Scientia Iranica B (2013) 20(5), 1445–1454

Sharif University of Technology
Scientia Iranica
Transactions B: Mechanical Engineering
www.scientiairanica.com

Spiral heat exchanger optimization using genetic algorithm

M. Bidabadi, A.K. Sadaghiani and A. Vahdat Azad*

School of Mechanical Engineering, Iran University of Science and Technology, Tehran, P.O. Box 16846, Iran.

Received 15 September 2011; received in revised form 18 April 2013; accepted 8 June 2013

KEYWORDS
Heat transfer;
Spiral heat exchanger;
Multi-objective;
Optimization;
Genetic algorithm.

Abstract. This paper investigates optimization methods based on Genetic Algorithms (GAs) for spiral heat exchangers. The purpose of designing heat exchanger depends on its application and could be total cost, heat transfer coefficient or both of them. The current targeting methods identify optimum points from both economic and thermodynamic views, and capture a trade-off between two objectives. Optimizations, using single objective functions, are performed in order to investigate parameter behavior in two different applications of SHEs. Also, this work takes care of numerous geometric parameters in the presence of logical constraints. Multi-objective and weighted function optimizations, using genetic algorithm, are developed in order to obtain a set of geometric design parameters, which lead to minimum pressure drop and the maximum overall heat transfer coefficient. Optimized heat transfer coefficient, compared to its first value at basic design, had a 13% increase, and total cost in optimized case presents 50% reduction compared to the basic design. Also in trade-off cases, heat transfer coefficient and total cost have been improved up to 60% increment and 20% reduction, respectively. Therefore, designing heat exchanger, using presented optimal methods in this research, are proposed as useful methods for designers, engineers and researchers.

1. Introduction

Heat exchanger is a device used for effective heat transfer between two fluids (gas or liquid) from one to another. Heat exchangers are used in various industries including HVAC, plants, power and process industries and many others [1]. Spiral heat exchangers have proved to be more efficient and reliable than other types of heat exchangers. The Spiral-Plate Heat Exchanger (SHE) may be one exchanger selected primarily on its virtues and not on its initial cost. SHEs offer high reliability and on-line performance in many severely fouling services such as slurries [2]. Spiral plate heat exchangers are ideal for cooling slurries and viscous fluids. This type of exchanger is common in the paper, petrochemical, food and sugar industries with applications in evaporation and condensation [3]. In the main application of spiral heat exchangers, hot and cold stream temperatures during the process are important. For example, the removal of water from food provides microbiological stability, reduces deteriorative chemical reactions, and reduces transportation and storage costs. Both evaporation and dehydration are methods used in the dairy industry for this purpose.

To keep the product temperature low and constant and the difference in temperatures high, the heat exchanger is enclosed in a large chamber and transfers heat from the heating medium, usually low pressure steam, to the product usually via indirect contact surfaces.

An important feature of spiral plate exchangers that distinguishes them from other heat exchangers is

*. *Corresponding author.*
E-mail address: abazar.vahdat@gmail.com (A. Vahdat Azad)

their capacity to handle high viscosity and dirty fluids exhibiting lower tendency to fouling [4]. Spiral heat exchangers consist of two long plates rolled together, forming a spiral. Studs welded to the plates fix the spacing between the plates and provide mechanical strength [5]. Due to the particular geometry which creates a constant change in direction, local turbulence increases, thus eliminates fluid stagnant zones. Also, since there are no dead spaces in a SHE, the helical flow pattern produces high turbulence creating a self-cleaning flow passage [1].

Due to the widespread use of these equipment, their efficient design has been analyzed from different points of views, such as exergetic which was analyzed by Wu et al. [6]. Bes and Roetzel [7] developed an analytical rating study to determine the temperature profile within the plates; in their work constant overall heat transfer coefficients were assumed. They also studied the influence of various geometrical parameters in design and thermal performance. In the later work, Bes and Roetzel [8] developed a simple formula to determine the temperature difference correction factor that applies for any heat capacity rate ratio and for any number of turns. Egner and Burmeister [9] worked on a numerical study of spiral ducts of rectangular section, using computational fluid dynamics techniques, and determined the Nusselt number as a function of the Dean number, showing the strong dependence of the heat transfer coefficient upon the spiral radii. They demonstrated that except for the entry regions, the heat transfer coefficient is nearly constant, however, at entry regions, heat transfer coefficients may be even as 50% larger than the fully developed values. An important contribution of their work is the general conclusion for estimating the thermal entry length for laminar Reynolds numbers between 100 and 500. Burmeister [10] developed a more approximate solution to determine the thermal effectiveness versus the number of heat transfer units of this type of exchangers. A limitation of the method, however is that it applies only to cases where the heat capacity rates of the two fluids are equal. Empirical correlations for spiral heat exchangers were reported by Minton [11]. He presented a set of correlations for heat transfer coefficient and pressure drop for the laminar and turbulent regions based on average plate curvature. Martin [12] also reported empirical correlations for heat transfer and friction factor that cover a wide range of Reynolds numbers. Picon et al. [5], using the allowable pressure drop as a design objective, introduced a methodology for the preliminary sizing of a spiral heat exchanger. Heat exchanger design can be a complex task, therefore, advanced optimization tools proved to be useful to identify the best and the cheapest heat exchanger for a specific duty. Genetic Algorithms (GAs) are among the current options to perform such work [13].

In the present work, new methods, using genetic algorithms, are presented to find the optimal design parameters of a SHE within allowable pressure drops. Case studies reported by Minton [11] and Angelo A. Moretta [14] are presented below to demonstrate the capabilities of the new methods. In the current methods, it is necessary to specify a target point prior to the design of the heat exchanger. Since heat exchangers are important components of all thermal systems, their design should be adapted well to the applications in which they are used; otherwise their performance will be deceiving and their costs excessive. We performed design optimizations, considering importance of heat transfer coefficient or cost or both of them in different applications of heat exchangers. This was done in order to take a step towards optimizing important dimensions of SHE and to study thermodynamical limitations that a designer meets. The results are interesting even though the authors used simplified relations based on thermodynamic parameters and process data with a sufficient accuracy.

2. Spiral heat exchanger geometry

A thorough description of the geometry of spiral heat exchangers was published by Dongwu [15], where expressions to calculate spiral diameter, number of turns and length of the semicircles are provided. The main geometrical dimensions of spiral heat exchangers (plate spacing, spiral width and length) and operation manner are shown in Figure 1.

2.1. Heat transfer, pressure drop and cost function

Based on engineering judgment and consultation [16], the heat losses to the surroundings are assumed negligible. Also, physical properties of cold stream at 283°K and 288°K are those specified in [17]. Based on the first law of thermodynamics, the actual heat balance

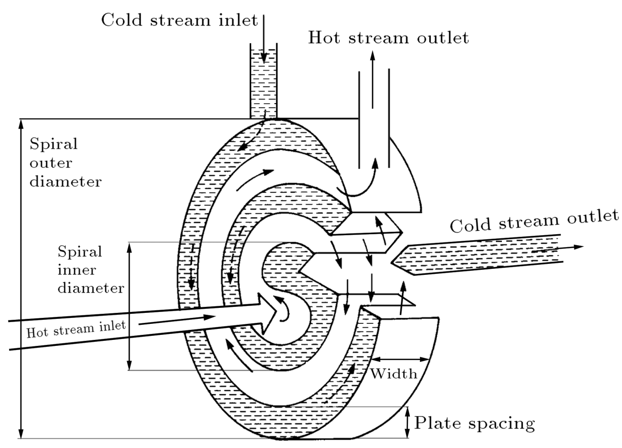

Figure 1. Geometrical features and flow pattern in spiral heat exchanger.

equation in a spiral plate heat exchanger is presented as:

$$Q = m_h c_h (T_{hi} - T_{ho}) = m_c c_c (T_{co} - T_{ci}). \tag{1}$$

Using the overall heat transfer coefficient, U, the heat transfer rate for heat exchanger is expresses as:

$$Q = UA\text{LMTD}. \tag{2}$$

The overall heat transfer coefficient, U, is calculated by the following expression [18]:

$$U = \frac{1}{\frac{1}{h_h} + \frac{t}{k_p} + \frac{1}{h_c} + R_f}. \tag{3}$$

The logarithmic mean temperature difference is determined using:

$$\text{LMTD} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}}. \tag{4}$$

Heat transfer area is calculated from the equation:

$$A = 2LH. \tag{5}$$

Morimoto and Hotta developed correlations to determine the film heat transfer coefficient in a spiral plate heat exchanger [19]:

$$\text{Nu} = .0239 \left(1 + 5.54 \frac{D_h}{R_m}\right) \text{Re}^{0.806} \text{Pr}^{0.268}, \tag{6}$$

where M is the mass flux and is defined as follows:

$$M = \frac{m}{A_c} = \frac{m}{HS}. \tag{7}$$

D_h is the average hydraulic diameter and is defined as:

$$D_h = \frac{(2HS)}{(H+S)}. \tag{8}$$

Velocity is calculated by dividing the mass flux by density:

$$V = \frac{M}{\rho}. \tag{9}$$

The spiral mean diameter is calculated from:

$$R_m = \frac{R_{\min} + R_{\max}}{2}. \tag{10}$$

The Reynolds number and Prandtl number are defined by Eqs. (11) and (12), respectively [20]:

$$\text{Re} = \frac{MD_h}{\mu}, \tag{11}$$

$$\text{Pr} = \frac{\mu C_p}{k}. \tag{12}$$

By calculating the Nusselt number from Eq. (7), heat transfer coefficient is obtained as follows:

$$h = \frac{k\text{Nu}}{D_h}. \tag{13}$$

Pressure drop in spiral heat exchanger is determined using the following expression [2]:

$$\Delta P = \frac{1.45(LV^2\rho)}{1.705 \times 10^3}. \tag{14}$$

The outside diameter of spiral is determined using the empirical equation as presented in [11]:

$$D_s = [1.28L(b_h + b_c + 2t) + C^2]^{1/2}. \tag{15}$$

The total cost is calculated by:

$$C_{\text{tot}} = C_i + C_{od}, \tag{16}$$

where C_i represents the manufacturing costs of the heat exchanger, and is obtained via Hall equation [21] as a function of the surface area, as follows:

$$C_i = a_1 A^b, \tag{17}$$

where $a_1 = 5873$ and $b = 0.59$ for heat exchanger made of titanium.

The second term in Eq. (16) represents the operational costs, involving overcoming the pressure loss caused by friction, evaluated using the following:

$$C_{od} = \sum_{k=1}^{ny} \frac{C_0}{(1+i)^k}, \tag{18}$$

$$C_0 = P \times C_E \times H, \tag{19}$$

$$P = \frac{1}{\eta}\left(\frac{m_h}{\rho_h} \times \Delta P_h + \frac{m_c}{\rho_c} \times \Delta P_c\right). \tag{20}$$

In the above equations, overall pumping efficiency is taken as 0.75, and an annual amount of work hours is $H = 8000$ h/yr. All values of discounted operating costs were computed with energy cost 0.12 €/kW.h, equipment life $ny = 15$ years, and annual discount rate $i = 10\%$.

3. Genetic algorithm

Genetic algorithms are stochastic search methods that lead a population towards an optimum, using the principles of evolution and natural genetics. With the proper encoding, they can manipulate integer or continuous variables and they can handle linear and non-linear constraints. Genetic algorithms require little information of the problem itself. In this case, computations based on the algorithms are attractive to users. Moreover, it is well known that genetic algorithms have been successfully applied to many optimization problems [22]. In recent years, application of genetic algorithms in thermal engineering has received much attention for solving real-word problems [23]. For example, the fin-tube heat exchanger performance was predicted using a GA [24]. Plate-fin heat exchangers were optimized by means of GAs [25-27]. A new design method was proposed to optimize a shell-and-tube heat exchanger from an economic point of view by a GA [28]. Optimization of the geometry of cross-wavy and cross-corrugated primary surface recuperators was studied via GAs [29,30]. The coefficients of heat transfer correlations for compact heat exchangers were obtained using GAs, based on experimental data, and, in turn, these correlations were used to estimate their performance [31,32].

These reports suggested that GAs have a strong ability for searching, and can successfully optimize and predict thermal problems. Applications of GA in the field of thermal engineering are new challenges. At this point, the GA technique may be used in the optimization design process in order to obtain optimal results under a specified heat duty within allowable pressure drops. In this work, GA is used to optimize a spiral heat exchanger and to design a low-cost, high heat transfer coefficient and trade-off between them. We use a computer program in MATLAB software, finding optimum value for each case.

The sequence of different operations of GA is shown in Figure 2. In this algorithm, optimized fitness functions are heat transfer coefficient and total cost. Heat transfer coefficient, total cost, multi-objective and weighted functions defined in MATLAB program are objective functions. Inputs are spiral length and width, plate spacing in hot and cold stream and outer diameter of heat exchanger. Table 1 presents the decision variables, their lower and upper bounds and their optimum values in both cases.

Also, for better comparison between various designs, heat transfer rate between hot and cold sides, inlet and outlet temperature and flow rates have been considered constant and physical aspects of heat exchanger considered as variables during the optimization procedure.

In this study, four non-linear constraints including pressure drops in both hot and cold streams, amount of heat load applied to streams and relation between outer diameter has been considered and exerted to MATLAB program.

1. Hot stream pressure drop constraint which is:

$$\Delta P_h - \frac{1.45LV_h^2\rho_h}{1705} = 0.$$

2. Cold stream pressure drop constraint:

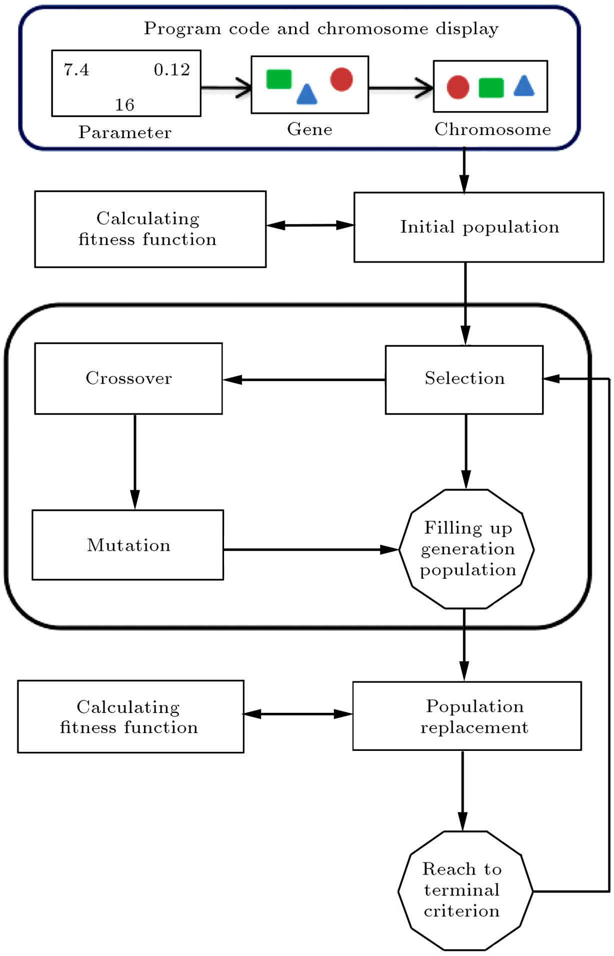

Figure 2. Block diagram representation of genetic algorithms (GAs).

$$\Delta P_c - \frac{1.45LV_c^2\rho_c}{1705} = 0.$$

3. Outer diameter constraint:
$$D_s^2 - [C^2 + (15.36L(b_h + b_c + t))] = 0.$$

4. Heat load constraint:
$$Q - (2ULH\text{LMTD}) = 0.$$

Genetic algorithm is considered with a population of 50 and 100 numbers of iterations. The program with calculating fitness valu in 50 random points as genes or individuals enters into the algorithm. After comparing fitness values in parents, children are produced with random changes on one parent or both parents' vectors. In the next generation, people who were more qualified or who have more fitness values are selected as parents. Algorithm stops when one of the stopping conditions, for example stall generation is achieved.

GAs are probabilistic, and therefore two runs of the GA with the exact same settings could lead to two different results. Therefore, for each case, the GA optimization was performed five times. The best result among these runs was retained.

Table 1. Decision variables, their lower, upper and optimum values.

Value	Minimum	Maximum	Case [14]		Case [11]		
			Coefficient optimization	Cost optimization	Multi-objective function	Weighted objective	Coeff.
Pressure drop in hot stream	0 bar	172 bar	154	12	0.9240	0.4576	7.700
Pressure drop in cold stream	0 bar	172 bar	37	0.889	0.9768	0.5720	10.920
Outer diameter	0.5 m	1.5 m	0.6895	0.726	0.8100	0.6465	0.7952
Length	5 m	22 m	6.080	5.933	20.05335	20.4275	21.5444
Width	0.05 m	2.3 m	0.634	2.288	0.4499	0.4858	0.1561
Plate spacing, hot side	0.005 m	0.032 m	0.0069	0.0073	0.0051	0.0068	0.0053
Plate spacing, cold side	0.005 m	0.032 m	0.0317	0.0311	0.0063	0.0078	0.0057

4. Discussion

The main hypothesis of this research is to demonstrate how powerful and effective tool GA is in optimizing physical and thermodynamically aspects of heat transfer equipment. The aim of designing heat exchanger could be total cost during equipment life or maximum heat transfer between hot and cold streams or a trade-off between them, and depends on the application of heat exchanger.

In industries such as the food processing industries, the maximum heat transfer rate in heat exchanger is more important than its cost. In such cases, the designer attempts to maximize heat transfer rate by disregarding total cost. On the other hand, in some applications, the designer, considering purchasing power of customer, chooses cost as the objective function. In other cases, both heat transfer and total cost are important. To develop our study, we investigate GA optimizations in all three cases.

In our cases, titanium has been considered as the material of construction of heat exchanger. Due to Titanium's super strong corrosion resistant ability, major industrial application for titanium remains in heat transfer applications in which the cooling medium is seawater, brackish water or polluted water. The initial cost of titanium in this application is not only competitive at the time of installation, but over time, the life cycle costs are lower as a result of lower maintenance costs.

The GA optimizations studied in this paper are performed on heat transfer coefficient and total cost individually and simultaneously in the first and second case studies, respectively.

The first case study is a heat exchanger for cooling a non-Newtonian fluid, taken from [14]. The average physical properties for this case study are shown in Table 2.

GA results obtained in heat transfer coefficient optimization show that pressure drop in heat exchanger reaches its maximum values in feasible region, which causes maximum inlet pressure and brings about velocity and Reynolds number increment. Consequently, heat transfer rate increases. In heat transfer coefficient optimization, despite a 13 percent increase in heat transfer coefficient, total cost has almost 30 percent increment compared to the basic design [14].

Total cost is a composition of capital and operational costs, therefore when cost is chosen as the objective function, the algorithm incorporates both costs simultaneously. Note that larger dimension causes more heat transfer area and capital cost, the algorithm chooses an optimal length for heat exchanger to prevent increase of heat transfer area. On the other hand, total cost optimization needs to reduce pressure drops in both hot and cold streams. So, algorithm optimizes total cost by increasing plate spacing. Maximum flow area is obtained by increasing the plate spacing to its maximum value. It causes smoother flow movement and pressure drop reduction. When total cost is defined as the objective function, cost has 55 percent reduction which is a remarkable amount by contemplating energy value, and heat transfer coefficient is half of the value in basic design.

In heat transfer coefficient optimization process, GA led to significant increase up to 1.280×10^3 (W/m^{2}K), although the heat exchanger coefficient in basic design [14] is 1.113×10^3 (W/m^{2}K). Also when

Table 2. Stream data for case studies [11,14].

Average physical properties	First case study [14]		Second case study [11]		Unit
	Hot stream	Cold stream	Hot stream	Cold stream	
Mass flow rate	127.75	18.92	0.7833	0.7444	kg.s^{-1}
Inlet temperature	298.60	273.15	473.2359	333.1500	K
Outlet temperature	298.15	285.92	393.1500	423.8258	K
Heat capacity	2.3768	8.4186	2973	2763	J kg^{-1} K^{-1}
Thermal conductivity	0.6231	0.5815	0.3480	0.3220	W m^{-1} K^{-1}
Density	1350	1000	843	843	kg m^{-3}
Relative density	1.35	1	0.843	0.843	-
Pressure drop	110.19	85.43	0.2004	0.3298	kPa
Velocity	4.4006	3.2597	0.1139	0.1461	m s^{-1}
Plate thickness	0.0032	0.0032	0.0031754	0.0031754	m
Inernal diameter	0.3048	0.3048	0.203	0.203	m
Plate spacing	0.0318	0.0063	0.00635	0.00635	m

cost is chosen as an objective function, optimal value is 22499 € whereas this amount in basic design is 44813 €. Figure 3 compares total cost and heat transfer coefficient for this case study and optimization method.

The proposed methodology in second case study is demonstrated using the case study reported by Minton [11]. The average physical properties are shown in Table 2. Since mentioned objectives are conflicting, no single solution can well-satisfy both objective functions simultaneously. In other words, any attempt to increase the value of the total rate of heat transfer leads to higher total cost of the system, which is certainly undesirable. Therefore, multi-objective and weighted optimizations, using genetic algorithms, are utilized in order to achieve a set of optimal solutions, each of which are trade-off between objectives, and can satisfy both objective functions in an appropriate level. The main advantage of this work is providing a set of optimal solutions each of which can be selected by the designer based on the project's limits and the available investment.

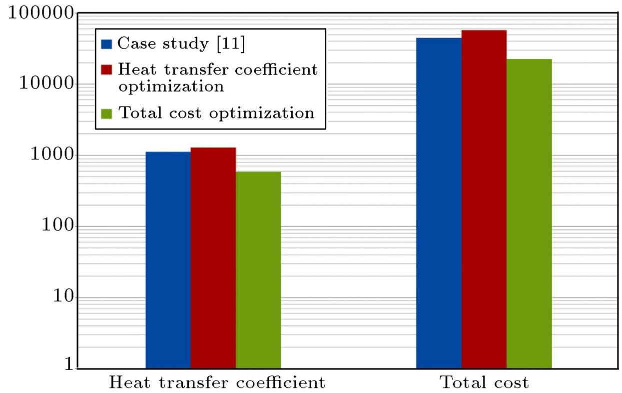

Figure 3. Geometrical features and flow pattern in spiral heat exchanger.

In order to compare the results obtained by weighted function with multi-objective function, we defined equal weighted factors (between 0 and 1) for heat transfer coefficient and total cost.

Pressure drop and heat transfer coefficient are interdependent quantities, and both of them essentially influence the total cost and heat transfer rate of any heat exchanging system. It is necessary to suggest such dimensions which result acceptable pressure drop and heat transfer rate under given conditions.

Using weighted factors in GA weighted optimization helps the designer to design more effective heat exchanger for a specific application. In such cases, the designer by defining weighted factors, based on importance of each one of heat transfer and total cost, calculate optimal dimensions.

It is obvious that when program uses multi-objective function rather than weighted objective function, estimated dimensions have better results at all. It is because when multi-objective function is chosen as objective function, program itself calculates a trade-off between two goals (coefficient and cost) and defines optimal parameters.

By defining another objective function in second case study, we tried to demonstrate the differences between a single optimization, where heat transfer coefficient is chosen as fitness function and multi-objective optimization, where both heat transfer coefficient and total cost are selected as fitness functions. Results show that when heat transfer coefficient is chosen as the objective function, the obtained optimized value is 508 (W/m^{2}K), although this amount in basic design [14]

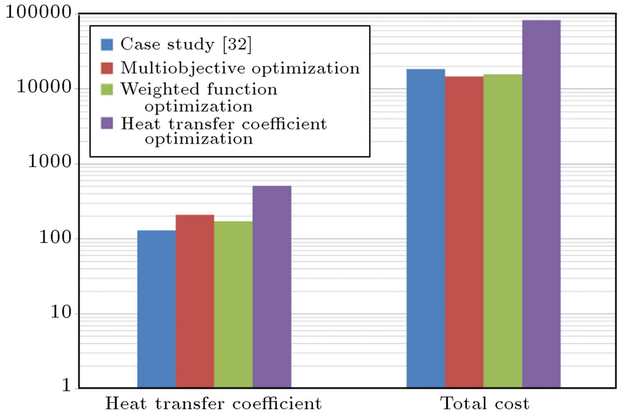

Figure 4. Geometrical features and flow pattern in spiral heat exchanger.

is 130 (W/m^2K). Figure 4 compares total cost and heat transfer coefficient for case study and optimization methods.

In weighted and multi-objective optimizations, both heat transfer coefficient and total cost have better results up to 60% increment and 20% reduction, respectively in comparison to basic design. Algorithm by increasing pressure drops up attempts to improve heat transfer coefficient. Also GA chooses maximum length from its feasible region. It can be seen that HE width in optimized cases are almost half of this amount in basic design.

It seems that increasing heat transfer area and pressure drops in cold and hot streams results in increasing the heat transfer rate.

In heat transfer coefficient optimizations, although the algorithm chooses longer length, it selects shorter width to raise pressure drops and Reynolds number. It can be seen from the results that shorter width offers best heat transfer coefficient in all optimizations and basic case studies, and this results in smaller hydraulic diameter. It is worth mentioning that in heat transfer coefficient optimizations, the algorithm selects width and length of heat exchanger regarding logical constraints.

Also, when cost is chosen as the objective function, the algorithm considers an acceptable width and length for the heat exchanger, which are small enough to prevent heat transfer area enlargement and more pressure drop. Regarding that the life time of these equipments are considered to be 15 years, from the results, it can be concluded that in total cost optimization, operational costs are more important than capital costs. This means that the algorithm calculates the dimensions based on the minimum pressure drop. Calculated parameters in optimizations are presented in Figures 5 and 6.

The targeting methods are evaluated by comparing their target results from predicted dimensions with the performance obtained from other approaches. It

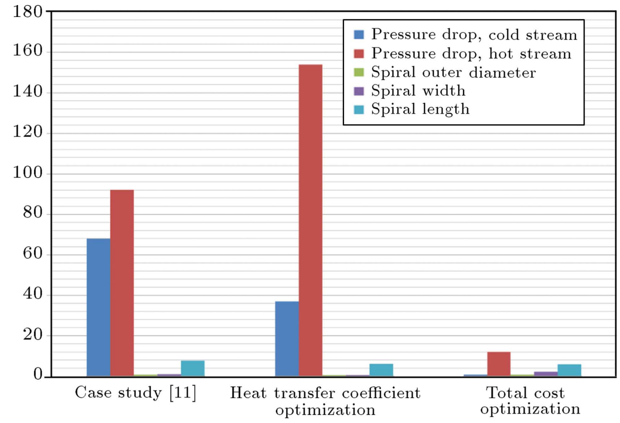

Figure 5. Calculated parameters in first case optimization.

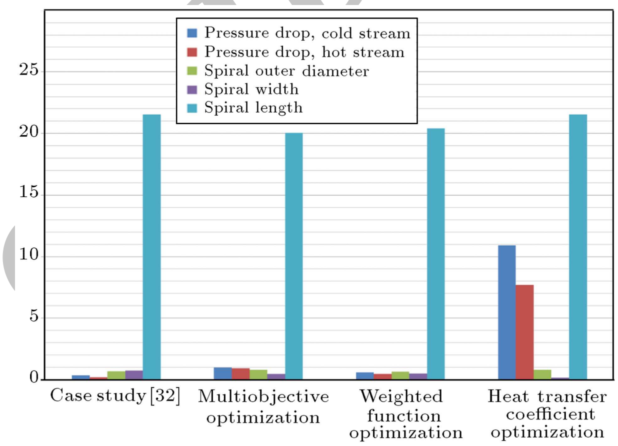

Figure 6. Calculated parameters in second case optimization.

is clear from results that GA is an effective tool for optimization in thermal sciences. Regarding that the basic designs are a step by step guidance to calculate the size of SHEs, optimal results are not expected. Tables 3 and 4 present case studies decision variables and results obtained from optimizations.

5. Conclusion

In this paper, spiral heat exchangers have been optimized using genetic algorithms. Not only calculated dimensions for spiral heat exchangers by considering heat transfer coefficient and total cost illustrate better results, but also the obtained higher heat transfer and lower cost values by defining weighted and multi-objective functions, in comparison to basic designs, proved that GA is an effective and powerful tool in optimization. Optimized heat transfer coefficient, compared to its first value at basic design, had a 13% increment, and total cost in optimized case presents 50% reduction, compared to the basic designs. Also, in trade-off cases, heat transfer coefficient and total cost

Table 3. Decision variables and results obtained from the first case study [14] optimization.

Case study	Heat transfer coeff. optimization	Total cost optimization	Unit
$\Delta Pcs = 85.43$	$\Delta Pcs = 37$	$\Delta Pcs = 0.889$	kPa
$\Delta Phs = 110.19$	$\Delta Phs = 154$	$\Delta Phs = 12$	kPa
$Ds = 0.849$	$Ds = 0.6895$	$Ds = 0.726$	m
$H = 0.9144$	$H = 0.634$	$H = 2.288$	m
$L = 7.817$	$L = 6.080$	$L = 5.933$	m
$U = 1.113E + 003$	$U = 1.280E + 003$	$U = 586$	W/m^{2}K
$Ct = 4.4813E + 004$	$Ct = 5.7719E + 004$	$Ct = 2.2491 + 004$	€

Table 4. Decision variables and results obtained from the second case study [11] optimization.

Case study	Multi-objective optmization	Weighted optimization	Heat transfer coeff optimization	Unit
$\Delta Pcs = 0.3298$	$\Delta Pcs = 0.9768$	$\Delta Pcs = 0.5720$	$\Delta Pcs = 10.920$	kPa
$\Delta Phs = 0.2004$	$\Delta Phs = 0.9240$	$\Delta Phs = 0.4576$	$\Delta Phs = 7.700$	kPa
$Ds = 0.69$	$Ds = 0.8100$	$Ds = 0.6465$	$Ds = 0.7952$	m
$H = 0.73$	$H = 0.4499$	$H = 0.4858$	$H = 0.1561$	m
$L = 21.53$	$L = 20.05335$	$L = 20.4275$	$L = 21.5444$	m
$U = 130$	$U = 208.8328$	$U = 172.2879$	$U = 508.5298$	W/m^{2}K
$Ct = 1.8446E + 004$	$Ct = 1.4782E + 004$	$Ct = 1.5633E + 004$	$Ct = 8.30711E + 004$	€

have been improved up to 60% increment and 20% reduction, respectively. The results of this study are reported for the first time and present evidence from several outcomes, proving GA as a powerful tool and useful method for designers, engineers and researchers.

Nomenclature

A	Heat-transfer surface area, m^2
b	Channel spacing, m
b_h	Spacing of the hot side, m
b_c	Spacing of the cold side, m
C	Core diameter, m
c_c	Specific heat of cold fluid, J kg^{-1}°K^{-1}
C_E	Energy cost, €/kW h
c_h	Specific heat of hot fluid, J kg^{-1}°K^{-1}
C_i	Capital investment, €
C_{od}	Total discounted operating cost, €
C_o	Annual operating cost, €/yr
c_p	Specific heat of fluid, J kg^{-1}°K^{-1}
C_{Tot}	Total cost, €
D_H	Average hydraulic diameter of channel, m
D	Outside spiral diameter, m
H	Width of the plate, m
H	Annual operating time (h/yr)
h	Film heat-transfer coefficient, W m^{-2}°K^{-1}
h_h	Hot-side convection heat-transfer coefficient, W m^{-2}°K^{-1}
h_c	Cold-side convection heat transfer coefficient, W m^{-2}°K^{-1}
i	Annual discount rate
k	Thermal conductivity of fluid, W m^{-1}°K^{-1}
k_P	Thermal conductivity of the plate providing the heat-transfer surface, W m^{-1}°K^{-1}
L	Length of the plate, m
LMTD	Log Mean Temp. Difference, °K
M	Mass flux, kg m^{-2} s^{-1}
m	Mass flowrate, kg s^{-1}
Nu	Nusselt number, unitless
ny	Equipment life, year
P	Pumping power, W
Pr	Prandtl number, unitless
ΔP	Pressure drop, kPa
ΔP_h	Hot side pressure drop, kPa
ΔP_c	Cold side pressure drop, kPa

Q	Total heat transfer, W
$R_{\max}$	Maximum radius of spiral, m
$R_{\min}$	Minimum radius of spiral, m
Re	Reynolds number, unitless
R_f	Fouling factor, W^{-1} $m^{2\circ}K$
R_m	Mean spiral radius, m
t	Plate thickness, m
T_{ho}	Outlet temperature of hot fluid, °K
T_{hi}	Inlet temperature of hot fluid, °K
T_{co}	Outlet temperature of cold fluid, °K
T_{ci}	Inlet temperature of cold fluid, °K
U	Overall heat transfer coefficient, W $m^{-2\circ}K^{-1}$
V	Fluid mean velocity, m s^{-1}
ρ	Density, kg m^{-3}
ρ_h	Hot stream density, kg m^{-3}
ρ_c	Cold stream density, kg m^{-3}
η	Pumping efficiency
m_h	Mass flowrate of hot fluid, kg s^{-1}

Subscripts

h	Hot stream
c	Cold stream
min	Minimum value
max	Maximum value

References

1. Vahdat Azad, A. and Amidpour, M. "Economic optimization of shell and tube heat exchanger based on constructal theory", *Energy*, (**36**), pp. 1087-1096 (2011).
2. Perry, *Chemical Engineers' Handbook*, 7th Ed., McGraw-Hill, New York (1997).
3. Trom, L. "Use spiral plate exchangers for various applications", *Hydrocarbon Processing*, **74**(5), pp. 73-81 (1995).
4. Wilhelmsson, B. "Consider spiral heat exchangers for fouling application", *Hydrocarbon Processing*, July**(83)** (2005).
5. Picón-Núñez, M., Canizalez-Dávalos, L., Martínez-Rodríguez, G. and Polley, G.T. "Shortcut design approach for spiral heat exchanger", *Trans IChemE, Part C, Food and Bioproducts Processing*, **85**(C4), pp. 322-327 (2007).
6. Shuang-Ying Wu, Xiao-Feng Yuan, You-Rong Li and Lan Xiao "Exergy transfer effectiveness on heat exchanger for finite pressure drop", *Energy*, (**32**), pp. 2110-20 (2007).
7. Bes, T. and Roetzel, W. "Distribution of heat flux density in spiral heat exchangers", *International Journal of Heat and Mass Transfer*, **35**(6), pp. 1331-1347 (1992).
8. Bes, T. and Roetzel, W. "Thermal theory for spiral heat exchanger", *International Journal of Heat and Mass Transfer*, **36**(3), pp. 765-773 (1993).
9. Egner, M.W. and Burmeister, L.C. "Heat transfer for laminar flow in spiral ducts of rectangular cross section", *Journal of Heat Transfer*, **127**, pp. 352-356 (2005).
10. Burmeister, L.C. "Effectiveness of a spiral plate heat exchanger with equal capacitance rates", *Journal of Heat Transfer*, **128**, pp. 295-301, (2006).
11. Minton, P.E. "Designing spiral heat exchangers", *Chemical Engineering*, May(**4**), pp. 103-112 (1970).
12. Martin, H., *Heat Exchangers*, Hemisphere Publication Corporation, pp. 73-82 (1992).
13. Gosselin, L., Tye-Gingras, M. and Mathieu-Potvin, F. "Review of utilization of genetic algorithms in heat transfer problems", *International Journal of Heat and Mass Transfer*, (**52**), pp. 2169-2188 (2009).
14. Moretta, A.A. "Spiral plate heat exchangers: Sizing units for cooling non-Newtonian slurries", *Chemical Engineering* (May, 2010).
15. Dongwu, W. "Geometric calculations of the spiral heat exchanger", *Chemical Engineering Technology*, (**26**), pp. 592-598 (2003).
16. Rohsenow, W.M., Harnett, J.P. and Cho, Y.I., *Handbook of Heat Transfer*, 3rd Ed., McGraw Hill, New York (1988).
17. Lindeburg, M.R., *Mechanical Engineering Reference Manual for PE Exam*, 10th Edn. (1997).
18. Bailey, K.M. "Understand spiral heat exchangers", *Chem. Eng. Prog.* (May, 1994).
19. Morimoto, E. and Hotta, K. "Study of the geometric structure and heat transfer characteristics of a spiral plate heat exchanger", *Heat Transfer, Japanese Research*, (**17**), pp. 53-71 (1988).
20. Wen Jei Yang and Rundle, D. "Optimized thermal design of plate and spiral-type heat exchangers", In *HTD-Vol 282, "Challenges of High Temperature Heat Transfer Equipment", ASME* (1994).
21. Hall, S.G. "Capital cost targets for heat exchanger networks comprising mixed materials of construction, pressure ratings and exchangers types", *Chem. Eng.*, **14**(3), pp. 319-335 (1990).
22. Liu, F.B. "A modified genetic algorithm for solving the inverse heat transfer problem of estimating plan heat source", *International Journal of Heat and Mass Transfer*, **51**, pp. 3745-3752 (2008).
23. Sen, M. and Yang, K.T. "Applications of artificial neural networks and genetic algorithms in thermal engineering", *The CRC Handbook of Thermal Engineering*, Kreith, F., Ed., CRC Press, Boca Raton, FL, pp. 620-661 (2000).
24. Pacheco-Vega, A., Sen, M., Yang, K.T. and McClain, R.L. "Genetic algorithms-based predictions of fin-tube heat exchanger performance", *Proceedings of 11th International Heat Transfer Conference*, August 23-28, Kyongju, Korea, **6**, pp. 137-142 (1998).

25. Ozkol, I. and Komurgoz, G. "Determination of the optimum geometry of the heat exchanger body via a genetic algorithm", *Numerical Heat Transfer, Part A*, **48**, pp. 283-296 (2005).

26. Xie, G.N. and Wang, Q.W. "Geometrical optimization of plate- fin heat exchanger using genetic algorithms", *Proceedings of the Chinese Society for Electical Engineering*, **26**(7), pp. 53-57 (2006) (in Chinese).

27. Mishra, M., Das, P.K. and Sarangi, S. "Optimum design of cross flow plate-fin heat exchangers through genetic algorithm", *International Journal of Heat Exchangers*, **5**(2), pp. 379-401 (2004).

28. Selbas, R., Kizilkan, O. and Reppich, M. "A new design approach for shell-and-tube heat exchangers using genetic algorithms from economic point of view", *Chemical Engineering and Processing*, **45**(4), pp. 268-275 (2006).

29. Liang, H.X., Xie, G.N., Zeng, M., Wang, Q.W. and Feng, Z.P. "Application genetic algorithm to optimization recuperator in micro-turbine", *The 2nd International Symposium on Thermal Science and Engineering*, October 23-25, Beijing, China (2005).

30. Wang, Q.W., Liang, H.X., Xie, G.N., Zeng, M., Luo, L.Q. and Feng, Z.P. "Genetic algorithm optimization for primary surfaces recuperator of microturbine", *ASME Journal of Engineering for Gas Turbines and Power*, **129**, pp. 436-442 (2007).

31. Pacheco-Vega, A., Sen, M., Yang, K.T. and McClain, R.L. "Correlations of fin-tube heat exchanger performance data using genetic algorithms simulated annealing and interval methods", *Proceedings of ASME the Heat Transfer Division*, **369-5**, pp. 143-151 (November 11-16, 2001).

32. Pacheco-Vega, A., Sen, M. and Yang, K.T. "Simultaneous determination of in-and-over-tube heat transfer correlations in heat exchangers by global regression", *International Journal of Heat and Mass Transfer*, **46**(6), pp. 1029-1040 (2003).

Biographies

Mehdi Bidabadi received his BS degree in Mechanical Engineering from the Iran University of Science and Technology, and MS and PhD degrees from the University of Sharif, Iran and McGill University, Canada, respectively. He is a Faculty Member at Iran University of Science and Technology. His PhD research involved an experimental and analytical study of laminar dust flame propagation. His core research interests are dust flame propagation mechanisms, development of a new experimental apparatus to produce laminar, and optimization of the smoke wind tunnel. These interests have led to publishing several research papers in the dust combustion. His recent study includes combustion, wind tunnel and gas dynamics.

Abdolali Khalili Sadaghiani received his BS degree in Mechanical Engineering from Iran University of Science and Technology in July, 2012. His preference is applied mechanics related research. He is interested in developing and utilizing novel experimental, analytical and numerical techniques to model and simulate practical problems in Fluid mechanics. His research focuses on the microfluidics, dust combustion, CFD, heat and mass transfer.

Abazar Vahdat Azad received his BS degree in Mechanical Engineering from Islamic Azad University, South Branch, Tehran, in September 2006, and MS degree in Mechanical Engineering from Khaje Nasir Tusi University in September 2009. Currently, he is a PhD candidate in Mechanical Engineering in Iran University of Science and Technology. His Ph.D. research involved the analytical study of laminar dust flame propagation. His research focuses on the dust combustion, HCCI Engine, micro-combustors, heat transfer and energy optimization. These interests have led to publishing several research papers in the mentioned subjects.