Genetic algorithm for the design and optimization of a shell and tube heat exchanger from a performance point of view

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Abstract *– A new approach to optimize the design of a shell and tube heat exchanger (STHX) is developed via a genetic algorithm (GA) to get the optimal configuration from a performance point of view. The objective is to develop and test a model for optimizing the early design stage of the STHX and solve the design problem quickly. GA is implemented to maximize heat transfer rate while minimizing pressure drop. GA is applied to oil cooler type OKG 33/244, and the results are compared with the original data of the STHX. The simulation outcomes reveal that the STHX's operating performance has been improved, indicating that GA can be successfully employed for the design optimization of STHX from a performance standpoint. A maximum increase in the effectiveness achieves 57% using GA, while the achieved minimum increase is 47%. Furthermore, the average effectiveness of the heat exchanger is 55%, and the number of transfer units (NTU) has improved from 0.475319 to 1.825664 by using GA.*

Keywords: *Genetic algorithm, Optimization, Overall heat transfer coefficient, Shell and tube heat exchanger*

1. INTRODUCTION

The heat exchanger is a thermal medium that transfers heat between two or more fluids at different temperatures [1-3]. Heat exchangers are widely utilized in industrial applications such as chemical processing systems, waste heat recovery units, power plants, food processing systems, air conditioning systems, refrigeration, heating, and automobile radiators [2].

According to specific heat exchange requirements, various types of heat exchanger equipment such as casing and tube, bare tube, finned tube, spiral, plate, frame, and plate coil are used [4].

Among these heat exchangers, STHX is the most commonly used type due to its easy maintenance, application versatility, and resistance to high temperature and pressure [5-7]. This type comprises several round tubes mounted inside a cylindrical shell and has five major components [8]: the shell, tube bundle, front head, rear head, and the baffles [8,9]. The fluid enters and exits the tube side through the rear and front headers. Baffles support the tubes by increasing the turbulence of the shell fluid and directing the fluid flow to the tubes (approximately transversely), increasing the heat transfer intensity. Heat exchange occurs when one fluid flows outside the tubes, and the other fluid flows through the tubes [8].

Several geometric parameters determine the STHX performance [9,10], including the flow rate ratio between the tube and shell sides, the heat transfer coefficient on the shell and tube sides, the type and spacing of baffles, pressure drop, fouling, and turbulence. There are three common types of STHX as follows; STHX with segmental baffles (STHX-SG), STHX with continuous helical baffles (STHX-CH), STHX possessing staggered baffles (STHX-ST). STHX-SG is most common and widely used because of its ease of installation and low cost. STHX-SG, on the other hand, offers high heat transfer performance due to its crossflow on the shell side. A type of STHX known as STHX-CH produces shell-side helical flow. The last one, STHX-ST used both continuous helical baffles and segmental; it has the convenience of segmental baffles in terms of fabrication and installation and the helical flow generated by helical baffles. Shell inner diameter, outer tube diameter, baffle spacing, baffle cut, and baffle orientation angle are all design parameters that substantially impact the overall performance of this heat exchanger [5].

The design of STHXs that meets a specified set of design constraints and provides the optimum heat duty includes many geometrical and operative variables [11].

An optimum heat exchanger configuration has been extensively applied with artificial intelligence (AI) methodology, particularly AI-based on metaheuristics. For the cost-effective design of STHX, GAs have been adopted as an optimization method to improve the design [9].

Many standards for STHX aimed to help designers, engineers, and users work more efficiently. Many producers and consumers widely use tubular exchanger manufacture association (TEMA) standards, covering manufacturing tolerances, thermal relationships, performance data, installation, maintenance and operation, vibration standards, mechanical standards, and recommended good practices [12].

Optimization of STHX has been conducted with metaheuristics and deterministic methods [13]. High-dimensional problems cannot be solved with an exact optimization algorithm. It is impossible to conduct a comprehensive search with the size of the problem because the search space grows exponentially with size. The population-based optimization algorithms can be used to find near-optimal solutions to difficult optimization problems. Metaheuristic algorithms are optimization methods based on a stochastic approach that can produce solutions with good and reliable approximations in a reasonable amount of time [16]. These approaches are one of the most complex computational intelligence models that greatly approximate optimization problems [13,17].

The objective function does not need to be differentiated for metaheuristics. Metaheuristics are more efficient than simple heuristics or calculus-based methods. As a result, they may be used to search over many solutions with less computational effort than traditional calculus-based methods [14,15]. However, algorithms of this sort are often constructed on disordered solving strategies based on random numbers rather than robust and accurate computations and hence may not always reach the global optimal point [16]. Although they have no guarantee of good performance, metaheuristic algorithms have been found to perform acceptably in many use cases [18,19].

Bio-inspired and physics/chemistry-based algorithms are the major divisions. Biogeography-based optimization (BBO), estimation of distribution algorithms, differential evolution (DE), (EDAs), and flower pollination algorithm (FPA) are mentioned as an example of the so-called bio-inspired algorithms. Some other algorithms are swarm intelligence-based, a subcategory of bio-inspired algorithms such as artificial bee colony (ABC), ant colony optimization (ACO), cuckoo search (CS), grey wolf optimizer (GWO), particle swarm optimization (PSO), and whale optimization algorithm (WOA). Simulated annealing (SA), big bang-big crunch (BBBC), and harmony search algorithm (HSA) are examples of physics/chemistry-based algorithms that were inspired by physical or chemical phenomena [13].

GA has successfully obtained optimal designs for STHE in several works, including [20, 21].

Selbas et al. [22] applied GA to optimize the STHX economically by varying the design variables: outer shell diameter, outer tube diameter, baffle cut, baffle spacing, number of tube passes, and tube layout. In addition, they determined the heat transfer area as an objective function using the logarithmic mean temperature difference (LMTD) method. They concluded that the heat transfer area increases as the total cost increases.

Antonio et al. [23] used GA in Toolbox to optimize a heat exchanger; the objective function is based on the heat exchanger's total cost. They compare their results to conventional approaches by reducing the objective function while considering decision variables such as tube diameter, casing diameter, and septum area. Compared with traditional methods, the results showed that the performance of the heat exchanger was improved.

Guo et al. [24] developed a new approach for STHX optimization design using entropy generation minimization and GA. The rate of dimensionless entropy generation was used as the objective function. A variety of design variables were taken into account. They found that the effectiveness of the STHX was significantly increased while pumping power was reduced simultaneously.

Patel et al. [25] investigated the optimization of STHXs from an economic viewpoint using PSO. They compared the optimization results to those obtained by the GA and found that the PSO algorithm outperforms the GA in terms of predictive performance.

Vahdat Azad and Amidpour [26] optimized STHX using a GA to lower the total cost of the heat exchanger. Although GAs, CS, and firefly algorithm (FA) were used by Khosravi et al. [27], they concluded that when GAs were implemented, it was impossible to find designs that met the constraints while FA could come up with good designs.

Dastmalchi et al. [28] investigated the PSO algorithm in a double pipe heat exchanger with finned tubes.

Their findings revealed that as the Reynolds number increased, the optimum height of the fin increased as well.

Saijal and Danish [5] designed the STHX-ST by incorporating helical and segmental baffles features. They investigated the influence of five design parameters through numerical analysis: outer tube diameter, inner shell diameter, septal orientation angle, septum cut-out, and septum spacing on STHX-ST performance. They implemented multi-objective optimization using GA, where the heat transfer rate is maximized while the pressure drop is minimized. They used artificial neural networks (ANNs) to approximate the optimization of the objective function. Using the computational fluid dynamics (CFD) and Taguchi orthogonal test table analysis, the training data for ANNs are generated. Finally, they provided the optimal design parameters for minimum pressure drop and maximum heat transfer rate.

There may be contradictions regarding the efficiency of using GA to optimize STHX because of the confusing relationships between optimizing STHXs economically and optimizing them from a performance standpoint.

The GA enables the design problem to be solved quickly and enables the designer to examine some high-quality alternative solutions, giving the designer more flexibility concerning traditional methods in his final selection [23].

Improving the heat exchange capacity of the heat exchanger used in the industry by improving its effectiveness to increase production capacity is a great challenge. Therefore, this study aims to improve the effectiveness of STHX, which already works for an industrial application, by using GA. We used GA to improve the design optimization of STHX from a performance point of view. MATLAB and the optimization toolbox of MATLAB are used to apply our mathematical model. The proposed algorithm is compared with the STHX data of each run to demonstrate the effectiveness and best points under each run.

The main contributions of this research paper are:

- Formulate a mathematical model for oil cooler type OKG 33/244 STHX
- Applying the GA to an industrial model of STHX (oil cooler type OKG 33/244).
- Improve the effectiveness of the oil cooler type OKG 33/244.
- Deciding the issue of whether or not GA can improve the effectiveness of STHX, from a performance point of view.

Following is the remainder of this paper; a description of the hydraulic thermal design formula of an STHX can be found in section 2, and an overview of the GA can be found in section 3. Next, section 4 describes the results and computational analysis, while the last section (section 5) provides the concluding remarks.

2. DESIGN FORMULATIONS OF A SHELL AND TUBE HEAT EXCHANGER

This section presents the equations used in the current study to calculate the heat transfer coefficients (HTC) of STHX and the objective function of the study.

2.1. SHELL AND TUBE HEAT TRANSFER COEFFICIENT

The convective HTC depends on the flow regime and the fluid velocity. The HTC for the flow in the tubes can be determined using several equations. Regarding the phenomenon of intra-tube flow and according to the pressure drop (PD) calculations and the flow regime in HTCs, the intra-tube flow is divided into transition, relaxation, and developed turbulence. The dimensionless Reynolds number in the mobility factor concept is the criterion for separating these three areas. The fluid acts as a barrier to its movement [29]. Laminar HTC is calculated using the Seider-Tate correlation described in [30,31]. Hausen correlation [32] is applied to transient conditions, whereas Dittus-Boelter correlation [29] is widely used to describe fully developed turbulent (turbulent area) flow conditions in tubes.

The heat transfer surface area, A, for the exchanger is firstly determined according to the following equations (1) to (6) [33]:

$$d_{t,o} = d_{t,i} + S_t \tag{1}$$

Calculate the number of tubes

$$\mathrm{n_t} = \frac{\dot{\mathrm{m}}_{\mathrm{c}}}{\rho_{\mathrm{c}} * \frac{\pi}{4} * \mathrm{d}_{\mathrm{t,i}}^2 * \mathcal{V}_{\mathrm{t}}} \tag{2}$$

Calculate heat transfer surface area

$$\mathrm{A_s} = \pi * \mathrm{d_{t,o}} * \mathrm{l_t} * \mathrm{n_t} * \mathrm{n_p} \tag{3}$$

Calculate the Tube side Reynolds number

$$\mathrm{Re_t} = \frac{\rho_{\mathrm{t}} \mathcal{V}_{\mathrm{t}} \mathrm{d_{t,i}}}{\mu_{\mathrm{t}}} \tag{4}$$

Calculate Darcy friction factor

$$\mathrm{f_t} = (1.82 \log_{10} \mathrm{Re_t} - 1.64)^{-2} \tag{5}$$

Calculate tube side convective coefficient

$$h_t = \begin{cases} \dfrac{k_t}{d_{t,i}} \left[3.657 + \dfrac{0.0677 \left(R_{et} P_{rt} \left(\frac{d_{t,i}}{l} \right) \right)^{1.33}}{1 + 0.1 P_{rt} \left(R_{et} \left(\frac{d_{t,i}}{l} \right) \right)^{0.3}} \right]; R_{et} < 2300 \\ \dfrac{k_t}{d_{t,i}} \left[\dfrac{\left(\frac{f_t}{8} \right) (R_{et} - 1000) P_{rt}}{1 + 12.7 \left(\frac{f_t}{8} \right)^{\frac{1}{2}} \left(P_{rt}^{\frac{2}{3}} - 1 \right)} \left(1 + \frac{d_{t,i}}{l} \right)^{0.67} \right]; 2300 < R_{et} < 10^3 \\ 0.027 \dfrac{k_t}{d_{t,i}} R_{et}^{0.8} P_{rt}^{1/3} \left(\dfrac{\mu_t}{\mu_{tw}} \right) 0.14 ; R_{et} > 10^4 \end{cases} \tag{6}$$

In the above equation (6), the coefficients are calculated; for laminar flow (R_{et}<2300), for transition flow (2300<R_{et}<10^3), for fully developed turbulent flow (R_{et}>10^4).

Where ft and kt are the Darcy friction factor and the tube side thermal conductivity, respectively. All unmentioned symbols are listed in Table (7) in the nomenclature table.

The hydraulic shell diameter D_e is computed as:

$$D_e = \left(\frac{4 * A}{P}\right) \tag{7}$$

$$A = \frac{\pi}{4}D^2 - \frac{\pi}{4}{d_{t,o}}^2 n \qquad \& \qquad P = \pi d_{t,o}\, n + \pi\, D \tag{8}$$

The fluid velocity inside the tube, Reynolds number, and Prandtl are calculated from the equations 9-10:

$$v_s = \frac{m_h}{\rho_s \frac{\pi}{4} D_e^{\,2} n_t} \tag{9}$$

$$Re_s = \frac{\rho_s \mathcal{V}_s D_e}{\mu_s} \tag{10}$$

The shell side heat transfer coefficient h_s is calculated using Kern's formula for STHX-SG [9].

$$h_s = 0.36 \frac{k_t}{D_e} Re_s^{0.55} Pr_s^{1/3} \left(\frac{\mu_s}{\mu_{sw}}\right)^{0.14} \tag{11}$$

On both the shell and tube sides, the overall heat transfer coefficient (U) is calculated using the heat transfer coefficients and fouling resistances. Fouling resistances are calculated based on literature data for various fluid types and operating temperatures. The overall heat transfer coefficient is calculated using the formula [11]:

$$U = \frac{1}{\frac{1}{h_s} + R_{fs} + \left(\frac{d_{t,o}}{d_{t,i}}\right)\left(R_{ft} + \frac{1}{h_t}\right)} \tag{12}$$

The minimum and maximum thermal capacity, C_{min} and C_{max}, respectively, are defined as below,

$$C_c = \dot{m_c} * C_{p,c} \text{ and } C_h = \dot{m_h} * C_{p,h}\ , if \quad C_c > C_h \tag{13}$$

$$C_{max} = C_c\ ,\ C_{min} = C_h \quad or \quad C_{max} = C_h,\ C_{min} = C_c \tag{14}$$

2.2. PROBLEM DESIGN

In this study, the objective function for the design optimization issue is the thermal efficiency of the oil cooler type OKG 33/244 STHX (Fig.1) by varying the design variables: tube inside diameter($d_{t,i}$), tubes length (L), shell diameter (D), number of tubes (n_t), effectiveness (ε), and output temperature of hot fluid ($T_{h,o}$).

The effectiveness-number of transfer units (ε-NTU) method and the LMTD method are often used for heat exchanger design and analysis [34]. In heat exchanger analysis, LMTD is straightforward when both the outlet and inlet temperatures of the hot and cold fluids can be determined from the energy balance. In addition, it is great for determining the size of a heat exchanger to achieve the right outlet temperature.

On the other hand, NTU is a direct measure of the surface area of the heat transfer; consequently, the size of the heat exchanger is proportional to the NTU [35].

Fig.1. Oil cooler-Type: OKG 33/244

Accordingly, the ε-NTU approach is chosen in the proposed work. It can estimate outlet temperatures without requiring a numerical iterative solution to the nonlinear equations system.

The heat exchanger's size and heat transfer rate can be measured by the number of thermal units (NTU) using Eq. 15 [36]. All symbols are listed in table [7].

$$NTU = \frac{A_s * U}{C_{min}} \tag{15}$$

The heat capacity ratio C_r is measured according to Eq. 16 [37].

$$C_r = \frac{C_{min}}{C_{max}} \tag{16}$$

The effectiveness ($\mathcal{E}$) is calculated according to Eq. 17 [37].

$$\mathcal{E} = 2 \times \left[1 + C_r + \left(1 + {C_r}^2\right)^{1/2} \times \left[\frac{1 + exp\ \left(-NTU\left(1 + {C_r}^2\right)^{1/2}\right)}{1 - exp\ \left(-NTU\left(1 + {C_r}^2\right)^{1/2}\right)}\right]\right]^{-1} \tag{17}$$

The following equation gives the heat exchange rate between cold and hot currents [13].

$$T_{h,o} = T_{h,i} - \varepsilon\left(T_{h,i} - T_{c,i}\right) \quad and \quad T_{c,o} = \varepsilon\left(T_{h,i} - T_{c,i}\right) + T_{c,i} \tag{18}$$

Where C_{min}=C_h

The transfer rate (heat duty) (q_C=q_h) are calculated as follows [13] :

$$q_c = \dot{m_c} * C_{p,c} * \left(T_{c,o} - T_{c,i}\right) \quad \& \quad q_h = \dot{m_h} * C_{p,h} * \left(T_{h,i} - T_{h,o}\right) \tag{19}$$

$$T_{h,o} = -\frac{q_c}{\dot{m_h} * C_{p,h}} + T_{h,i} \tag{20}$$

$$T_{c,o} = \frac{q_h}{\dot{m_c} * C_{p,c}} + T_{c,i} \tag{21}$$

3. OPTIMIZATION TECHNIQUE

GAs are parameter search procedures for artificial systems based on the mechanics of natural genetics [38]. The GA methodology is used to solve optimization problems by performing a stochastic search of the solution space using strings of integers representing the optimized parameters, known as chromosomes. For these modeling applications, each integer within these chromosomes is referred to as a gene, and each gene has a decimal value between 0 and 9 [39]. They begin with a population, a collection of solutions (represented by chromosomes). Next, a population's solutions are taken and used to create a new population. This is driven by Darwinian survival of the fittest and a structured random exchange of information using reproduction, crossover, mutation, and permutation operators. First, solutions (parents) are chosen to create new solutions (offspring), which are chosen based on their fitness—the more fit they are, the better their chances of reproducing. This is repeated until certain conditions are met, such as the number of generations or the improvement of the best solution [38].

This paper uses the GA to solve the optimization design problem for the STHX with a single tube pass. The original design data and the original data for shell and tube are given in Table 1 and Table 2, respectively.

Table 1. Original design data of oil cooler type OKG 33/244 under study

Data of cold fluid		Data of hot fluid	
$T_{c,i}$	30 (C°)	$T_{h,i}$	60 (C°)
$T_{c,o}$	35 (C°)	$T_{h,o}$	45 (C°)
Cp_c	4/86 (kJ/kg k)	Cp_h	2035 (kJ/kg k)
k_t	0.613 (w/m k)	k_c	141 * 10^{-3}
m_c	245.6197 (kg/s)	m_h	169.0715 (kg/s)
ρ_t	1000 (kg/m$^{3)}$	ρ_t	865.8 (kg/m^3)
μ_t	855 * 10^{-6} Pa.s	μ_t	8.36 * 10^{-2} Pa.s
Pr_t	5.83	Pr_s	1205

Table 2. Original data for shell and tube

Data of cold fluid		Data of hot fluid	
n_t	420	D	0.62 (m)
$d_{t,i}$	0.019 (m)	n_p	1
$d_{t,o}$	0.0192 (m)		
S_t	0.0021 (m)		
L	3.050 (m)		
P_t	0.005 (m)		

The range of design variables of the geometric parameters is given in Table 3. The GA employs the Roulette Wheel Selection method. The probability of being chosen increases with increasing fitness. To create the offspring population, uniform crossover and random uniform mutation are used. With a probability of 0.85, the integer-based uniform crossover operator switches each corresponding binary bit between two different parents. After crossover, the mutation operator modifies each binary bit with a 0.01 mutation probability [40].

In MATLAB a first generation of 30 individuals is generated. There are three genes on each chromosome: tube inside diameter, shell diameter, and tube length. These three gens are binary coded. The range of each chromosome is shown in Table 3.

The roulette method is used for selection. According to most scholars, crossover and mutation probabilities are 0.85 and 0.01, respectively [40-41].

Table 3. The range of design variables of the geometric parameters

Data of input			
Variable	Original design	Range	
		From	to
$d_{t,i}$	0.015 (m)	0.011 (m)	0.02 (m)
D	0.62 (m)	0.58 (m)	0.67 (m)
L	3.050 (m)	2.65 (m)	3.55 (m)

3.1 CASE STUDY

The known information of the STHX, six design variables (tube inside diameter ($d_{t,i}$), tubes length L_t, shell diameter (D), the number of tubes (n_t), effectiveness (ε), and the output temperature of the hot fluid ($T_{h,o}$) ˚C) are selected and listed in Table (4).

Table 4. The six chosen variables to improve oil cooler type OKG 33/244 understudy.

Data of input		Data of output	
$d_{t,i}$	tube inside diameter (m)	n_t	number of tubes
L	tubes length (m)	ε	effectiveness
D	shell diameter (m)	$T_{h,o}$	Output temperature of hot fluid (C°)

The flowchart for the proposed implementation steps of GA solving the STHX design problem is shown in Fig. 2 as follows:

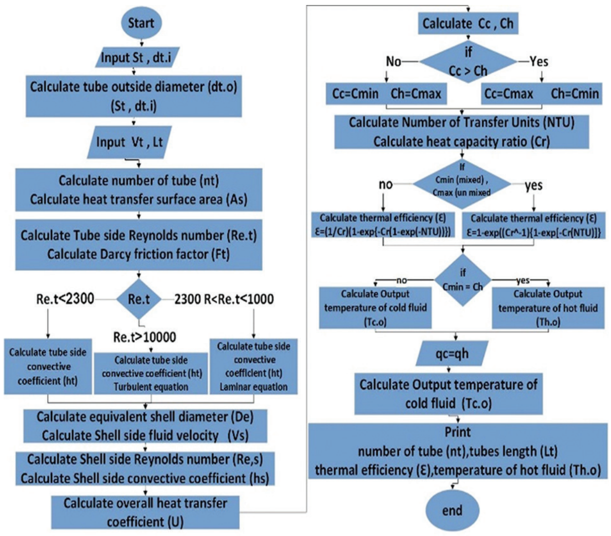

Fig.2. Flowchart for the proposed steps of GA solving design problem of STHX

The proposed implementation steps of GA solving the STHX design problem can be summarized as shown in Algorithm1:

Algorithm1:

Start

Input: m_c, $d_{t,i}$, S_t, Rf_t, n_p, Rf_s.

1. Calculate tube outside diameter $d_{t,o}=d_{t,i}+S_t$
2. Calculate the number of tubes $n_t = \frac{\dot{m}_c}{\rho_c * \frac{\pi}{4} * d_{t,i}^2 * \mathcal{V}_t}$
3. Calculate heat transfer surface area of tubes $A_s = \pi * d_{t,o} * l_t * n_t * n_p$
4. Calculate the Tube side Reynolds number $Re_t = \frac{\rho_t \mathcal{V}_t d_{t,i}}{\mu_t}$
5. Calculate Darcy friction factor $f_t=(1.82 \log_{10} Re_t-1.64)^{-2}$
6. Calculate tube side convective coefficient (h_t)

If $Re_t < 2300$ then

$$h_t = \frac{k_t}{d_{t,i}}\left[3.657 + \frac{0.0677\{Re_t Pr_t (d_{t,i}/l)\}^{1.33}}{1+0.1Pr_t\{Re_t(d_{t,i}/l)\}^{0.3}}\right]$$

Else if $2300 < Re_t < 1000$

$$h_t = \frac{k_t}{d_{t,i}}\left[\frac{(f_t/8)(Re_t-1000)Pr_t}{1+12.7(f_t/8)1/2\left(Pr_t^{2/3}-1\right)}\left(1+\frac{d_{t,i}}{l}\right)0.67\right]$$

Else if $Re_t > 10000$

$$h_t = 0.027\frac{k_t}{d_{t,i}} Re_t^{0.8} Pr_t^{1/3}\left(\frac{\mu_t}{\mu_{tw}}\right)^{0.14}$$

End if

7. Calculate equivalent shell diameter $D_e = \left(\frac{4*A}{P}\right)$
8. Calculate Shell side fluid velocity $\mathcal{V}_s = \frac{\dot{m}_h}{\rho_s \frac{\pi}{4} D_e^2 n_t}$
9. Calculate Shell side Reynolds number $Re_s = \frac{\rho_s \mathcal{V}_s D_e}{\mu_s}$
10. Calculate Shell side convective coefficient $h_s = 0.36\frac{k_t}{D_e} Re_s^{0.55} Pr_s^{1/3}\left(\frac{\mu_s}{\mu_{sw}}\right)0.14$
11. Calculate the overall heat transfer coefficient $U = \frac{1}{\frac{1}{h_s}+R_{fs}+\left(\frac{d_{t,o}}{d_{t,i}}\right)\left(R_{ft}+\frac{1}{h_t}\right)}$
12. Calculate $C_c=\dot{m}_c * C_{p,c}$, $C_h=\dot{m}_h * C_{p,h}$

$C_{max} = \max(C_c, C_h)$

$C_{min} = \min(C_c, C_h)$

13. Calculate the Number of Transfer Units $NTU = \frac{A_s * U}{C_{min}}$
14. Calculate heat capacity ratio $C_r = \frac{C_{min}}{C_{max}}$
15. Calculate effectiveness (Ɛ)

If C_{min} (mixed), C_{max} (unmixed) then

$$\varepsilon = 1 - \exp\left(-C_r^{-1}\{1-\exp[-C_r(NTU)]\}\right)$$

Else C_{min} (unmixed), C_{max} (mixed)

$$\varepsilon = \left(\frac{1}{C_r}\right)(1\text{-}\exp\{-C_r[1-\exp(-NTU)]\})$$

End if

16. Calculate Output temperature $T_{h,o}$ and $T_{c,o}$

If $C_{min} = C_h$ then

$T_{h,o} = T_{h,i} - \varepsilon\,(T_{h,i} - T_{c,i})$

Else

$T_{c,o} = \varepsilon(T_{h,i} - T_{c,i}) + T_{c,i}$

End if

17. Heat duty ($q_c=q_h$)

$q_c = \dot{m}_c * C_{p,c} * (T_{c,o} - T_{c,i})$ and

$q_h = \dot{m}_h * C_{p,h} * (T_{h,i} - T_{h,o})$

$T_{h,o} = -\frac{q_c}{\dot{m}_h * C_{p,h}} + T_{h,i}$ and $T_{c,o} = \frac{q_h}{\dot{m}_c * C_{p,c}} + T_{c,i}$

End

4. RESULTS AND DISCUSSION

MATLAB (version: R2018a(9.4.0.813654)) genetic algorithm toolbox is used to solve the optimization problem described. This section compares the effectiveness of the original oil cooler type OKG 33/244 with the proposed design. The results of the effectiveness of STHX are presented in Figures 3-6, where the relationship between the number of individuals in populations and the effectiveness has been plotted for each generation (1, 10, 20, 30). By applying GA to the presented case study discussed in section 3.1, the heat transfer coefficient should exceed 0.5 (of the original design) for success.

In Figure 3, the effectiveness of generation number 10 has been compared with that of generation number 1. The figure shows that the effectiveness has improved in generation 10 compared to generation 1. However, in the first generation, effectiveness suffers from a wide range (0.31268 to 0.72207). At the same time, the effectiveness in the tenth generation falls between 0.65804 and 0.7432 and mostly between 0.73142 and 0.7432.

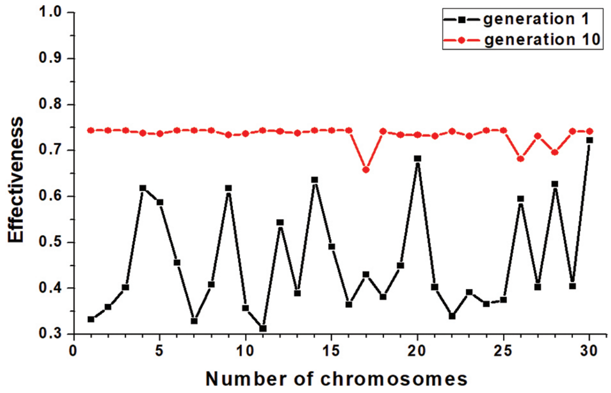

Fig. 3. Number of chromosomes in populations versus effectiveness for generations 1 and 10

Figure. 4 shows the effectiveness of generation 20 compared to the first generation. Figure 4 clearly illustrates that the effectiveness in generation 20 has improved compared to generation 1. Generation 20 has average effectiveness in the range of 0.73293 - 0.76883.

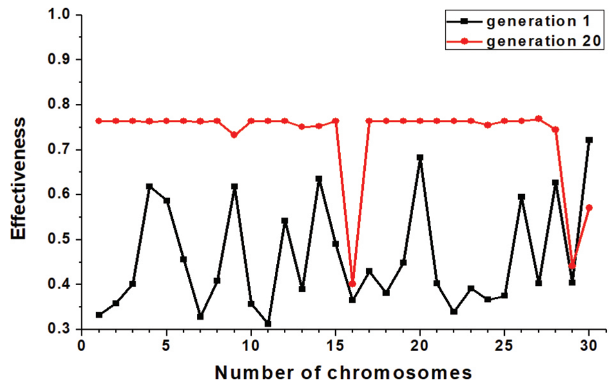

Fig. 4. Number of chromosomes in populations versus effectiveness for generations 1 and 20

The effectiveness of generation number 30 compared to the first generation is shown in Figure 5 as well, Generation 30 has effectiveness in the range of 0.73334 - 0.76923.

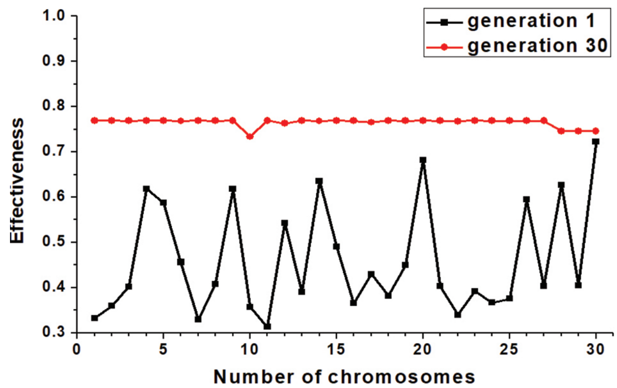

Fig. 5. Number of chromosomes in populations versus effectiveness for generations 1 and 30

Figure 6 proves the proposed claim for effectiveness improvement over generations.

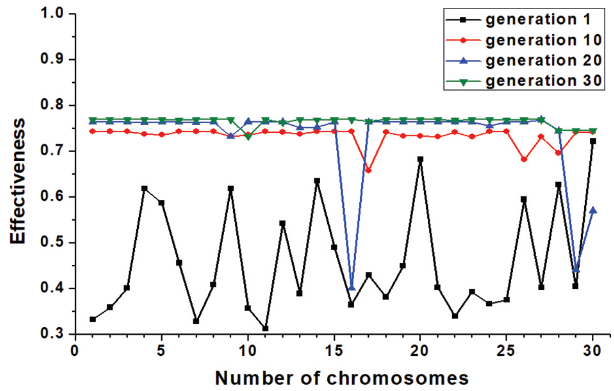

Fig. 6. Number of chromosomes in populations versus effectiveness for generations 1, 10, 20, and 30

Figure. 7 shows the relationship between the best effectiveness value for each generation. The effectiveness significantly improves through generations number 1 to 20. While the effectiveness in generation number 30 shows a slight increase in effectiveness compared to generation number 20.

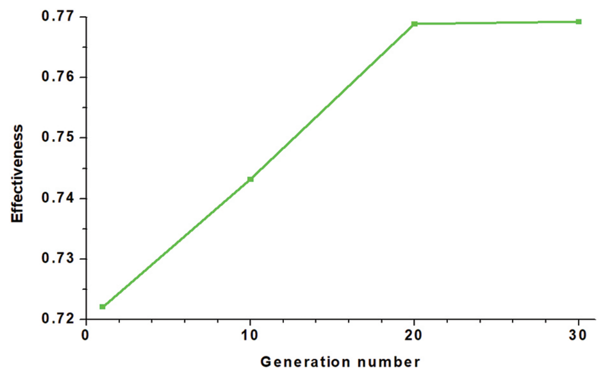

Fig. 7. The best value of effectiveness in each generation

Figure 8 illustrates the best point in each run of 50 runs; the robustness of the proposed method is illustrated due to the obvious convergence of best points in the range of (0.73456 to 0.78458).

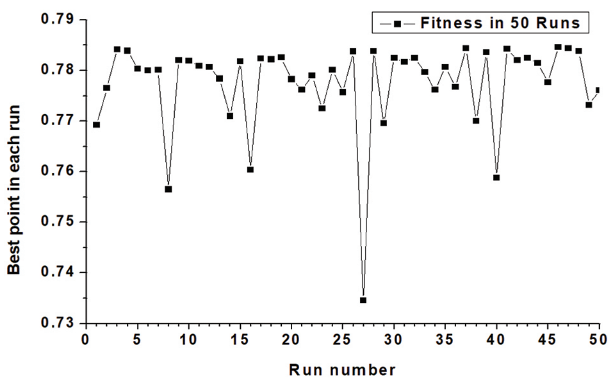

Fig. 8. Best fitness in 50 Runs

The simulation results using GA for design optimization of an oil cooler type OKG 33/244 reveal that the performance of STHX has been improved, indicating that GA can be successfully employed for solving the design optimization problem of a STHX from a performance point of view. Using GA, a maximum increase in the effectiveness of 57% and a minimum increase in the effectiveness of 47% have been achieved. Furthermore, the average effectiveness of the presented oil cooler type OKG 33/244 has improved by 55%, and NTU improved from 0.475319 to 1.825664 using GA. The details for optimal primitive and effectiveness for maximum and minimum fitness in 50 runs are reported in Tables 5 and 6, respectively.

Table 5. Details of the maximum fitness in 50 runs

Optimal primitive			effectiveness	NTU
$d_{t,i}$	L	D	0.784583	1.825664
0.011	3.549969	0.58		

Table 6. Details of the minimum fitness in 50 runs

Optimal primitive			effectiveness	NTU
$d_{t,i}$	L	D	0.734555	1.535565
0.012062	3.549977	0.58		

5. CONCLUSION

The design optimization of an STHX is developed from a performance point of view by using GA to achieve the optimal configuration. The objective is to develop and test a model (Fig.1) for optimizing the early design stage of the STHX, which is quick. GA is implemented to maximize heat transfer rate while minimizing pressure drop. The GA is applied to the oil cooler type OKG 33/244. By comparing the results obtained using GA with the original data of the STHX, the following conclusions are obtained:

- A maximum increase in the effectiveness of 57% was achieved using GA.
- A minimum increase in the effectiveness of 47% was also achieved. Furthermore, the heat exchanger's average was 55% by using GA.
- NTU improved from 0.475319 to 1.825664
- Finally, the simulation outcomes reveal that the STHX's operating performance has been improved, indicating that GA can be successfully employed for design optimization of a STHX from a performance standpoint.

In the future investigate, the simultaneous approach using GA on a larger scale (heat exchanger network) can be employed to improve overall plant performance.

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7. APPENDIX

Table 7.

Nomenclature

Abbreviations:

A	surface area (m^2)	n_p	number of tubes passages
A_s	heat transfer surface area of tubes (m^2)	Pr_s	shell side Prandtl number
Cp_c	Heat capacity of cold fluid $(kJ/kg\ k)$	Pr_t	tube side Prandtl number
Cp_h	Heat capacity of hot fluid $(kJ/kg\ k)$	P_t	tube pitch (m)
C_c	Heat capacity of cold fluid $(kJ/kg\ k)$	q_c	heat duty of cold fluid (w)
C_h	Heat capacity of hot fluid $(kJ/kg\ k)$	q_h	heat duty of hot fluid (w)
C_{max}	Maximum of heat capacity $(kJ/kg\ k)$	Re_s	Shell side Reynolds number
C_{min}	minimum of heat capacity $(kJ/kg\ k)$	Re_t	Tube side Reynolds number
C_r	heat capacity ratio	Rf_s	shell side fouling resistance $(m^2\ k/w)$
D	shell diameter (m)	Rf_t	shell side fouling resistance $(m^2\ k/w)$
D_e	equivalent shell diameter (m)	$T_{c,i}$	Input temperature of cold fluid (C^o)
$d_{t,i}$	tube inside diameter (m)	$T_{c,o}$	Output temperature of cold fluid (C^o)
$d_{t,o}$	tube outside diameter (m)	$T_{h,i}$	Input temperature of hot fluid (C^o)
ε	effectiveness	$T_{h,o}$	Output temperature of hot fluid (C^o)
f_t	Darcy friction factor of tube	U	overall heat transfer coefficient $(w/m^2\ k)$
f_s	Darcy friction factor of shell	S_t	thickness of tube (m)
h_t	tube side convective coefficient $(w/m^2\ k)$	v_s	Shell side fluid velocity $(\ m/s)$
h_s	Shell side convective coefficient $(w/m^2\ k)$	v_t	tube side fluid velocity $(\ m/s)$
k_t	thermal conductivity $(w/m\ k)$	μ_s	shell side dynamic viscosity $(Pa\ s)$
L_t	tubes length (m)	μ_t	Tube side dynamic viscosity $(Pa\ s)$
m_c	Mass flow rate of cold fluid (kg/s)	ρ_s	Shell side fluid density (kg/m^3)
m_h	Mass flow rate of hot fluid (kg/s)	ρ_t	tube side fluid density (kg/m^3)
NT	Number of Transfer Units	π	numerical constant
n_t	number of tube		