7 Thermodynamic Optimization

The thermodynamic optimization has been used to obtain the optimum design of plate heat exchanger. The entropy generation causes the irreversibility in the thermodynamic system. Due to irreversibility, there is loss of useful energy in the thermodynamic system. Energy loss because of irreversibility which can be minimized. The loss of useful energy is increased with entropy generation. In the present study, the entropy generation has been taken as single objective which will be minimized. The GA toolbox has been used to obtain the optimized value of six decision variables. The entropy generation unit has been optimized for the two cases:

- For unconstrained case
- For heat transfer constrained case

In this study, for two cases mentioned above the optimum value of geometrical parameters (vertical port distance (L_v), horizontal port distance (L_h), port diameter (D_p), thickness of plate (t), gap between two plates (b) and number of plates (N_t)) have been obtained and compared.

The rate of entropy generation for both fluids is:

$$\dot{S} = m_h(\Delta S_h) + m_c(\Delta S_c) \tag{39}$$

The above expression can be expressed in terms of temperature and pressure:

$$\Delta S_H = m_h c_{p,h} \ln\frac{T_{h,2}}{T_{h,1}} + m_c C_{p,c} \ln\frac{T_{c,2}}{T_{c,1}} \tag{40}$$

Now, the effectiveness of the exchanger is

$$\varepsilon = \frac{c_h(T_{h,1}-T_{h,2})}{C_{min}(T_{h.1}-T_{c,1})} = \frac{c_c(T_{c,2}-T_{c,1})}{c_{min}(T_{h,1}-T_{c,1})} \tag{41}$$

So the outlet temperature of hot fluid and cold fluid

$$T_{h,2} = T_{h,1} - \varepsilon\frac{c_{min}(T_{h,1}-T_{c,1})}{c_h} \tag{42}$$

$$T_{c,2} = T_{c,1} - \varepsilon\frac{c_{min}(T_{h,1}-T_{c,1})}{c_c} \tag{43}$$

Entropy change due to heat transfer between two fluids is given by [14]

$$\Delta S_H = c_h \ln\left\{1 - \varepsilon\frac{c_{min}}{c_h}\left(1 - \frac{T_{c,1}}{T_{h,1}}\right)\right\} + C_c \ln\left\{1 + \varepsilon\frac{c_{min}}{c_c}\left(\frac{T_{h,1}}{T_{c,1}} - 1\right)\right\} \tag{44}$$

In the similar fashion the entropy change due to friction pressure drop is given by 14]

$$\Delta S_P = m_h\frac{\Delta P_h}{\rho_{h,1}T_{h,1}} + m_c\frac{\Delta P_c}{\rho_{c,1}T_{c,1}} \tag{45}$$

The number of entropy generation units may be defined as follows:

$$N_S = \frac{\dot{S}}{C_{max}} \tag{46}$$

$$N_S = c_h ln\left\{1 - \varepsilon\frac{c_{min}}{c_h}\left(1 - \frac{T_{c,1}}{T_{h,1}}\right)\right\} + C_c \ln\left\{1 + \varepsilon\frac{c_{min}}{c_c}\left(\frac{T_{h,1}}{T_{c,1}} - 1\right)\right\} + \left\{m_h\frac{\Delta P_h}{\rho_{h,1}T_{h,1}} + m_c\frac{\Delta P_c}{\rho_{c,1}T_{c,1}}\right\} \tag{47}$$

$$NTU = \frac{A\,U}{Cmin} \tag{48}$$

The effectiveness of plate heat exchanger [2]

$$\varepsilon = \frac{[1-\exp\{-NTU(1-Cr\}]}{[1+Cr\exp\{-NTU(1-Cr)\}]} \tag{49}$$

As L_w is effective width of plate, b is channel gap, N_c is number of channels. The total free flow area of plate heat exchanger may be defined as:

$$A_{ff} = L_w\, b\, N_c \tag{50}$$

Total free flow area for hot fluid and cold fluid can be obtained as:

$$A_{ffh} = L_W\, b\, N_{ch} \tag{51}$$

$$A_{ffc} = L_w\, b\, N_{cc} \tag{52}$$

Total heat transfer area for both fluid streams can be obtained [2]

$$A_{hh} = \left\{\left(L_{vh} - D_{ph}\right).\left(L_{hh} + D_{ph} + 0.015\right)\right\}(N_t - 2) \tag{53}$$

$$A_{hc} = \left\{\left(L_{vc} - D_{pc}\right).\left(L_{hc} + D_{pc} + 0.015\right)\right\}(N_t - 2) \tag{54}$$

Finally, the rate of heat transfer is defined as:

$$Q = \varepsilon C_{min}(T_{h,1} - T_{c,1}) \tag{55}$$

The pressure drop for both fluids can be defined as:

$$\Delta P_h = 4f_h\frac{L_{vh}N_{ph}}{D_h}\cdot\frac{{G_{ch}}^2}{2\rho_h} + 1.4N_{ph}\frac{{G_{ph}}^2}{2\rho_h} \tag{56}$$

$$\Delta P_c = 4f_c\frac{L_{vc}N_{pc}}{D_h}\cdot\frac{{G_{cc}}^2}{2\rho_c} + 1.4N_{pc}\frac{{G_{pc}}^2}{2\rho_c} \tag{57}$$

Then, the total pressure drop (ΔP) may be calculated as:

$$\Delta P = \Delta P_h + \Delta P_c \tag{58}$$

$$\Delta P = \left[4f_h\frac{L_{vh}N_{ph}}{D_h}\cdot\frac{{G_{ch}}^2}{2\rho_h} + 1.4N_{ph}\frac{{G_{ph}}^2}{2\rho_h} + 4f_c\frac{L_{vc}N_{pc}}{D_{hc}}\cdot\frac{{G_{cc}}^2}{2\rho_c} + 1.4N_{pc}\frac{{G_{pc}}^2}{2\rho_c}\right] \tag{59}$$

Now, statement of objective function of the optimization problem for entropy generation defined as above is as follows:

$$\text{Minimise } f(X) = N_s \tag{60}$$

Subjected to the following constraints:

$$h_1(X) \Rightarrow 1.1 \le L_v \ge 1.5$$
$$h_2(X) \Rightarrow 0.1 \le D_p \ge 0.4$$
$$h_3(X) \Rightarrow 0.0025 \le b \ge .005$$
$$h_4(X) \Rightarrow 0.3 \le L_h \ge 0.7$$
$$h_5(X) \Rightarrow 0.0005 \le t \ge 0.0012$$
$$h_6(X) \Rightarrow 90 \le N_t \ge 105$$
$$h_7(X) \Rightarrow \xi(X) - Q = 0 \tag{61}$$

$h_7(X)$ is heat transfer equality constraint which can be obtained from the equation (55). In the equation (61), the term $\xi(X)$ represents the LHS of the equation (55) and Q represents the minimum heat transfer requirement for plate heat exchanger which is considered as 11712 kW for the particular case.

The objective of present study is to obtain the minimum value of entropy generation units with heat transfer constraint (11712 kW) and to obtain optimized value of geometrical parameters (vertical port distance (L_v), horizontal port distance (L_h), port diameter (D_p), thickness of plate (t), gap between two plates (b) and number of plates (N_t)) within given range. The operating parameters as shown in table 1.

Table 1: Different operating parameters

Parameters	Fluid (a)	Fluid (b)
Mass flow rate, m (Kg s^{-1})	140	140
Inlet temperature,T_1 (K)	338	295
Specific heat, C_p(J kg^{-1}K^{-1})	4183	4178
Density,ρ (kg m^{-3})	985	995
Dynamic viscosity,μ (N s m^{-2})	5.09$\times 10^{-4}$	7.6$\times 10^{-4}$
Prandtl number, Pr	3.31	5.19
Thermal conductivity, k (Wm^{-1}K-1)	.645	.617

3.8 Optimization based on total annual cost (TAC)

In this section, the optimized value of geometrical parameters for plate heat exchanger based on minimum total annual cost has been obtained through GA for the two cases which are given below

- For unconstrained case
- For heat transfer constrained case

The total cost of plate heat exchanger may vary according to applications. The total annual cost of plate heat exchanger consists of initial cost and operating cost. Initial cost includes cost of heat exchanger core, cost of pump used for hot fluid and cost of pump used for cold fluid. Operating cost includes operating cost of two pumps. The initial cost is known as purchase cost of plate heat exchanger. The initial and operating costs both contain fixed and variable components. The variable components of heat exchanger core will depend on heat transfer area and variable components of pump will depend on product of heat capacity and pressure drop. The power consumption is function of operating cost. The expression of TAC has been obtained from [12] TAC can be defined as

Total Annual Cost (TAC) = Initial cost of the (heat exchanger core + pump for hot gas + pump for cold air) + Operating cost of (pump for hot gas + pump for cold air)

$$TAC = Af.\left[\{Ca + Cb.A^r\} + \left\{Cc + Cd.\left(\frac{m_h}{\rho_h}.\Delta P_h\right)^s\right\} + \left\{Cc + Cd.\left(\frac{m_c}{\rho_c}.\Delta P_c\right)^s\right\}\right] + \frac{C_{pow}(time/year)}{\eta_{pump}}\left[\frac{m_h}{\rho_h}.\Delta P_h + \frac{m_c}{\rho_c}.\Delta P_c\right] \quad (62)$$

Where ρ_h and ρ_c are densities of hot and cold fluid. ΔP_h and ΔP_c are pressure drops for hot and cold fluid respectively in kPa.

For this present example, following are the values of the cost factors [12] and other parameters:

Af = 0.322, Ca = 30000, Cb = 750, Cc = 2000, Cd = 5, C_{pow} = 0.1 \$/kW-hr, r = 0.8, s = 0.68, η_{pump} = 0.7

Total operation time/year = 8760 hours, specific heat transferred (Q) = 11712 kW

Now, the statement of optimization problem for TAC defined as above is as follow:

$$\text{Minimise } f(X) = TAC \quad (63)$$

Subjected to the following constraints:

$$\begin{aligned} &h_1(X) \Rightarrow 1.1 \le L_v \ge 1.5 \\ &h_2(X) \Rightarrow 0.1 \le D_p \ge 0.4 \\ &h_3(X) \Rightarrow 0.0025 \le b \ge .005 \\ &h_4(X) \Rightarrow 0.3 \le L_h \ge 0.7 \\ &h_5(X) \Rightarrow 0.0005 \le t \ge 0.0012 \\ &h_6(X) \Rightarrow 90 \le N_t \ge 105 \\ &h_7(X) \Rightarrow \xi(X) - Q = 0 \end{aligned} \quad (64)$$

$h_7(X)$ is heat transfer equality constraint which can be obtained from the equation (55). In the equation (64), the term $\xi(X)$ represents the LHS of the equation (55) and Q represents the minimum heat transfer requirement for plate heat exchanger which is considered as 11712 kW for the particular case.

3.9 Multi objective optimization based on entropy generation units (N_s) and total annual cost (TAC)

GA optimization toolbox in MATLAB has 'gamultiobj' option which has been used for multi objective optimization. For multi objective optimization, two objective functions are considered which are entropy generation (Ns) and total annual cost (TAC). The objective of this study is to find optimized the parameters on which entropy generation unit and total annual cost are minimum. In this section, multi objective optimization has been carried out for two cases which are given below

- For unconstrained case
- For heat transfer constrained case

Now, the statements of optimization problem for both entropy generation unit (N_s) and total annual cost (TAC) defined are as follow:

$$Minimise\ f(X) = N_s \quad (65)$$

$$Minimise\ g(X) = TAC \quad (66)$$

Subjected to the following constraints:

$$h_1(X) \Rightarrow 1.1 \le L_v \ge 1.5$$

$$h_2(X) \Rightarrow 0.1 \le D_p \ge 0.4$$
$$h_3(X) \Rightarrow 0.0025 \le b \ge .005$$
$$h_4(X) \Rightarrow 0.3 \le L_h \ge 0.7$$
$$h_5(X) \Rightarrow 0.0005 \le t \ge 0.0012$$
$$h_6(X) \Rightarrow 90 \le N_t \ge 105$$
$$h_8(X) \Rightarrow \xi(X) - Q = 0 \quad (67)$$

$h_7(X)$ is heat transfer equality constraint which can be obtained from the equation (55). In the equation (67), the term $\xi(X)$ represents the LHS of the equation (55) and Q represents the minimum heat transfer requirement for plate heat exchanger which is considered as11712 kW for the particular case.

4. Discussions

The optimization of plate heat exchanger has been done by using GA for minimum value of entropy generation unit (N_s)and total annual cost (TAC). The optimized geometrical parameters have been obtained and compared at different constraint conditions. The optimization of plate heat exchanger has been done for single objective function as well as multi objective functions. The results have been given into three sections.

4.1 Thermodynamic optimization

For the thermodynamic optimization, entropy generation has been taken as single objective with six geometrical parameters which are (vertical port distance (L_v), horizontal port distance (L_h), port diameter (D_p), thickness of plate (t), gap between two plates (b) and number of plates (N_t)).

The optimized parameters have significant effect of GA parameters, the results may be changed by varying the population size, crossover probability (p_c) and mutation probability (p_m). The variation of entropy generation unit (N_s) with population size, crossover probability and mutation probability has been shown by the Fig 7.

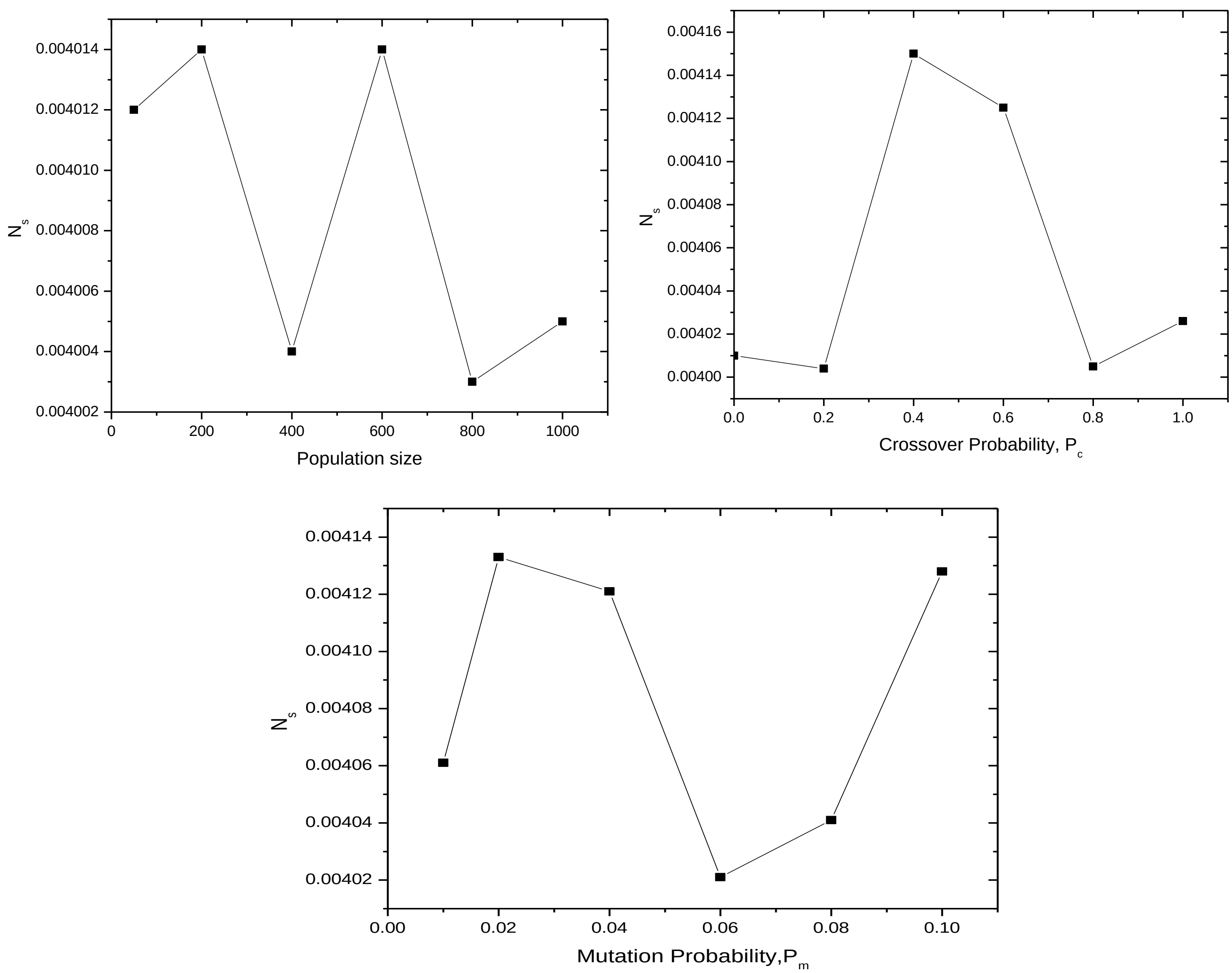

Figure 7: Effect of GA parameters (a) population size (b) crossover probability (c) mutation probability on the entropy generation unit

For unconstrained problem, no constraint has been considered. The GA parameters selected for this study are given as population size=800, crossover probability=0.8, elite count= 0.05*pop. size, crossover technique=constraint dependent, mutation technique= adaptive feasible and for nonlinear constraint algorithm operator, 'Augmented lagrangian' option has been selected.

For the constrained optimization, a heat transfer constraint has been considered as 11712 kW. The GA parameters are same as taken in unconstrained problem.

The optimum solutions by using GA parameters have been obtained for unconstrained and heat transfer constrained cases after a number of iteration of the optimization. The optimization results have been listed in the table 2.

Table 2: Optimum solutions for unconstraint and constraint

Design parameters	Unconstraint	Heat transfer constraint
L_v (m)	1.5	1.1
D_p (m)	0.2	0.1
b (m)	0.003	0.005
L_h (m)	0.7	0.3
t(m)	0.001	0.001
N_t (m)	105	90
N_s	0.0041	0.0044

It can be seen from the table 2 that entropy generation unit for unconstraint is less than the entropy generation unit for heat constraint. The increment in the entropy generation unit for constraint case is because of heat transfer constraint.

4.2 TAC optimization

For the thermodynamic optimization, total annual cost has been taken as single objective with six geometrical parameters which are (vertical port distance (L_v), horizontal port distance (L_h), port diameter (D_p), thickness of plate (t), gap between two plates (b) and number of plates (N_t)). The results of optimization have been obtained for two conditions which are given below

- For unconstrained case
- For heat transfer constrained case

The optimised parameters have significant effect of GA parameters, the results may be changed by varying the population size, crossover probability (p_c) and mutation probability (p_m). The variation of total annual cost (TAC) with population size, crossover probability and mutation probability has been shown by the Fig 8.

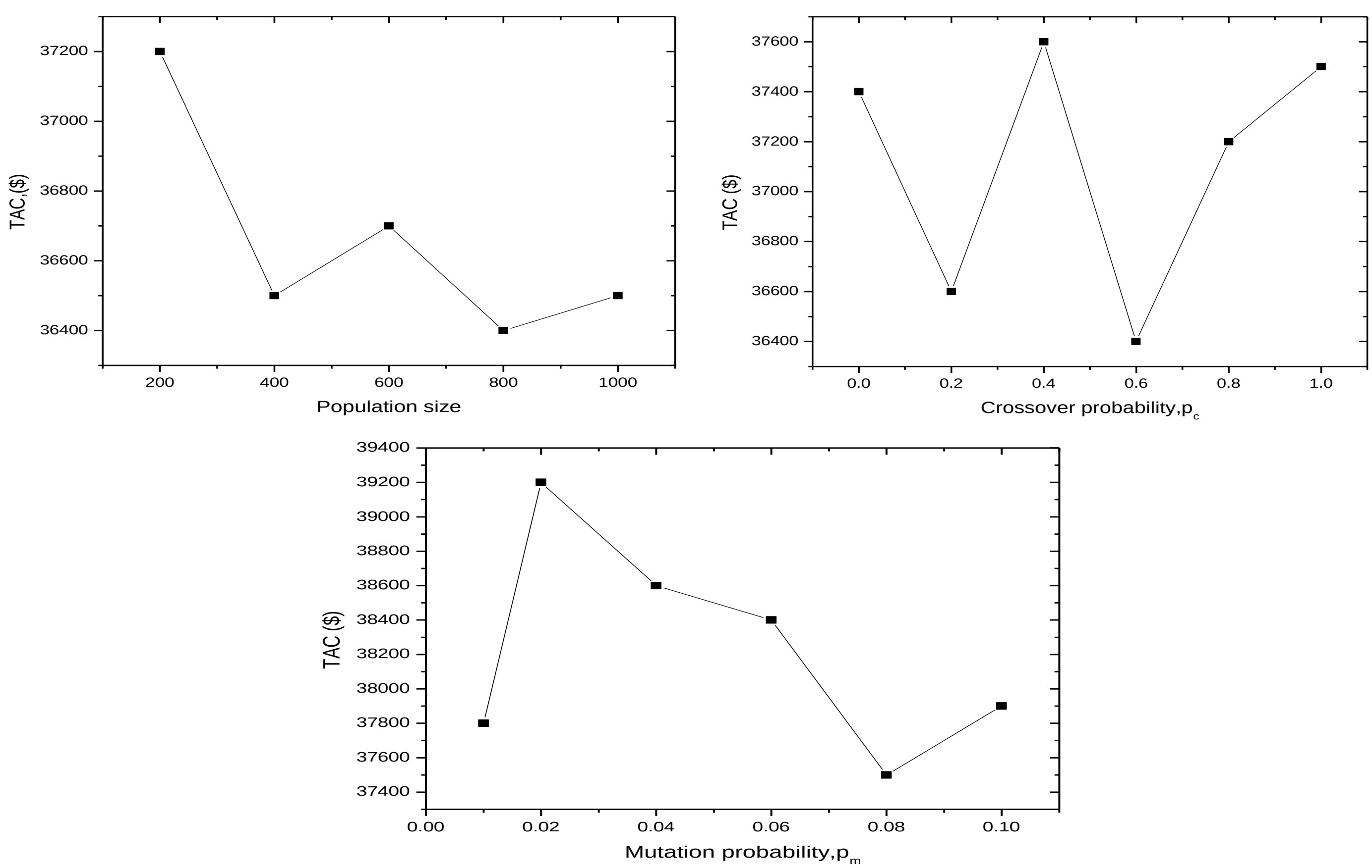

Figure 8: The effect of GA parameters (a) population size (b) crossover probability (c) mutation probability on the total annual cost

For unconstrained problem, no constraint has been considered. The GA parameters selected for this study are given as population size=800, crossover probability=0.6, elite count= 0.05*pop. size, crossover technique= constraint dependent, mutation technique= adaptive feasible and for nonlinear constraint algorithm operator, 'Augmented lagrangian' option has been selected.
For the constrained optimization, a heat transfer constraint has been considered as11712 kW. The GA parameters are same as taken in unconstrained problem.
The optimum solutions by using GA parameters have been obtained for unconstrained and heat transfer constrained cases after a number of iteration of the optimization. The optimization results have been listed in the table 3.

Table 3 optimum solutions for unconstraint and heat transfer constraint

Design parameters	Unconstraint	Heat transfer constraint
L_v (m)	1.1	1.17
D_p (m)	0.2	0.178
b (m)	0.005	0.004
L_h (m)	0.7	0.345
N_t	90	102
TAC ($)	40.719	53,625

It can be seen from the table 2 that the results for TAC optimization are different from the last case. For unconstraint case TAC is minimum as 40,719 $ and after adding heat transfer constraint the TAC of plate heat exchanger increases as 53,625$

4.3 Multi-objective optimization for Ns and TAC

In this section, multi-objective has been done for two objective functions which are entropy generation unit (N_s) and total annual cost (TAC) with six geometrical parameters (vertical port distance (L_v), horizontal port distance (L_h), port diameter (D_p), thickness of plate (t), gap between two plates (b) and number of plates (N_t)). For the multi-objective optimization, the selected GA parameters are population size= 200 crossover probability=0.8 and mutation probability = 0.08 for unconstrained and constrained.
The results of multi-objective optimization have been obtained in the 70 set of solutions which is known as pareto front for unconstrained as well as constrained optimization problem. The results of multi-objective show the conflict between two objective functions, the entropy generation unit (N_s) and total annual cost (TAC). As entropy generation unit increases, total annual cost decreases and as entropy generation decreases, total annual cost increases.
The relationship between entropy generation and total annual cost is same for unconstraint and constraint case. The is only difference in the range of two objective functions. For unconstraint case, entropy generation unit varies from 0.00405 to 0.00445 and total annual cost varies from 34,000 to 40,000 $. For the heat transfer constraint, the entropy generation varies from 0.00415 to .00450 and total annual cost varies from 40,000 to 50,000 $.

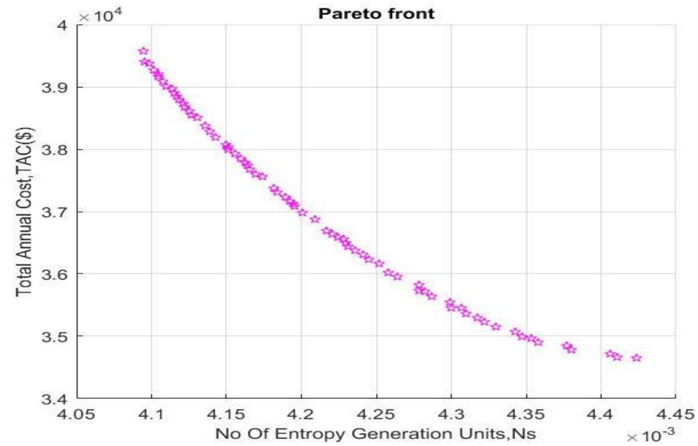

Figure 9: Pareto front for unconstrained case

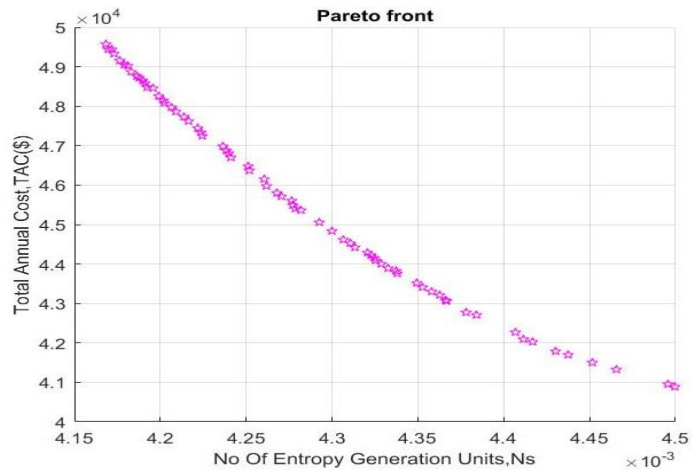

Figure 10: Pareto front for heat transfer constrained case

5. Conclusions

The present work shows the numerical simulation and thermal design optimization of plate heat exchanger based on the entropy generation unit and total annual cost. The thermo-hydraulic performance of plate heat exchanger has been studied. For the single objective and multi- objective functions, plate heat exchanger has been optimized with six geometrical parameters by using genetic algorithm.
The performance of plate heat exchanger has been obtained in terms of the effectiveness, heat transfer and pressure drop for hot fluid as well as for cold fluid. The geometrical (vertical port distance, horizontal port distance and port diameter) and operating parameters (mass flow rate of hot fluid and mass flow rate of cold fluid) have been used to study the effect on the performance of plate heat exchanger. Further, the optimization of plate heat exchanger has been presented using GA. The plate heat exchanger has been optimized for two objective functions which are entropy generation unit and total annual cost for single objective as well as for multi-objective functions. The six geometrical parameters have been optimized and compared for unconstrained and constrained conditions. The results obtained from multi-objective optimization reveal conflict between two objective functions,

the entropy generation unit and the total annual cost.
The analysis may be used by the designers to design the plate heat exchanger with optimum design variable for low entropy generation unit and low cost. In the future, work of the optimization can be extended for other geometrical parameters as well as objective functions. Further, the effect of the other geometrical and operating parameters on the effectiveness, heat transfer and pressure drop for both fluids can also be analyzed which will help the new researchers/designers for designing better the plate heat exchanger. The simulation results will also help the designers to design and control such plate heat exchangers.

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