A robust approach for optimal design of plate fin heat exchangers using biogeography based optimization (BBO) algorithm

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A robust approach for optimal design of plate fin heat exchangers using biogeography based optimization (BBO) algorithm

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HIGHLIGHTS

- The first use of a BBO algorithm for optimization of plate-fin heat exchangers.
- Total cost, pressure drop and the heat transfer area of exchanger minimized by BBO.
- A quick method proposed to optimal design of heat exchangers with low run time.
- All of available and possible constraints and restriction is handled.

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ABSTRACT

Design of plate-fin heat exchangers is a very complex task generally based on trial and error process. Traditional designing methods are very time consuming and do not guarantee the archive of an optimal solution; therefore heuristic based computation methods are used, usually. So, in present paper a new design method proposed for optimization of plate fin heat exchangers using biogeography-based optimization (BBO) algorithm. The BBO algorithm has some advantages in detecting the global minimum compared with other heuristic algorithms. In present research the BBO scheme has been employed for optimal design of the plate fin heat exchanger by minimization of the total annual cost, heat transfer area and total pressure drops of the equipment considering main structural and geometrical parameters of the exchanger as design variables. Based on proposed approach, a full computer code was developed and three various case studies are investigated by it to illustrate the effectiveness and accuracy of the proposed method. Comparison of the results with those obtained by previous methods reveals that the BBO algorithm can be successfully employed for optimization of plate fin heat exchangers. Finally, parametric analysis carried out to evaluate the sensitivity of the proposed method to the cost and structural parameters.

1. Introduction

One of the most important types of compact heat exchangers is a plate fin heat exchanger which has widespread engineering applications [1,2]. In a plate fin heat exchanger, hot and cold fluids flow through plates of the exchanger known as parting sheets and fins. A plate fin exchanger is shown in Fig. 1, schematically [3].

Designing procedure of plate fin exchangers comprises of thermodynamic and fluid dynamic design, geometry and structure design, cost calculation and optimization. This procedure reveals a complex process. In practice, the design of the exchanger is a complex and heavy trial and error procedure; hence, there is always the possibility that the designed results are not the optimum. Consequently, researchers attempt to optimize thermal equipment and systems using heuristic based optimization algorithms. Accordingly, many interesting studies have been conducted using artificial intelligence methods, recently. Zhou et al. [4] used a multi-level, multi-factor and non-structural fuzzy optimum decision model in the optimal selection of compact heat exchangers. They considered the performance of two different plate fin heat exchangers made of stainless steel and PTFE composite. They concluded that the plate-fin heat exchanger made of PTFE composite is feasible and optimal to be used as an acid solution cooler. Other studies have been conducted on optimization of compact or other types of heat exchangers as a part of other industrial equipment [4–16]. Lee et al. [5] have developed a novel heat exchanger with new geometries for application in the low temperature lift heat pump (LTLHP). Their main design strategy were regulating the flow area ratio and offsetting plates in order to balance the heat transfer

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Nomenclature

A, A_{HT}	heat transfer area (m^2)
A_f	annual coefficient factor
A_{ff}	free flow area (m^2)
C	heat capacity (W/K)
C_A	cost per unit area ($/$m^2$)
C_{in}	investment cost ($/year)
C_{op}	operating cost ($/year)
C_r	C_{min}/C_{max}
D_h	hydraulic diameter (m)
f	friction factor
$f(x)$	objective function (pressure loss)
G	mass flow velocity (kg/m^2 s)
h	convective heat transfer coefficient (W/m^2 K)
H	height of fin (m)
j	Colburn factor
k_{el}	electricity price ($/MW h)
l_f	fin offset length (m)
L	heat exchanger length (m)
m	mass flow rate (kg/s)
n	fin frequency (fins per meter)
n_1	numerical constant
N_a, N_b	fin layers number for fluid a and b
NTU	number of transfer units
P	pressure
Pr	Prandtl number
Q	heat transfer rate (W)
r	interest rate
Re	Reynolds number
t	thickness of fin (m)
T	temperature (°C)
TAC	total annual cost ($/year)
U	overall heat transfer coefficient (W/m^2 K)
y	depreciation time

Greek symbols

ε	effectiveness
η	efficiency of the pump or fan
μ	dynamic viscosity (Pa s)
ρ	density (kg/m^3)
τ	hours of operation
ΔP	pressure drop (Pa)

Subscripts

a, b	fluid a and b
max	maximum
min	minimum

and pressure drop of the heat exchanger. Nagarajan et al. [12] proposed a fin named rip saw fin and applied it to a high temperature ceramic plate-fin heat exchanger. They optimized the new designed exchanger using CFD analysis. They showed in their research that application of the designed fins in the heat exchanger enhances the performance of the heat exchanger. Bayer et al. [13] have used a mathematical optimization approach for optimal design of borehole heat exchangers. They concluded that the benefit from mathematical optimization increases with heat extraction/injection imbalance. Varun and Siddhartha [14] have optimized a flat plate solar air heater using a genetic algorithm. They considered the thermal performance of the exchanger and optimized the exchanger considering the different system and operating parameters to obtain maximum thermal performance. Bellis and Catalano [15] have used a CFD approach and two evolutionary algorithms for optimization of an immersed particle heat exchanger. They considered geometric parameters of the heat exchanger such as diameters, angles of inlet and outlet pipes and particle injection mode as optimization variables. The objective of their research was to maximize the heat exchanger efficiency by maximizing the dispersion of the particles falling in countercurrent within the flow. They demonstrated that this type of the exchangers has been proposed recently and further attempts are needed to optimal design of these exchangers. There is a need for use stronger optimization algorithms for optimal design of this type of exchangers. Luo et al. [16] considered design and optimization of cross flow fin-tube type internally-cooled dehumidifiers. They considered the geometry parameters and operating conditions of the dehumidifier as optimization variables. They calculated optimum length of the air flow direction. They have not used an intelligence based method for optimization of the exchanger so their optimization procedure involved only with one design parameter; where by using intelligence based optimization algorithms, various and numerous parameters of the exchanger can be used as optimization variables and consequently reach to the global optimal design.

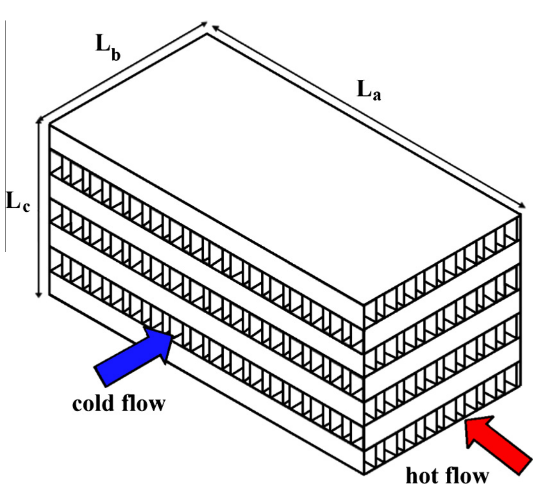

Fig. 1. Diagram of a typical plate fin heat exchanger [3].

Several studies have been conducted on optimization of the plate fin exchanger using genetic algorithms (GA) [3,17–20]. The research of Mishra et al. [17] is an example of application of GA for optimization of PFHEs. They optimized plate fin heat exchangers using a genetic algorithm considering the given heat duty and flow restrictions as optimization constraints. Zhao and Le [21] proposed an effective layer pattern optimization model for multi-stream plate-fin heat exchanger using a genetic algorithm. These researchers indicated that multi-stream flow and heat transfer in one plate fin heat exchanger are used in petroleum, chemical, air separation and other industrial systems; so optimization of this type of plate fin exchangers considered in their research. Also Yujie and Yanzhong [22] studied the heat transfer behavior and optimization of multi-stream plate-fin heat exchangers. Zhe and Yanzhong [23] conducted an experimental investigation on the thermal performance of multi-stream plate-fin heat exchanger based on genetic algorithm layer pattern design. They presented few layer pattern criterion models to determine an optimal stacking pattern. They developed mentioned model by employing a genetic algorithm with binary chromosome ring representing alternatively placed hot and cold layer fluid streams. These researchers have concluded that the performance of plate fin heat exchangers in relation to heat transfer and fluid flow was effectively improved by the optimal design of the genetic algorithm layer pattern. The particle swarm optimization (PSO) algorithm was used by Peng et al. [24] for optimization of a PFHE. They minimized total cost and weight of the exchanger under specified constraints. Their results showed that PSO has better results and shorter computational time in comparison to the genetic algorithm. As another example of using the PSO method for optimization of plate fin exchangers, the work of Rao and Patel [25] can be mentioned. They minimized the entropy generation, volume and annual costs of a plate fin exchanger using PSO method. The results of these researchers also indicate that the PSO algorithm is superior to the traditional genetic algorithm. They showed that by using the PSO method, number of entropy generation units in the considered plate fin heat exchanger is reduced considerably compared with the results of genetic algorithm. Other researchers have used new and various evolutionary algorithms in the optimization of plate fin exchangers and illustrated the need for seeking the new design methods of these equipment [26–30]. In these studies, different fitness functions such as minimization of pressure loss, weight, costs, heat transfer area considered with various constraints and restrictions. The results have shown the ability of these algorithms in optimal design and optimization of plate fin exchangers. Nevertheless, because of the continuous development and advances in evolutionary and metaheuristic computational methods, more studies on the application of powerful and recently proposed new schemes in optimal design of the plate fin exchanger are needed. Biogeography-based optimization (BBO) algorithm is one such method and the same is demonstrated in this study for its effectiveness.

The BBO algorithm is a powerful and novel optimization approach proposed by Simon [31], recently. Hadidi and Nazari [32] used this algorithm for optimization of shell and tube heat exchangers, recently. These researchers studied three different case studies and minimized total cost of the exchanger by varying geometrical and hydrodynamic parameters of the exchanger. They reported that the results of optimization of shell and tube exchangers using the BBO method are superior to preliminary design results and the ones obtained using other available intelligence based methods. Their work is the first use of the BBO algorithm in the optimization of thermal and energy systems so far. They demonstrated that the BBO algorithm shows better results in comparison to GA, PSO, ABC and traditional design approaches for optimal design of shell and tube exchangers. They also showed that the run time of the BBO algorithm is much smaller than other algorithms used previously for optimization of these heat exchangers. Consequently, they demonstrated that the BBO method can be successfully used for optimization of other types of heat exchangers. In the current study the BBO algorithm is employed for optimization of plate fin exchangers. Present research is the first use of this method in optimal plate fin heat exchanger design problem.

Biogeography is the knowledge of the geographical dispersion of biological species. In the BBO scheme, an index is allocated to the geographic zones named as a habitat suitability index (*HSI*). Zones with good conditions of residences have a high *HSI*. Habitability of the habitats and areas is specified by a variable called suitability index variable (*SIV*).

The number of species in low *HSI* habitats is less than high *HSI* habitats; nevertheless the tendency of species of those habitats to emigrate is greater than high *HSI* habitats. The immigration rate in high *HSI* habitats also is low due to saturation of these habitats. Accordingly, the distribution of species in high *HSI* habitats is more static in comparison to habitats with low *HSI* [33].

As mentioned, the strategy of the nature for the distribution of organisms is termed as biogeography. In this manner, it is similar to problem solutions. Considering a problem with some possible solutions which there is a measurable criterion of the suitability of a certain solution, a good solution is similar to a habitat with a high *HSI*, and a poor solution demonstrates a habitat with a low *HSI*. Resistance of good solutions (high *HSI* habitats) to change is greater than poor solutions. In this regard, solutions with high *HSI* tend to induce their features to low *HSI* solutions where poor solutions (low *HSI* habitats) confirm lots of these new features [33]. This novel scheme for solving the problem is named as biogeography-based optimization (BBO) method [32]. Optimization procedure in the BBO algorithm is mainly performed based on two mechanisms which are migration and mutation. There are some common features between BBO and other biology-based algorithms. One of them is sharing information between the solutions which also exists in GAs and PSO methods. Solutions of genetic algorithm "die" at the end of each generation, whereas solutions of PSO and BBO survive ever more; although their characteristics change in the optimization process. Solutions of the PSO algorithm tend to clump together in similar groups, whereas solutions of GA and BBO have not this property [32,33].

As seen in literature survey, studies on optimal design of plate fin and compact exchangers using the BBO method were not found. In the present research, the BBO method is applied to the new field; in this way, this algorithm is successfully employed for optimal and economical design of plate fin exchangers. Based on the proposed technique in this research, a full computer code was developed for optimization of plate fin exchangers and three various case studies are solved by it as benchmarks to illustrate the effectiveness and ability of the presented method. The results of this research are compared with those obtained by previous studies conducted with previously presented optimization algorithms and revealed that the presented approach is very quick, powerful and economic scheme for optimization of plate fin exchangers. In the present research the sensitivity analysis is also conducted by the investigation of the sensitivity of the proposed method on electricity price parameters and also on structural and geometrical parameters of the considered heat exchanger. Finally, multi-objective optimization of the studied heat exchanger has been done and a set of optimal points named Pareto optimal solutions have been presented. The main novelty of the current research is proposing a new optimization method for optimal design of the plate fin heat exchangers which results in the optimum and better results in comparison to previously used algorithms for optimization of the consider exchanger by satisfaction of all of the constraints and restrictions which one or some of them were not considered in previous studies.

2. Overview of biogeography based optimization method

The biogeography based optimization algorithm has been proposed based on the theory of biogeography discussed briefly in the previous section. The main idea of the BBO is principally consists of migration and mutation operations [31,33–35]. The brief concept and mathematical expression of migration and mutation operations of the BBO algorithm are presented in following.

2.1. Migration

A population of candidate solutions of the optimization problem using the BBO method similar to other population based optimization algorithms such as GA is expressed as a set of real numbers where each of these numbers is taken as an *SIV* [32,33]. Suitability of candidate solutions in the BBO algorithm is determined using *HSI*; so that the quality of solutions with high *HSI* is higher than those have low *HSI*. The quality of the solution can be improved by modification of them based on other solutions where this modification can be done using migration operation. The migration can be in the form of emigration or immigration. The immigration rate of the solutions, λ_s is a measure to make a decision whether or not to modify each *SIV* [33]. As the *SIV* selected for modification, emigration rates, μ_s of other solutions are used to probabilistically select which solutions between the population set will migrate [33]. It should be noted that similar to other evolutionary methods, some elite and high quality solutions of populations in the BBO method is kept away from migration operation to preventing them from probabilistic corruption [33,34].

2.2. Mutation

In the nature and geographical areas because of some natural disasters or other happenings, *HSI* of a habitat can change abruptly. In such situations, the habitat may deviate from its equilibrium value of *HSI*. In the BBO technique, this event is expressed by the mutation of *SIV* and mutation rates evaluated using species count probabilities [32–34]. The probability of each species count can be computed as follows [31]:

$$P_S^h = \begin{cases} -(\lambda_S + \mu_S)P_S + \mu_{S+1}P_{S+1} & S = 0, \\ -(\lambda_S + \mu_S)P_S + \lambda_{S-1}P_{S-1} + \mu_{S+1}P_{S+1} & 1 \leqslant S \leqslant S_{\max} - 1, \\ -(\lambda_S + \mu_S)P_S + \lambda_{S-1}P_{S-1} & S = S_{\max}. \end{cases} \quad (1)$$

where P_S, λ_s and μ_s are the probability, the immigration and emigration rate of habitats contains S species, respectively. Also $S_{\max}$ represent maximum species count.

Immigration and emigration rates can be calculated as below [31]:

$$\lambda_S = I\left(1 - \frac{S}{S_{\max}}\right), \quad (2)$$

$$\mu_S = \frac{ES}{S_{\max}}, \quad (3)$$

In the above equations, I and E represents the maximum immigration and emigration rates, respectively.

A probability is assigned for each population member, which represents the possibility of that member to be a solution for a specified problem. This probability determines that a specified member will mutate or not. The high probability of a member indicates that the solution is closer to the final solution of the problem; so that member should not mutate whereas members with low probability should mutate to other solutions [33,34]. The mutation rate of each solution as a function of species count probability given as below [31,34]:

$$m(S) = m_{\max}\left(\frac{1 - P_S}{P_{\max}}\right), \quad (4)$$

where $m(s)$, $m_{\max}$, and $P_{\max}$ are the mutation rate for habitat contains S species, maximum mutation rate and maximum probability, respectively.

One of the advantages of mutation is that the variety in population members increased. Also, it prevents the highly probable solutions to be dominant in the population [33]. Mutation action is generally an opportunity to low and high *HSI* solutions to improve. However, mutation is a high-risk action and there is a probability to decrease the quality of solutions after the mutation. Therefore, some elite solutions are kept to save the features of a solution; so that if the solution becomes poorer after the mutation process, it can resume features before the mutation [31,34].

3. Mathematical modeling of the exchanger

3.1. Thermal modeling of plate fin heat exchanger

As mentioned, in this study a cross-flow plate fin exchanger with offset strip fin is considered for optimization. A typical rectangular offset strip fin is shown schematically in Fig. 2. In thermal modeling of the exchanger, working condition is considered to be steady state and the heat transfer coefficients are set to be constant. Also, the thermal resistance of the walls and fouling is neglected [26].

For the cross flow exchanger with both fluids unmixed, effectiveness can be calculated using the ε-NTU method as below [36]:

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C_r}\right)\mathrm{NTU}^{0.22}\left\{\exp\left[-C_r \cdot \mathrm{NTU}^{0.78}\right] - 1\right\}\right], \quad (5)$$

where $C_r = C_{\min}/C_{\max}$ and the NTU is evaluated considering aforementioned assumptions as: [26]:

$$\frac{1}{UA} = \frac{1}{(hA)_a} + \frac{1}{(hA)_b}, \quad (6)$$

$$\mathrm{NTU} = \frac{UA}{C_{\min}}. \quad (7)$$

Coefficient of heat transfer can be calculated using the Colburn factor j as [27]:

$$h = j \cdot G \cdot C_p \cdot Pr^{-\frac{2}{3}} \quad (8)$$

here $G = m/A_{\mathrm{ff}}$, where A_{ff} is free flow area and calculated as:

$$A_{\mathrm{ffa}} = (H_a - t_a)(1 - n_a t_a)L_b N_a, \quad (9)$$

$$A_{\mathrm{ffb}} = (H_b - t_b)(1 - n_b t_b)L_a N_b. \quad (10)$$

In the considered exchanger, the number of fin layers for the hot side is one less than for the cold side, i.e.: $N_a = N_b - 1$. Heat transfer areas for the two sides can be evaluated as below [27]:

$$A_a = L_a L_b N_a[1 + 2n_a(H_a - t_a)], \quad (11)$$

$$A_b = L_a L_b N_a[1 + 2n_b(H_b - t_b)]. \quad (12)$$

Consequently, the total heat transfer area of the exchanger is:

$$A_{\mathrm{HT}} = A_a + A_b. \quad (13)$$

The rate of heat transfer calculated as:

$$Q = \varepsilon C_{\min}(T_{a,1} - T_{b,1}). \quad (14)$$

Pressure drop caused by friction in hot and cold sides is evaluated as:

$$\Delta P_a = \frac{2 f_a L_a G_a^2}{\rho_a D_{h,a}}, \quad (15)$$

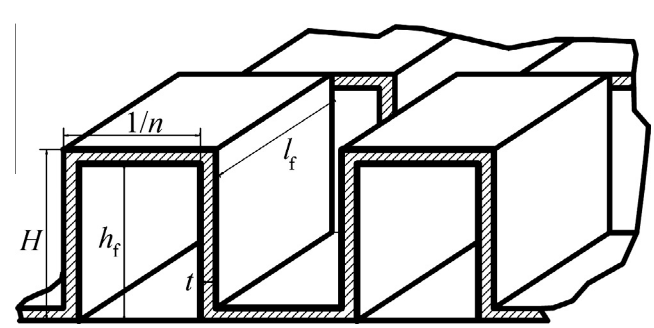

Fig. 2. Typical rectangular offset strip fin core [3].

$$\Delta P_b = \frac{2f_b L_b G_b^2}{\rho_b D_{h,b}}. \quad (16)$$

The Colburn factor j and the fanning factor f for offset-strip fins is calculated as [37]:

$$j = 0.6522(Re)^{-0.5403}(\alpha)^{-0.1541}(\delta)^{0.1499}(\gamma)^{-0.0678} \times \left[1 + 5.269 \times 10^{-5}(Re)^{1.34}(\alpha)^{0.504}(\delta)^{0.456}(\gamma)^{-1.055}\right]^{0.1} \quad (17)$$

$$f = 9.6243(Re)^{-0.7422}(\alpha)^{-0.1856}(\delta)^{0.3053}(\gamma)^{-0.2659} \times \left[1 + 7.669 \times 10^{-8}(Re)^{4.429}(\alpha)^{0.92}(\delta)^{3.767}(\gamma)^{0.236}\right]^{0.1} \quad (18)$$

where $\alpha = \frac{s}{h_f}, \delta = \frac{t}{l_f}, \gamma = \frac{t}{s}, s = \left(\frac{1}{n} - t\right)$ considering $h_f = H - t$.

Reynolds number defined as:

$$Re = \frac{G \cdot D_h}{\mu}, \quad (19)$$

where hydraulic diameter is given as:

$$D_h = \frac{4s \cdot h_f \cdot l_f}{2(s \cdot l_f + h_f \cdot l_f + t \cdot h_f) + t \cdot s}. \quad (20)$$

Eqs. (17) and (18) are valid for $120 < Re < 10^4$ [26,27].

3.2. Constraints and restrictions

In the considered optimization problem in this research, there are seven constraints and restrictions on the design parameters presented in the bellow.

Restrictions on the heat exchanger lengths:

$$0.1 < L_a < 1$$

$$0.1 < L_b < 1$$

Restrictions on the height and thickness of the fins:

$$2 < H < 10$$

$$0.1 < t < 0.2$$

Restriction on fin frequency and offset length:

$$100 < n < 1000$$

$$1 < l_f < 10$$

Restriction on fin layers number:

$$1 < N_a < 200$$

In addition to seven above mentioned restrictions, there are nine constraints related to mathematical formulation restrictions and practical limitations which are presented in the below.

A constraint on the Reynolds number of hot and cold flow of the fluid streams in the heat exchanger:

$$120 < Re_a < 10^4$$

$$120 < Re_b < 10^4$$

The formula used for calculation of the Colburn factor in Eq. (17) and fanning factor in Eq. (18) are valid only for above mentioned range of Reynolds number.

Constraint on the geometrical parameters of the heat exchanger:

$$0.134 < \alpha < 0.997$$

$$0.012 < \delta < 0.048$$

$$0.041 < \gamma < 121$$

Eqs. (17) and (18) for calculation of Colburn and fanning factor are valid for above mentioned limitations.

Constraint on the no-flow length of the heat exchanger (L_c):

$$L_c = 1.5$$

No-flow length of the plate fin heat exchanger is shown in Fig. 1 and can be calculated as:

$$L_c = H - 2t_p + N_a(2H + 2t_p) \quad (21)$$

Constraint of required heat duty of the heat exchanger:

$$Q = 1069.8 \text{ W}$$

Optimally designed heat exchanger should have the required heat duty mentioned above.

Constraints on the maximum pressure at hot and cold flow of the fluid streams in the heat exchanger presented as below.

For hot flow side:

$$P_a \leqslant 9.5 \text{ kPa}$$

and for cold flow side:

$$P_b \leqslant 8 \text{ kPa}$$

3.3. Objective functions

In this research, three various objectives have been considered for optimization of the plate fin exchanger and presented in following.

In the first test case, the objective function is to minimize the heat transfer area of the exchanger. The total heat transfer area of the exchanger is evaluated using Eq. (13) which is related to investment costs.

Minimization of the total pressure drop is selected as the second objective function and investigated in the case study 2. This objective function is related to the operating cost of the exchanger and expressed as below [26]:

$$f(x) = \frac{\Delta P_a}{P_{a,\max}} + \frac{\Delta P_b}{P_{b,\max}}. \quad (22)$$

The minimum total annual cost of the exchanger is settled as the third objective function in this research. The total annual cost of the exchanger consists of investment cost C_{in} and the operating cost C_{op}. The costs of the exchanger, including investment, operating and total annual costs are mathematically modeled as below:

$$C_{in} = A_f \cdot C_A \cdot A_{HT}^{n_1}, \quad (23)$$

$$C_{op} = \left[k_{el}\tau \frac{\Delta P \cdot m}{\eta \cdot \rho}\right]_a + \left[k_{el}\tau \frac{\Delta P \cdot m}{\eta \cdot \rho}\right]_b, \quad (24)$$

$$\text{TAC} = C_{in} + C_{op}. \quad (25)$$

In the above, C_A and n_1 are cost per unit surface area and numeric exponent respectively. As well as k_{el}, τ and η are the electricity price, hours of operation and compressor efficiency respectively [27]. In Eq. (23) A_f is the annual coefficient factor that given as:

$$A_f = \frac{r}{1 - (1 + r)^{-y}}, \quad (26)$$

here r and y are interest rate and depreciation time, respectively [27]. Parameters required for calculating the costs of the exchanger are listed in Table 1.

Table 1
Cost coefficients of heat exchanger [27].

Economic parameters	
Cost per unit area, C_A (\$/m^2)	90
Hours of operation, τ (h)	5000
Electricity price, k_{el} (\$/MW h)	20
Compressor efficiency, η	60%
Exponent of non-linear, n_1	0.6
Depreciation time, y (year)	10
Inflation rate, r	0.1

4. BBO algorithm applied to optimal design of the plate fin exchanger

In the present study, the BBO method has been used for optimization of a plate fin exchanger. The algorithm of the presented design method is demonstrated in the following.

1. Initialization of the BBO variables.
2. Generating initial position of *SIV* (i.e. Heat exchanger length of hot and cold sides, fin height, frequency and thickness, fin-strip length and number of hot side layers). The initial positions of *SIVs*, which are similar to the genes in the genetic algorithm, are generated randomly. These generated positions should satisfy the constraints of the problem, which listed in Section 3.2.
3. Some habitats are being generated. The number of these habitats determined considering the size of the population. Each of these generated habitats is a candidate solution of the problem [34]. The optimal number of the habitats is determined by a trial and error process [34]. The optimal value of the habitats in this problem is 50.
4. Perform design procedure and compute design variables of the exchanger as described in detail in Section 3.
5. Considering predefined emigration and immigration rates, *HSI* of habitats of the population is evaluated [34]. Maximum immigration (I) and emigration rate (E) in this problem is 1.
6. Based on calculated *HSI* values, elite habitats are recognized.
7. Probabilistically perform migration action on non-elite habitats to improve it and recalculate *HSI* of each edited habitat. The suitability of a solution is confirmed, i.e., Each *SIV* must satisfy the specified constraints of the problem.
8. The species count probability of habitats is updated using Eq. (1) [34] and probable mutation is performed.
9. The suitability of a solution is approved.
10. Go to the next iteration.
11. Stop computation loop after a predefined number of iterations or when a specified condition has been satisfied.

The flowchart of the designing algorithm is presented in Fig. 3.

5. Results and discussion

The validity, accuracy and effectiveness of the proposed method in the present research for optimization of plate fin exchangers were investigated by analyzing three various case studies. In these test cases a gas-to-air single pass cross-flow heat exchanger is

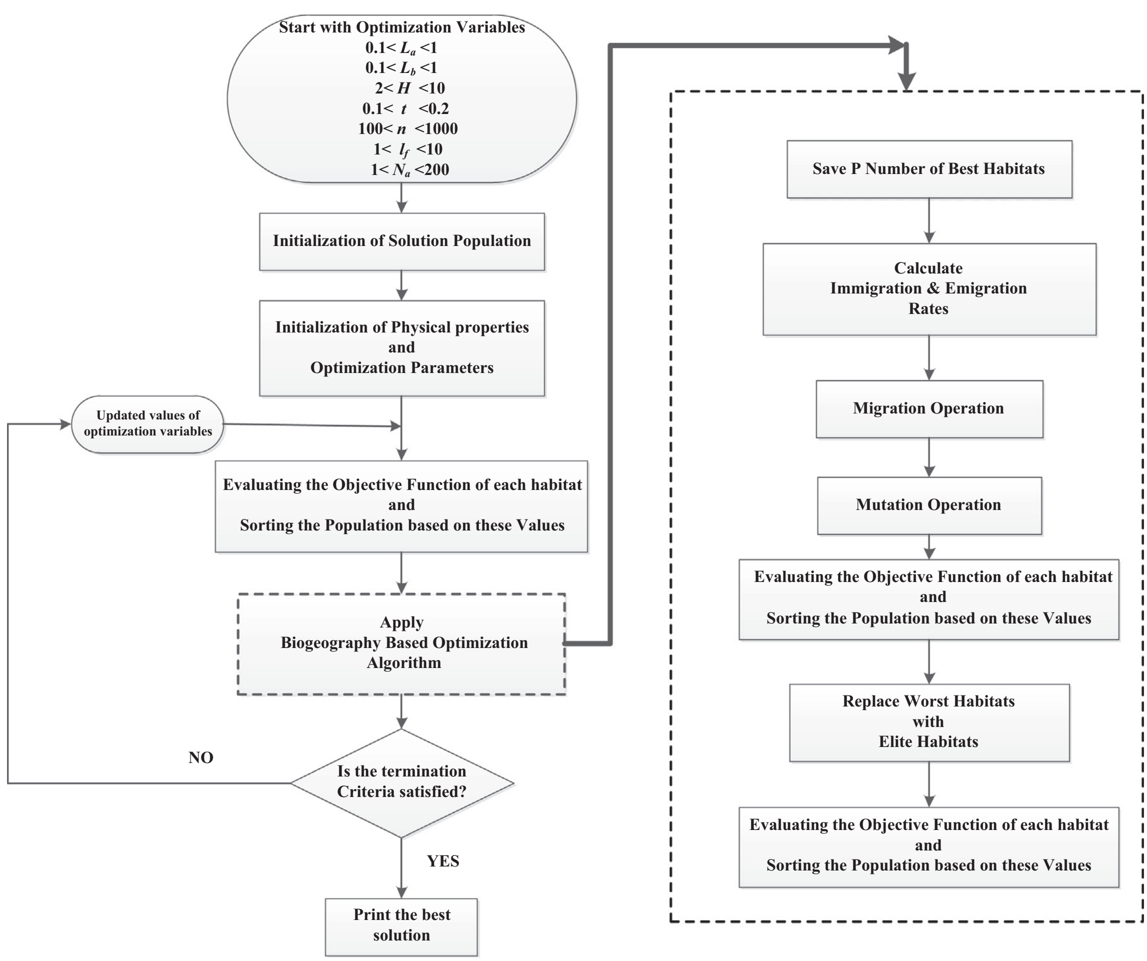

Fig. 3. Flowchart of designing procedure.

considered. The required heat duty of considered exchanger is 1069.8 kW and it is designed for the minimum heat transfer area, total pressure drop and total annual cost targets. Each of these goals has settled as different objective functions and studied in three test cases separately. The surface of the air and gas sides is considered to be offset strip fin. The construction material of the considered exchanger is aluminum with a density of 2700 kg/m^3. Table 2 shows the physical properties of fluids and operating conditions of considered exchanger in the present research [26,27].

In present research seven geometrical and operational parameters of heat exchanger, including the hot flow length (L_a), the cold flow length (L_b), the number of hot side layers (N_a), the fin frequency (n), the fin thickness (t), the fin height (H), and the fin strip length (l_f) are considered as the optimization variables. All of these design variables except the number of hot side layers are continuous parameters. The thickness of the plate, t_p, is a constant assigned to 0.5 mm. The range of variation of mentioned design variables is shown in Table 3. Unlike previous studies and papers which reported values of only some parameters, in the present paper values of all parameters have been reported for all case studies. There are some studies in the literature which have used different intelligent based approaches or other methods for optimization of the mentioned heat exchanger, but in those studies one or a number of the mandatory constraints listed in Section 3.2 have been violated; so, that results are not valid. Therefore, these results have not evaluated and are not compared with valid results of this research or other reliable results. Results of present research satisfy all of the necessary restrictions and constraints of the plate fin heat exchanger design problem presented in Section 3.2. Results of previous studies which have satisfied all of the constraints and consequently their results reliable and valid are considered for comparison of them with the results of the present research.

5.1. Test cases

In this section, three test cases have been considered and results of the novel approach presented in this research have been compared with those obtained from literature. In these case studies all of the restrictions and constraints mentioned in Section 3.2 should be satisfied. Only in the case study 1, the constraint of no-flow length of the heat exchanger is not required.

Table 2
Operating parameters selected for the case studies [26,27].

Parameters	Hot side (a)	Cold side (b)
Mass flow rate, m (kg/s)	1.66	2
Inlet temperature, T (°C)	900	200
Specific heat, C_p (J/kg K)	1122	1073
Density, ρ (kg/m^3)	0.6296	0.9638
Dynamic viscosity, μ (kg/s m)	401E−7	336E−7
Prandtl number, Pr	0.731	0.694
Maximum pressure drop, ΔP (kPa)	9.5	8.00

Table 3
Variation ranges of design parameters [26,27].

Parameters	Min	Max
Hot flow length, L_a (m)	0.1	1
Cold flow length, L_b (m)	0.1	1
Fin height, H (mm)	2	10
Fin thickness, t (mm)	0.1	0.2
Fin frequency, n (m^{-1})	100	1000
Fin offset length, l_f (mm)	1	10
Number of hot side layers, Na	1	200

5.1.1. Case 1 – minimum heat transfer area

In the first case study, minimization of heat transfer area of the heat exchanger is settled as the objective function considering the pressure drop restrictions and required heat duty. The results of the optimum plate fin exchanger designed with the minimum heat transfer area using the proposed novel method in this research are presented in Table 4 and is compared with the results of other previous methods which are hybrid genetic and preliminary design data. As shown in Table 4, the results obtained using the BBO algorithm are better than the results of other traditional methods or intelligence algorithms available in the literature.

As presented in Table 4, heat exchanger area in the BBO method reduced compared with other methods. Reduction of heat exchanger area is due to a reduction of fin layers and relative reduction of exchanger dimensions. In contrast, height of the exchanger fins in the present approach has been increased compared with previous schemes. This configuration in which height of fins has been increased and dimensions of the exchanger have been decreased, results in reduction of the total heat transfer area of the exchanger and pressure drop in both flow sides, too.

The results of the first case study show that the heat transfer area of the exchanger has decreased about 22.93% compared with the original design [38] and about 2.37% in comparison to hybrid genetic [38]. Heat transfer area comparison of present approach and other methods for considered exchanger is shown in Fig. 4 for case 1. Convergence of the objective function in the first case study is shown in Fig. 5. As shown in Fig. 5, the BBO approach is a very quick algorithm and it reaches to optimum value after about 15 iterations while the IHS algorithm reaches to its optimum value after about 5000 iterations [26]. There are eight constraints in this case study listed in Section 3.2 which should be satisfied. There are some results in the literature which are better than the results of the present research (the BBO method), but one or some of the constraints have been violated or have not been considered. So, these results are not valid and comparable with the results of this

Table 4
Parameters of the optimal PFHE for case study 1 (minimum heat transfer area).

	Preliminary design [38]	Hybrid GA [38]	BBO (present work)
L_a (m)	0.3	0.21	0.213
L_b (m)	0.3	0.23	0.223
H (mm)	2.49	5.9	6.77
t (mm)	0.1	0.1	0.108
n (m^{-1})	782	1000	1000
l_f (mm)	3.2	2.1	2.24
N_a	167	91	81
D_h (mm)	–	–	1.505
A_{ffa} (m^2)	–	–	0.10718
A_{ffb} (m^2)	–	–	0.10373
G_a (kg/m^2 s)	–	–	15.4881
G_b (kg/m^2 s)	–	–	19.2809
Re_a	–	–	581.315
Re_b	–	–	863.665
f_a	–	–	0.087273
f_b	–	–	0.068122
ΔP_a (kPa)	9.34	9.50	9.41
ΔP_b (kPa)	6.90	8.00	7.78
j_a	–	–	0.021329
h_a (W/m^2 K)	–	–	0.45655
j_b	–	–	0.017423
h_b (W/m^2 K)	–	–	0.45986
NTU	–	–	6.769
ε	–	–	0.82054
Q (W)	–	–	1069.8
A_a (m^2)	–	–	55.01
A_b (m^2)	–	–	55.01
Objective: A_{HT} (m^2)	142.75	112.69	110.02

research which have satisfied all of the restrictions and constraints. Satisfaction of the constraints in different approaches is investigated and results are shown in Table 5 and violated constraint is highlighted.

5.1.2. Case 2

The minimum total pressure drop of the exchanger is selected as next objective function and studied in the case study 2. Comparison of the results of the proposed method in this research with other results obtained by previous approaches for case study 2 is shown in Table 6. Considering this table it is clear that the BBO approach leads to better results in comparison to other available approaches.

In this case, a reduction in fin frequency compared with the original design and a slight increase of fine layers was observed compared to previous intelligence algorithms used to optimization of the considered exchanger. These changes in exchanger geometry result in a reduction in pressure drop and consequently lead in the reduction of the total pressure drop of the heat exchanger. Overall, reduction of total pressure drop is about 7.14% in comparison to hybrid GA [38] and 3.57% compared with the IHS algorithm [26]. Again performance of the BBO method is better than other available techniques, namely hybrid GA and IHS approaches. Comparison of the Total pressure drop of the considered exchanger in this study with literature methods is presented in Fig. 6 for case 2. Also the convergence of the objective function using the BBO method in this case study is shown in Fig. 7; as seen in this figure, the BBO approach quickly reaches to optimum value after only about 60 iterations while the IHS algorithm converges to its optimal value after about 2000 iterations [26]. Again, similar to the previous case study, satisfaction of the constraints in different approaches investigated and the results are presented in Table 7. When we put data presented by previous studies in the mathematical modeling of the problem, we see some violations of constraints in those results which are highlighted in Table 7. It should be noted that in the present research all of the constraints and restrictions have been satisfied as shown in Table 7. So if the constraints are not considered, the results of the present research would be so better than the results of mentioned approaches that violated or not considered the constraints.

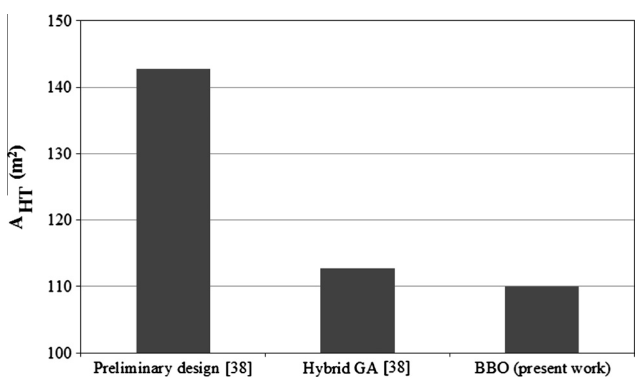

Fig. 4. Heat transfer areas comparison for case study 1.

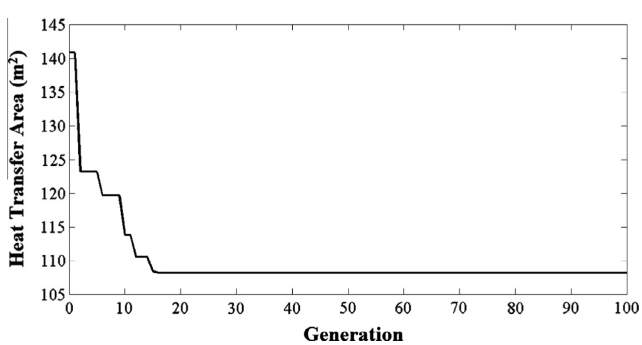

Fig. 5. Convergence of BBO for case study 1.

Table 5
Value of constraints in the approaches investigated in the case study 1.

	Preliminary design [38]	Hybrid GA [38]	IHS [26]	BBO (present work)
α	0.493	0.155	0.136	0.134
δ	0.031	0.047	0.058	0.48
γ	0.084	0.111	0.111	0.121
Re_a	577.09	565.29	572.66	581.32
Re_b	824.86	868.86	854.77	863.67
ΔP_a (kPa)	9.34	9.50	9.44	9.412
ΔP_b (kPa)	6.90	8.00	7.96	7.779
Q (W)	–	–	–	1069.8
A_{HT} (m^2)	142.75	112.69	109.62	110.02

Table 6
Parameters of the optimal PFHE for case study 2 (minimum pressure drop).

	Hybrid GA [38]	IHS [26]	BBO (present work)
L_a (m)	1.00	1.00	1.00
L_b (m)	1.00	1.00	1.00
H (mm)	10	10	9.89
t (mm)	0.1	0.1	0.2
n (m^{-1})	241	211	198.4
l_f (mm)	10	10	10
N_a	71	71	75
D_h (mm)	–	–	6.35
A_{ffa} (m^2)	–	–	0.6979
A_{ffb} (m^2)	–	–	0.7072
G_a (kg/m^2 s)	–	–	2.3785
G_b (kg/m^2 s)	–	–	2.828
Re_a	–	–	376.65
Re_b	–	–	534.45
f_a	–	–	0.0948
f_b	–	–	0.0732
j_a	–	–	0.02119
h_a (W/m^2 K)	–	–	0.06968
j_b	–	–	0.01786
h_b (W/m^2 K)	–	–	0.069149
A_a (m^2)	–	–	363.37
A_b (m^2)	–	–	363.37
A_{HT} (m^2)	–	–	726.74
NTU	–	–	6.77
ε	–	–	0.820566
Q (W)	–	–	1069.8
ΔP_a (kPa)	0.29	0.28	0.268
ΔP_b (kPa)	0.21	0.19	0.191
Objective: $f(x)$	0.056	0.054	0.052

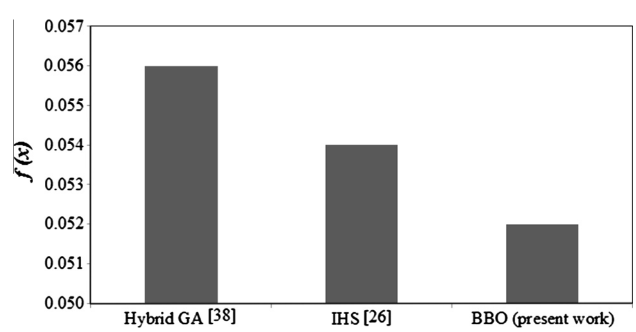

Fig. 6. Total pressure drops comparison for case study 2.

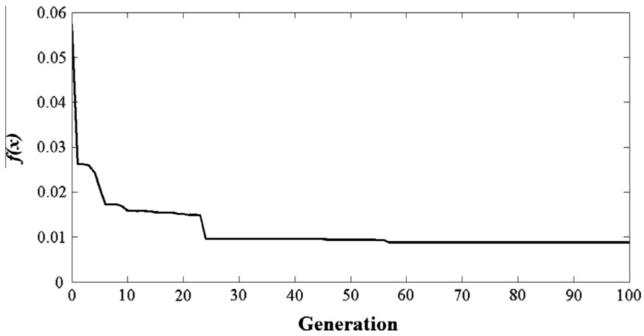

Fig. 7. Convergence of BBO for case study 2.

Table 7
Value of constraints in the approaches investigated the case study 2.

	Hybrid GA [38]	IHS [26]	BBO (present work)
α	0.409	0.468	0.492
δ	0.010	0.010	0.020
γ	0.025	0.021	0.041
L_c	1.4370	1.4370	1.5
Re_a	–	–	376.65
Re_b	–	–	534.45
ΔP_a (kPa)	0.319	0.272	0.268
ΔP_b (kPa)	0.227	0.194	0.191
Q (W)	1085.95	1068.67	1069.8
$f(x)$	0.0619	0.054	0.052

5.1.3. Case 3

The fitness function of the third case study is the minimum total annual cost of the exchanger. The results of the proposed algorithm for this test case are compared with available results in the literature and presented in Table 8; as demonstrated in this table, analogous to previous cases the performance of the BBO scheme is superior to other methods used to optimize of the considered exchanger.

In the case study 3, a decrease of about 2.84% (20.21 \$/year) of the investment cost of the exchanger was observed compared with ICA [27]; this is because of the decrease in the heat transfer area of the designed exchanger in this research compared with ICA [27]. Altogether, the total cost of the designed exchanger using the BBO approach has been reduced about 2% compared with the original design [27]. This means that the design of plate fin exchangers by using the BBO algorithm leads to an economic solution which reduces total cost of plate fin heat exchangers. Comparison of costs and investments of the proposed approach and the ICA method is shown in Fig. 8 for case 3. Considering the difference between investments of the designed exchanger in this study with the original design it is observed that the initial investment of exchanger is reduced about 20.21 \$ while annual operating costs increased only 1.8 \$ in the BBO method compared with the ICA method. So, the payback period needed to compensate the incremental investment through annual savings is 0.08 years. It should be noted that similar to previous case studies; in this test case also all of constraints and restrictions have been employed and satisfied. Whereas, when we are putting the results of Ref. [27] in the mathematical formulation of the problem we conclude that the constraint on the no-flow length of the heat exchanger was not considered and was not satisfied as highlighted in Table 9. So if this constraint were eliminated in our optimization, the results were so better than listed in Table 8.

The convergence of the objective function using the BBO method is shown in Fig. 9. As it is shown in Fig. 9, number of iterations for convergence is about 70 whereas the number of iterations

Table 8
Parameters of the optimal PFHE for case study 3 (minimum total annual cost).

	ICA [27]	BBO (present work)
L_a (m)	0.83	0.793
L_b (m)	1	1
H (mm)	9.7	10
t (mm)	0.2	0.2
n (m^{-1})	228.2	218
l_f (mm)	10	7
N_a	73	74
D_h (mm)	–	5.92
A_{ffa} (m^2)	–	0.6936
A_{ffb} (m^2)	–	0.5575
G_a (kg/m^2 s)	–	2.3934
G_b (kg/m^2 s)	–	3.5874
Re_a	–	353.219
Re_b	–	631.851
f_a	–	0.11026
f_b	–	0.07205
ΔP_a (Pa)	0.280	0.269
ΔP_b (Pa)	0.310	0.325
j_a	–	0.023315
h_a (W/m^2 K)	–	0.077154
j_b	–	0.017581
h_b (W/m^2 K)	–	0.086334
NTU	–	6.769
ε	–	0.820544
Q (W)	–	1069.8
A_f	–	0.162745
A_a (m^2)	–	309.45
A_b (m^2)	–	309.45
A_{HT} (m^2)	–	618.9
UA (W/K)	–	12.608
C_{in} (\$/year)	713.2	692.99
C_{op} (\$/year)	228.8	230.6
Objective: TAC (\$/year)	942	923.59

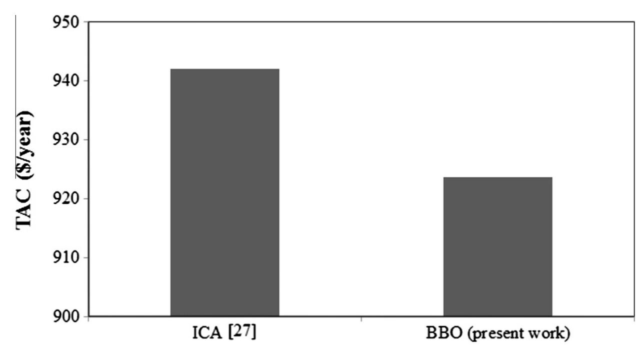

Fig. 8. Total annual costs comparison for case study 3.

for ICA is about 12,000; so the BBO approach has a quick performance in this case study, too. Unlike the previous studies which their results have been evaluated and investigated in above case studies, in the present research numerical values of all parameters of the plate fin exchanger are presented completely in Tables 4, 6 and 8 which are very useful.

Considering the studied test cases, it is obvious that the BBO algorithm is an accurate and economic approach for optimal design of plate fin heat exchangers. Proposed methodology in this paper optimizes the considered plate fin heat exchanger by employing and satisfying all of the constraints and reaches to better results.

Also, as seen in convergences diagrams (Fig. 5, 7 and 9), the BBO method is a fast approach and quickly converges to the optimal solution. Comparisons of run times in this study with other intelligence algorithms which are available in the literature are presented in Table 10 for all of three case studies. The code of the BBO scheme in this research has been developed in MATLAB7

Table 9
Value of constraints in the approaches investigated the case study 3.

	ICA [27]	BBO (present work)
α	0.440	0.448
δ	0.020	0.0286
γ	0.047	0.0455
L_c	1.433	1.5
Re_a	357.19	353.219
Re_b	610.44	631.851
ΔP_a (kPa)	0.280	0.269
ΔP_b (kPa)	0.310	0.325
Q (W)	1071.07	1069.8
TAC ($/year)	942	923.59

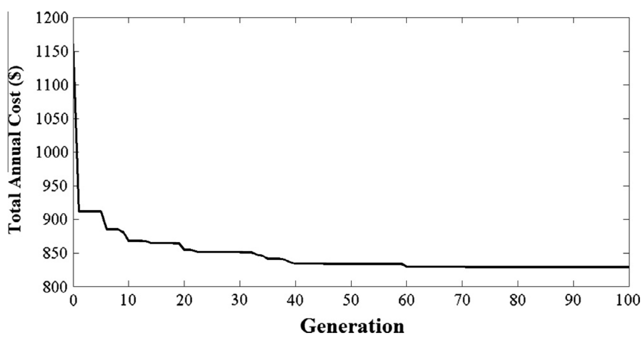

Fig. 9. Convergence of BBO for case study 3.

Table 10
Run time (s) comparison of BBO with other algorithms.

	GA [26,27]	PSO [26]	GAHPSO [26]	IHS [26]	ICA [27]	BBO
Case study 1	4.40	3.59	3.40	3.17	–	2.72
Case study 2	4.30	3.49	3.26	3.05	–	1.11
Case study 3	4.31	–	–	–	3.55	1.84

Table 11
BBO algorithm run time corresponding to 100 iterations for optimization of plate fin and shell and tube heat exchangers.

	Parameters	A	B	C
STHEs case studies [32]	Run time (s)	0.85	0.87	0.85
	Number of iterations	15	70	60
PFHEs case studies	Run time (s)	2.72	1.11	1.84
	Number of iterations	40	20	30

and executed on a 2.00-GHz Core (TM) i7 personal computer with 4-GB RAM. Table 10 shows that the BBO scheme has minimum run time compared with other algorithms, so it is the quickest algorithm in optimal design of plate fin exchangers in comparison with previous used approaches. Comparison of run-time of the BBO method for optimal design of the plate fin exchangers (present study) and shell and tube heat exchangers [32] have been presented in Table 11. Table 11 shows that the BBO method is generally a quick algorithm for optimal design of different type heat exchangers which has low run time and converges to optimum value with a low number of iterations.

5.2. Sensitivity of BBO

In this section sensitivity of the BBO approach to electricity cost parameters including cost per unit area and electricity price has been investigated. Also sensitivity of the BBO method to structural and geometrical parameters of the heat exchanger including fin frequency and height on performance of the exchanger including total annual cost, total heat transfer area and heat duty of the exchanger investigated for all of considered case studies.

5.2.1. Sensitivity to electricity cost parameters

For the sake of completeness, trials were also made by changing electricity cost parameters in the total annual cost in order to evaluate the sensitivity of the BBO to variation in the economic parameters. The effect of ±50% variation of electricity price parameters with respect to the nominal value was examined. Effect of variation of cost per unit area, C_A on the total annual cost is presented in Table 12. As can be seen in Table 12, the BBO responds correctly by trying to decrease the total heat transfer area of the exchanger when cost per unit area increases. This is because the investment cost of the exchanger, C_{in} is a function of cost per unit area; so an increase in cost per unit area results in an increase of the investment cost of the exchanger. Therefore, as cost per unit areas increases, the BBO correctly decreases the total heat transfer area of the heat exchanger to restrict the investment cost and prevent it from excessive increment (see Eq. (23)). So, when the cost per unit area of the exchanger increases up to 50%, heat transfer area of the exchanger decreases from 618.9 m^2 for C_A = 90 \$/m^2 to 618.1 m^2 for C_A = 135 \$/m^2. Conversely, when the cost per unit area decreases, the BBO method correctly increases the total heat transfer area of the exchanger as shown in Table 12. As cost per unit area of the exchanger decreases, the BBO increases the area to reduction of operating costs of the exchanger. So when the cost per unit area of the exchanger decreases up to 50%, heat transfer area of the compact heat exchanger increases from 618.9 m^2 for C_A = 90 \$/m^2 to 625.78 m^2 for C_A = 45 \$/m^2. In particular, when C_A increased by 50% C_{in} increased by 49.89% and C_{op} by 1.69%, while the total cost increased by 37.86%. When instead C_A decreased by

Table 12
Sensitivity analysis of BBO to cost per unit area, C_A.

	C_A = 45 \$/m^2	C_A = 90 \$/m^2	C_A = 135 \$/m^2
L_a (m)	0.81	0.793	0.8
L_b (m)	0.99	1	1
H (mm)	10	10	10
t (mm)	0.2	0.2	0.2
n (m^{-1})	218	218	219
l_f (mm)	7	7	7.1
N_a	74	74	73
D_h (mm)	5.92	5.92	5.90
A_{ffa} (m^2)	0.6866	0.6936	0.6841
A_{ffb} (m^2)	0.5694	0.5575	0.5548
G_a (kg/m^2 s)	2.4176	2.3934	2.4267
G_b (kg/m^2 s)	3.5125	3.5874	3.6052
Re_a	356.788	353.219	357.064
Re_b	618.668	631.851	633.103
f_a	0.10945	0.11026	0.10886
f_b	0.07314	0.07205	0.07158
ΔP_a (kPa)	0.278	0.269	0.276
ΔP_b (kPa)	0.313	0.325	0.327
j_a	0.023199	0.023315	0.023139
h_a (W/m^2 K)	0.077544	0.077154	0.077639
j_b	0.017757	0.017581	0.017522
h_b (W/m^2 K)	0.085381	0.086334	0.086474
NTU	6.827	6.769	6.789
ε	0.821282	0.820544	0.820794
A_f	0.162745	0.162745	0.162745
A_a (m^2)	312.89	309.45	309.08
A_b (m^2)	312.89	309.45	309.08
A_{HT} (m^2)	625.78	618.9	618.16
UA (W/K)	12.715	12.608	12.644
C_{in} ($/year)	348.80	692.99	1038.72
C_{op} ($/year)	230.55	230.6	234.50
TAC ($/year)	579.35	923.59	1273.22

50% C_{in} decreased by 49.67% and C_{op} by 0.02%, while TAC decreased by 37.27%.

Effect of variation of electricity price, k_{el} on the total annual cost is presented in Table 13. Electricity price is related to operating cost (Eq. (24)); so, reduction of electricity price leads in increment of total pressure loss in order to reduction of investment cost and consequently total cost of the exchanger. Therefore, when the electricity price decreases by 50%, total pressure drop of the heat exchanger increases from 0.594 kPa for k_{el} = 20 \$/MW h to 0.601 kPa for k_{el} = 10 \$/MW h. In particular, when k_{el} increased by 50% C_{in} increased by 0.12% and C_{op} by 52.34%, but the total cost of the exchanger increased by only 12.98%. But when k_{el} decreased by 50%, C_{in} decreased by 0.18% and C_{op} by 49.3%, while TAC decreased by 12.45%.

Table 13
Sensitivity analysis of BBO to electricity price, k_{el}.

	k_{el} = 10 \$/MW h	k_{el} = 20 \$/MW h	k_{el} = 30 \$/MW h
L_a (m)	0.8	0.793	0.83
L_b (m)	1	1	0.97
H (mm)	10	10	10
t (mm)	0.2	0.2	0.2
n (m^{-1})	218.5	218	217.1
l_f (mm)	7.1	7	7
N_a	73	74	73
D_h (mm)	5.91	5.92	5.94
A_{ffa} (m^2)	0.6841	0.6936	0.6638
A_{ffb} (m^2)	0.5548	0.5575	0.5758
G_a (kg/m^2 s)	2.4264	2.3934	2.5007
G_b (kg/m^2 s)	3.6049	3.5874	3.4735
Re_a	357.620	353.219	370.171
Re_b	634.088	631.851	613.641
f_a	0.10876	0.11026	0.10654
f_b	0.07152	0.07205	0.07360
ΔP_a (kPa)	0.275	0.269	0.296
ΔP_b (kPa)	0.326	0.325	0.301
j_a	0.023120	0.023315	0.022774
h_a (W/m^2 K)	0.077564	0.077154	0.078748
j_b	0.017509	0.017581	0.017826
h_b (W/m^2 K)	0.086400	0.086334	0.084757
NTU	6.770	6.769	6.769
ε	0.820551	0.820544	0.820542
A_f	0.162745	0.162745	0.162745
A_a (m^2)	308.50	309.45	308.86
A_b (m^2)	308.50	309.45	308.86
A_{HT} (m^2)	617.0	618.9	617.72
UA (W/K)	12.609	12.608	12.608
C_{in} (\$/year)	691.71	692.99	692.19
C_{op} (\$/year)	116.92	230.6	351.29
TAC (\$/year)	808.63	923.59	1043.48

5.2.2. Sensitivity to structural parameters of heat exchanger

In this section sensitivity of the proposed method in the present research on geometrical and structural parameters investigated. In this regard, effects of variations of height and frequency of the fins on the performance of the designed plate fin heat exchanger including its heat duty, the total annual cost and total heat transfer area have been studied.

Fin frequency is one of the design parameters of the optimal design of plate fin heat exchanger which range of variation of this parameter is 100–1000 (1/m). Fin height is one of other design parameters of the exchanger which can be changed from 2 to 10 (mm). Effects of variation of the fin height and frequency on the thermo-economic performance of the designed exchanger in the studied test cases are shown in Figs. 10–17. Effects of variation of fin height and frequency on heat duty of the designed exchanger in the case study 1 are presented in Fig. 10. It is obvious that increase of fin height and frequency results in increase of heat duty of the designed exchanger. This is because when fin height and frequency increases, the heat transfer area of the exchanger in fixed dimensions increases and consequently enhances the heat duty of the exchanger. Considering Eq. (5) it is clear that the effectiveness of the heat exchanger is a function of NTU and by considering Eq. (7) it is seen that NTU is a function of the total heat transfer. From Eqs. (11)–(13) we know that total heat transfer area of the exchanger is a function of fin height and frequency; so, the increase of these parameters leads in the enhancement of the exchanger total heat transfer area and this increases NTU of the exchanger. As NTU increases, effectiveness and consequently heat transfer rate of the exchanger enhances. Regarding Fig. 10 it is concluded that when n increases from 200 to 1000 (1/m), heat duty of the exchanger increases about 600 W (more than 130%). Effects of fin frequency and height variations on the heat duty of the heat exchanger designed in case study, 2 are shown in Figs. 11 and 12, respectively. It can be seen that there is a minimum in heat duty of the exchanger respect to the height of the fin. In other words, when fin height increases heat duty of the exchanger first decreases up to a minimum point, then more increasing of fin height increases the total heat transfer rate of the exchanger.

Effects of variation of fin height and frequency in the total annual cost of the exchanger are shown in Figs. 13–15. As seen in these figures, an increase in fin height results in reduction of the total annual cost of the exchanger. The total annual cost of the exchanger is the sum of the investment and operating costs. As the fin height increases, the total heat transfer area of the exchanger increases (as discussed above) and this results in an increase of the investment cost of the exchanger; because this cost is a function of the total heat transfer area of the exchanger (Eq.

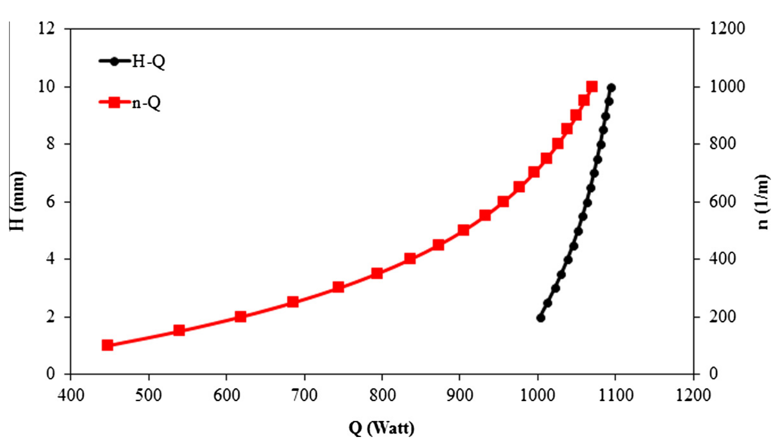

Fig. 10. Effects of the fin height and frequency on the heat duty of the designed heat exchanger using the BBO method in the case study 1.

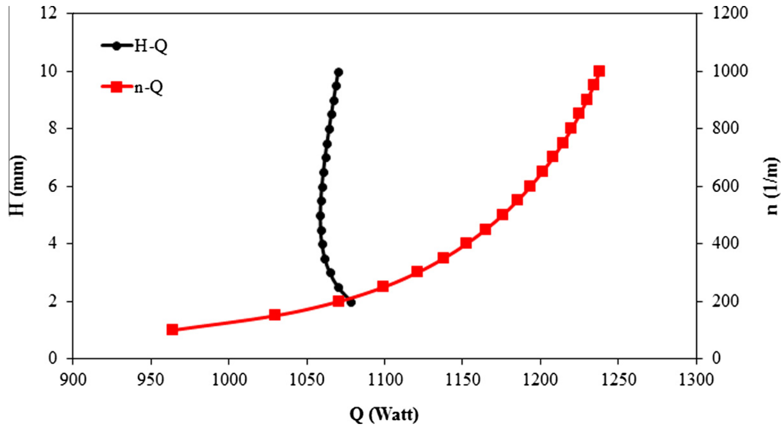

Fig. 11. Effects of the fin height and frequency on the heat duty of the designed heat exchanger using the BBO method in the case study 2.

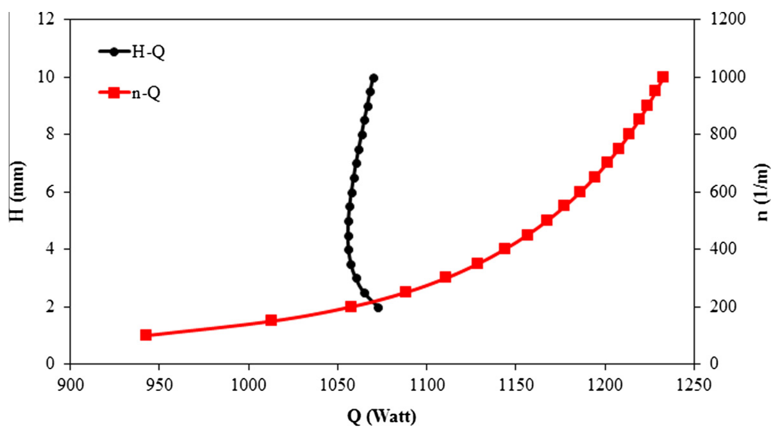

Fig. 12. Effects of the fin height and frequency on the heat duty of the designed heat exchanger using the BBO method in the case study 3.

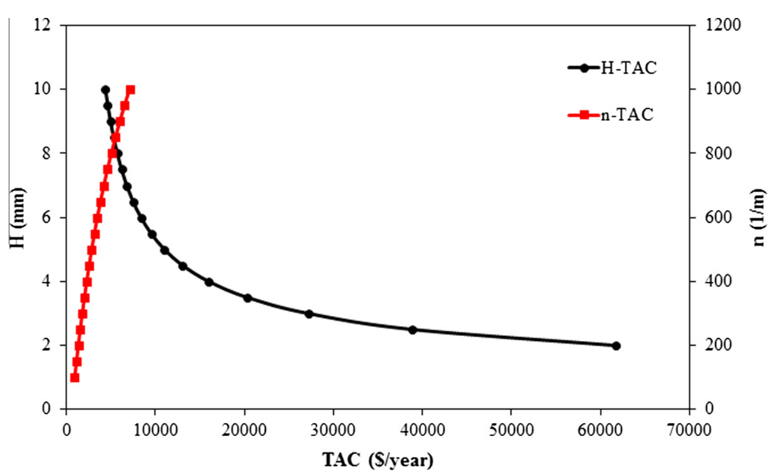

Fig. 13. Effects of the fin height and frequency on the total annual cost of the designed heat exchanger using the BBO method in the case study 1.

(23)). But when fin height increases, hydraulic diameter between fins increases and consequently pressure losses of cold and hot flows in the heat exchanger decreases. So, the operating cost of the heat exchanger decreases. Therefore, the height of the fins has a conflicting effect on the costs of the exchanger. Reduction of the operating cost due to increase in fin height dominates on the increases of investment cost; hence, increase of fin height results in reduction of the total annual cost of the exchanger.

The effect of increasing the fin frequency on the total annual cost of the heat exchanger also is shown in Figs. 13–15 for the designed exchanger in the studied test cases. As it can be seen, increasing the fin frequency results in continuous increase of the total annual cost of the heat exchanger. As mentioned later, investment cost of the exchanger is a function of the total heat transfer area of the exchanger and the total heat transfer area is a function of the fin frequency which can be seen in the Eqs. (11)–(13); so as

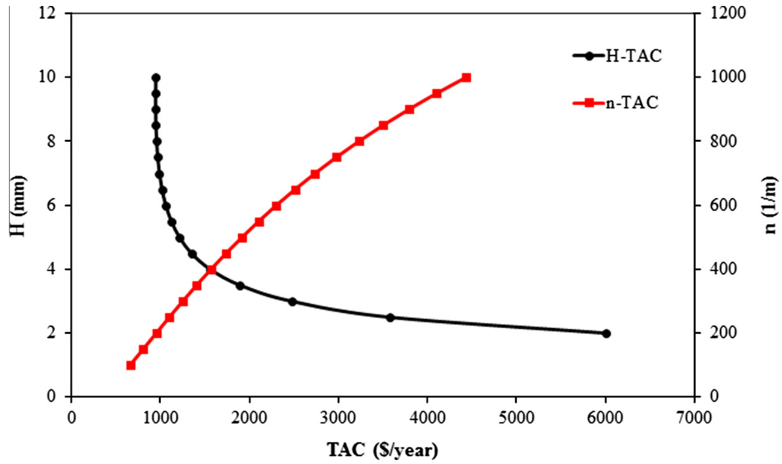

Fig. 14. Effects of the fin height and frequency on the total annual cost of the designed heat exchanger using the BBO method in the case study 2.

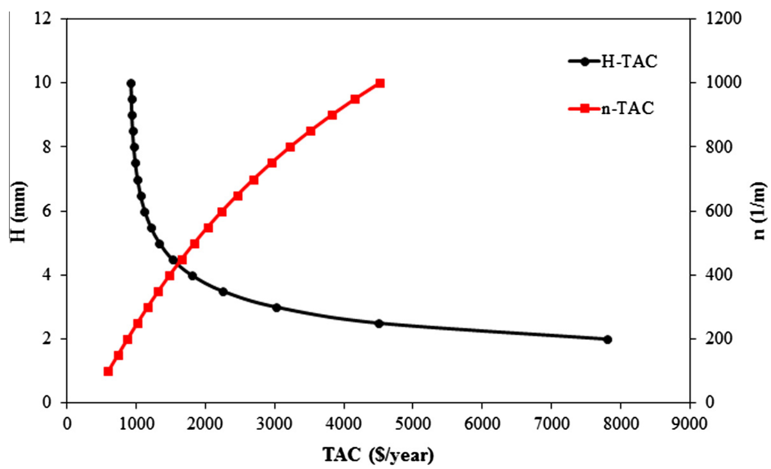

Fig. 15. Effects of the fin height and frequency on the total annual cost of the designed heat exchanger using the BBO method in the case study 3.

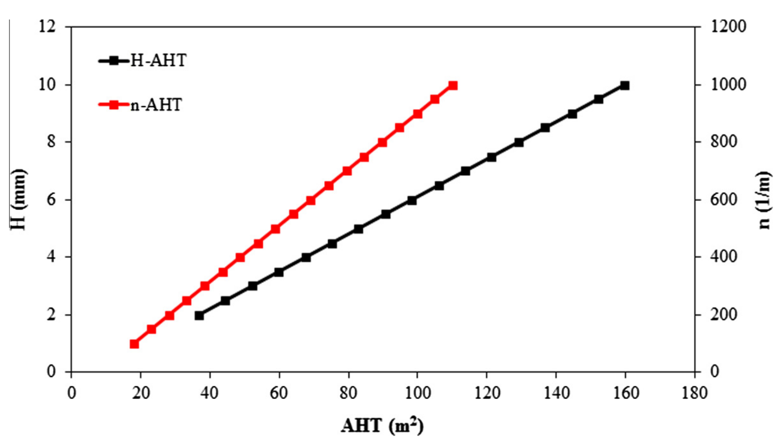

Fig. 16. Effects of the fin height and frequency on the total heat transfer area of the designed heat exchanger using the BBO method in the case study 1.

the fin frequency increases, total heat transfer area and investment of the exchanger increases. Also, when the fin frequency increases, fanning friction factor due to increases in heat transfer area increases and hydraulic diameter of the gaps between fin layers due to the increment of the fin frequency decrease; these phenomena increases pressure loss of the streams in the exchanger which leads to increase of operating cost, too. Therefore, an increase of the fin frequency increases the both investment and operating cost; so this phenomenon results in continuing increasing of the total annual cost of the heat exchanger.

Effects of variation of fin frequency and height of the exchanger on heat transfer area of the designed plate fin heat exchanger in the case study 1 are shown in Fig. 16. It is clear that the increase of fin height and frequency increases the total heat transfer area

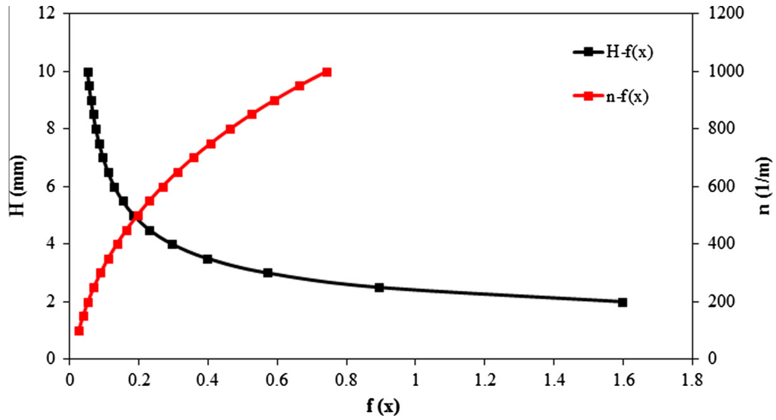

Fig. 17. Effects of the fin height and frequency on the total heat transfer area of the designed heat exchanger using the BBO method in the case study 2.

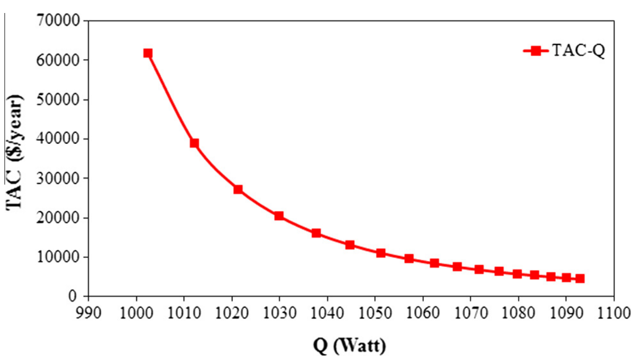

Fig. 18. Pareto optimal points for minimum heat duty and total annual cost using the BBO method.

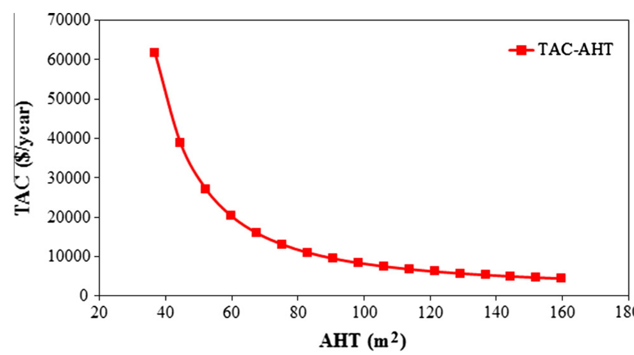

Fig. 19. Pareto optimal points for minimum total heat transfer area and total annual cost using the BBO method.

of the heat exchanger linearly. This is because the total heat transfer area of the heat exchanger is a linear function of fin height and frequency as it can be seen in Eqs. (11)–(13).

Effects of fin height and frequency variation also are investigated on the total pressure drop of the exchanger which is the objective function of the case study 2 and presented in Fig. 17. It is clear that the increase of fin height results in reduction of total pressure, loss of the cold and hot fluid streams in the designed heat exchanger. This is due to the increment of the hydraulic diameter of the passage between fins; but when the fin frequency of the fin layers increases, the total pressure loss of the flows in the exchanger increases; because when fin frequency increases, fanning friction factor increases and hydraulic diameter between fin layers decreases which results in an increment of the total pressure loss of the heat exchanger.

5.3. Pareto optimal solution

A multi-objective optimization have been conducted on considered heat exchanger in case study 1 and minimization of heat transfer rate and total annual cost of the exchanger is settled as objective functions in order to achieve a set of optimal solutions and the obtained results are presented in Fig. 18. By using the Pareto optimal points, for any specific required rate of heat transfer, the designer can select the optimal values which lead to the least possible total annual cost. Multi-objective optimization of the considered exchanger by minimization of the total heat transfer area and total annual cost of the exchanger also leads to a set of optimal points or Pareto optimal solutions which are shown in Fig. 19.

6. Conclusion

In the present research, a novel method for optimal design of plate fin heat exchangers is presented using a biogeography based optimization algorithm. A computer code was developed in this research based on the new presented design method and various case studies including different objectives were solved by it to show the performance and ability of the new method. Comparing the results of the present research with available results in the literature for the considered case studies reveals savings in investment costs up to 2.84% with an overall decrease of total costs up to 2%, reduction of total pressure drop up to 7.14% and reduction of the total heat transfer area of the exchanger up to 22.93% in the designed exchanger using the BBO method. This shows that the presented approach can improve performance of plate fin exchangers. It should be noted that in this research alike of a majority of the studies, all of constraints and restrictions have been considered and satisfied. So these improved results are obtained by considering and satisfying all of constraints; whereas, one or a number of constraints have been violated or not considered in the previous studies. In all of the investigated case studies, objective functions are improved using the proposed method. It is concluded that the biogeography based optimization is an accurate approach compared with traditional methods; in addition, the BBO method shows very quick performance and converges to optimum value in less computational time when compared with traditional algorithms such as ICA, GA, PSO which previously have been used to optimal design of this type of heat exchangers. Sensitivity of the proposed method in this research on energy price parameters including cost per unit area, electricity price and sensitivity to structural and geometrical parameters of the heat exchanger including fin height and frequency has investigated. The results of the sensitivity analysis indicate that the BBO performs correctly. Also multi-objective optimization of the considered heat exchanger has conducted and a set of optimal solutions, namely Pareto optimal solutions for the designed heat exchanger presented. In addition to researchers, the optimization method presented in this research can help engineers to optimize plate fin exchangers in practical applications.

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