International Journal of Heat and Mass Transfer 77 (2014) 1144–1154

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Optimization of shell-and-tube heat exchangers using a general design approach motivated by constructal theory

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ARTICLE INFO

Article history:
Received 18 November 2013
Received in revised form 9 June 2014
Accepted 12 June 2014
Available online 9 July 2014

Keywords:
Shell-and-tube heat exchangers
Constructal optimization design
Series-and-parallel arrangement
Genetic algorithm
Mixed discrete nonlinear programming problem

ABSTRACT

A general optimization design method motivated by constructal theory is proposed for heat exchanger design in the present paper. The simplified version of this design approach is suggested and the optimization problem formulations are given. In this method, a global heat exchanger is divided into several sub heat exchangers in series-and-parallel arrangement. The shell-and-tube heat exchanger is utilized for the method application, and the Tubular Exchanger Manufacturers Association (TEMA) standards are rigorously followed for all design parameters, e.g. tube diameter, arrangement, thickness and number. The fitness function is the total cost of the shell-and-tube heat exchangers, including the investment cost for initial manufacture and the operational cost involving the power consumption to overcome the frictional pressure loss. A genetic algorithm is applied to minimize the objective function by adjusting parameters. Three case studies are considered to demonstrate that the new design approach can significantly reduce the total cost compared to the methods of original design, traditional genetic algorithm design, and old constructal design.

1. Introduction

Shell-and-tube heat exchangers have been broadly utilized in various industrial fields such as petrochemical engineering, manufacture business, and other energy generation, preservation and conservation systems, due to their structural simplicity, design flexibility and low cost. According to Master and co-workers, they account for more than 35–40% of the heat exchangers used in global heat transfer processes [1]. Therefore, it is of great importance to improve the thermal–hydraulic performance, lower the economic cost, and reduce the irreversible dissipation as much as possible. The goal of designing procedure is to determine the most proper values for relevant variables, i.e. operating variables (e.g. fluid allocation, temperature, and mass flow rate) and geometric parameters (e.g. shell length, diameter, and tube arrangement). The conventional design approach is an iterative procedure based on past knowledge and the constraints of working conditions, such as allowable fouling and pressure drops. The final design results are chosen after a significant amount of trial-and-error design until the heat transfer capacity, pressure drops, and working longevity are within the allowable values. This traditional design approach is not cost-effective due to the lack of evaluation criteria.

In the pursuit of improved designs, considerable efforts [2–42] for various optimization methodologies have been devoted to optimizing heat transfer processes. Apart from little work on graphical tool [2,3], most of the research can be divided into two main categories: evolutionary algorithm optimization method [4–36] and mathematical programming optimization method [38–42]. These methods have been well developed and verified from different perspectives such as application and implementation of distinctive evolutionary algorithms [4–11], single-geometric optimization [12,13], economic cost optimization [14–20], the second law analysis of thermodynamics [21–25], practical application design [26–28], and multi-objective optimization [29–35]. According to Rao [43], the optimization design with respect to the practical engineer industry can be classified into three main problems, the first of which is the continuous non-linear programming (CNLP) problem in which all variables have no restrictions except for value range, the second of which is the discrete nonlinear programming (DNLP) problem in which all variables are restricted to adopt only discrete values, the last of which is the mixed-discrete nonlinear programming problem (MDNLP) in which some variables are limited to take discrete values only. It is understandable that among them

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.06.046
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Nomenclature

A	hydraulic area (m^2)
a_1	numerical constant (–)
a_2	numerical constant (–)
a_3	numerical constant (–)
A	heat transfer area (m^2)
A_s	cross flow area (m^2)
B	baffle spacing (m)
c_p	specific heat (kJ kg^{-1} K^{-1})
C_i	capital cost (€)
C_E	energy cost (€ KW^{-1} h^{-1})
Cl	clearance between adjacent tubes (m)
C_o	annual operating cost (€)
C_{od}	total discounted operating cost (€)
C_{tot}	total cost (€)
d_i	inner heat transfer tube diameter (m)
d_0	outer heat transfer tube diameter (m)
D_e	shell-side hydraulic diameter (m)
f	friction factor for tube-side (–)
h	heat transfer coefficient (W m^{-2} K^{-1})
H	annual operating time (h/year)
K	overall heat transfer coefficient (W m^{-2} K^{-1})
K_1	numerical constant (–)
L	heat transfer tube length (m)
$\dot{m}$	mass flow rate (kg s^{-1})
n	tube quantity (–)
n_1	numerical constant (–)
n_y	equipment life (year)
N_p	parallel heat exchanger number (–)
N_s	series heat exchanger number (–)
P	pumping power (W)
Pr	Prandtl number (–)
Pt	tube pitch (m)
Q	heat duty (W)
Re	Reynolds number (–)
R	fouling coefficient (m^2 K W^{-1})
T	temperature (K)
v_t	tube-side fluid velocity (m s^{-1})

Greek symbols

ΔT_{lm}	log mean temperature difference (K)
ΔP	pressure drop (Pa)
λ	thermal conductivity (W m^{-1} K^{-1})
μ	dynamic viscosity (Pa s)
ρ	density (kg m^{-3})
π	numerical constant (–)
η	pump efficiency (–)

Subscripts

c	cold-side
h	hot-side
in	inlet
out	outlet
s	shell-side
t	tube-side
w	tube wall

MDNLP is the most commonly-encountered case since in real circumstance some variables can only have discrete values such as pipe diameter increments (1/8 in), plate or tube bundle quantity (integer value), and machine or laborer number (integer value).

The publications in the open literature demonstrate that for a shell-and-tube heat exchanger either evolutionary algorithm optimization method or mathematical programming optimization method is qualified to resolve design problems. Selbas et al. [14] utilized a genetic algorithm (GA) for the shell-and-tube heat exchanger design and their results demonstrated that the approach has advantages in finding the global minimum heat transfer area (economic cost), obtaining multiple solutions of the same quality and providing more flexibility over past design methods. Later, the particle swarm optimization (PSO) [5], differential evolution (DE) [6], global sensitivity analysis and harmony search algorithm (GSA&HSA) [7], artificial bee colony algorithm (ABC) [16], imperialist competitive algorithm (ICA) [17], biogeography-based optimization algorithm (BBO) [18], quantum particle swarm optimization approach combining with Zaslavskii chaotic map sequences (QPSOZ) [8] were used for designing shell-and-tube heat exchangers. The conventional optimization design method using evolutionary algorithm has been proved to demonstrate the following advantages: fast convergence with good quality and accurate precision; easy implementation for different problems; providing a good chance on finding the global optimal; giving the designer more freedom in the final choice. However, a comparison of all evolutionary algorithm methods demonstrated that although the designs may vary from algorithm to algorithm, the differences between results obtained by using different algorithms are generally not significant for most case studies [5–8,14–18]. Besides, the overwhelming majority of the previous efforts on design optimization adopted continuous values to determine the mechanical parameters such as tube diameter, thickness and length. According to Smith [44], this type of approach provides just a preliminary specification for heat transfer equipment and the preliminary values must be corrected eventually to meet the industrial standard requirements, such as the standards of Tubular Exchanger Manufacturers Association (TEMA) [45] for the case of shell-and-tube heat exchangers. In other words, for the shell-and-tube heat exchanger optimization design process, a CNLP problem is obliged to convert into a MDNLP problem regarding to the real situation.

So far, replete of algorithms and programming formulations have been developed to optimize shell-and-tube heat exchangers. For the same problem, the optimized values obtained from different algorithms or programming methods may differ, but the differences between those results are generally insignificant, as demonstrated before. This statement is in full agreement with the publications in open literature [5–8,14–18]. Both the evolutionary and mathematical programming methods are capable of finding the optimal value or approaching extremely close to the optimal value. It is anticipated that the results and conclusions will remain the same or only change slightly even though the most advanced or superior algorithm or programming method is used. Therefore, in order to reduce economic cost or energy consumption, it is of great significance to explore new design approaches rather than new algorithm or programming techniques. In this paper, a general optimization design approach for heat exchangers motivated by constructal theory is proposed. The problem formulation is illustrated and rigorously followed by the TEMA standards. A genetic algorithm is applied to find the optimal values. Three cases are studied and they demonstrate that the novel method have the advantages in design optimization and broad applications on heat exchanger design in comparison to the existing design methods in the open literature.

2. A constructal theory based method and its application on heat exchangers design

2.1. Constructal theory and optimization design of heat exchanger

Bejan proposed constructal theory with the statement as follows: for a finite-size system to persist in time (to live), it must evolve in such a way that it provides easier access to the imposed currents that flow through it [46]. It has been used to interpret some natural phenomena, such as the geometric features of rivers, clouds, veins and arteries, and trees [47–49]. Also it can be adopted as an optimization design method for engineering applications, such as fins of cooling devices, fuel cells, and plate heat exchangers [50–52]. Replete of research has been conducted in open literature [53–62]. Bejan [50] applied constructal theory on conducting paths for cooling a heat generating volume. The results indicate that the sequence of optimization design has a definite time direction, which begins with the small element system and proceeds to large assemblies. The application demonstrates that when the definite time direction is reversed, i.e. from large elements to small elements, the tree-like network cannot be achieved.

For a bio or non-bio flow system, constructal theory indicates that it will morph towards the direction that facilitates flow. Take a shell-and-tube heat exchanger (an engineering system) as an example, constructal theory implies that a design (configuration, flow pattern, geometry) with higher stability, durability and conservation can be achieved by increasing the access level of elements (working fluid, energy) that flow through it. In order to imitate a "tree-like network" configuration as shown in Fig. 1, Azad [19] proposed a novel heat exchanger, so-called *constructal shell-and-tube heat exchanger*, to maximize the access of the cold stream to the heat flux of the hot stream and minimize the thermal resistance. For the practical application, a constructal shell-and-tube heat exchanger is defined as a heat exchanger with two or more bundles in-series section, the tube quantity of the latter heat exchanger being twice that of the former heat exchanger, as shown in Fig. 2. Azad and co-workers used this methodology in optimization design using a genetic algorithm, and the results demonstrated a dramatic total cost reduction compared to the original design. However, this design method arrays all the heat exchangers in a series arrangement and requires that the tube number in the second bundle must be twice that of the first bundle. The two requirements need to be improved and modified due to the following reasons. First, this method yields an impractical design of shell-and-tube heat exchanger with shell length of 0.161 m, shell diameter of 1.328 m and tube number of 3988. Second, it does not provide a better solution than the conventional genetic algorithm design method from the economic point of view (a conventional genetic algorithm design method refers to that uses the genetic algorithm to design a single heat exchanger).

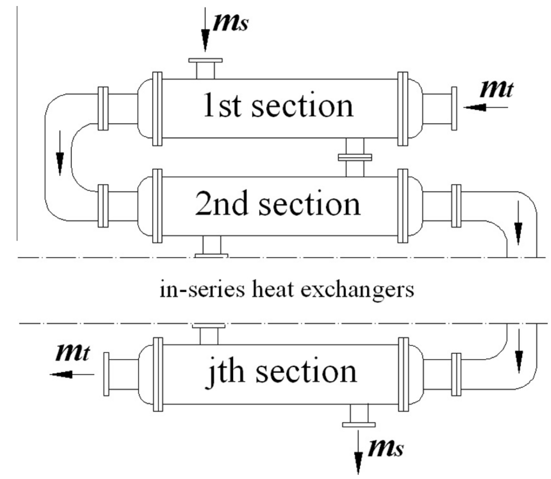

Fig. 2. The in-series shell-and-tube heat exchangers.

Motivated by the above applications of constructal theory, a heat exchanger optimization design method is proposed. In order to improve the design of heat exchanger, we utilized the concept of a constructal shell-and-tube heat exchanger, that the heat exchanger is split into a series of two shell-and-tube heat exchangers. In this method, a shell-and-tube heat exchanger is divided into $N_s \times N_p$ heat exchangers. A certain number of heat exchangers (quantity: N_s) are displayed in series arrangement, and a certain number of heat exchanger (quantity: N_p) are placed in parallel arrangement as shown in Fig. 3. The energy is transferred from hot stream to cold stream through the heat exchanger group. For the convenience of analysis, the heat exchanger group, which is restricted by the dotted lines in Figs. 3 and 4, is called *global heat exchanger* or global-HE. A global heat exchanger functions as a real heat exchanger, and each small heat exchanger is called *sub heat exchanger* or sub-HE. Unlike the path design optimization for a cooling volume based on constructal theory, the optimization design in this method does not have a certain time direction (optimize each sub-HE from small element to large element). By contrast, all sub-heat exchangers are optimized simultaneously. Through this method, the feasible solution domain is expected to increase since there are more combinations for $N_s \times N_p$ sub-heat exchangers rather than one heat exchanger. It should be noticed that large sub-heat exchanger number accompanies by large local pressure drop, which increases the pumping cost. The total cost (investment cost and operational cost) minimization of the global-HE which includes $N_s \times N_p$ sub-HEs was taken as the objective function. A genetic algorithm was used to optimize the objective function by adjusting mechanical and flow parameters of each sub-HE. During the design procedure, the TEMA standards are

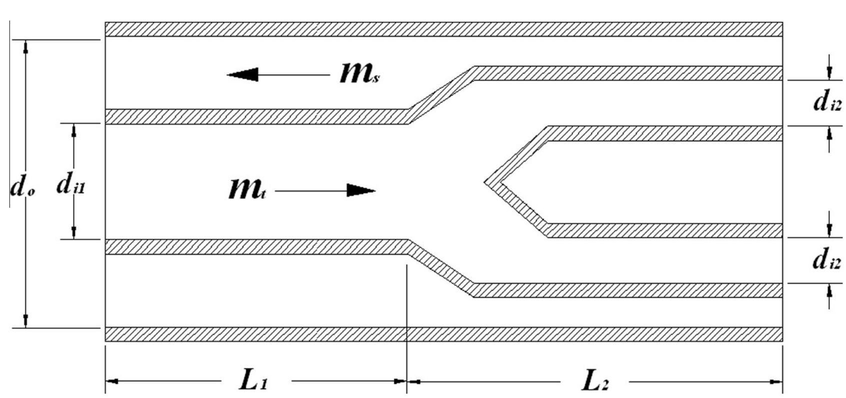

Fig. 1. Tree-shaped structure for a heat exchanger based on constructal theory.

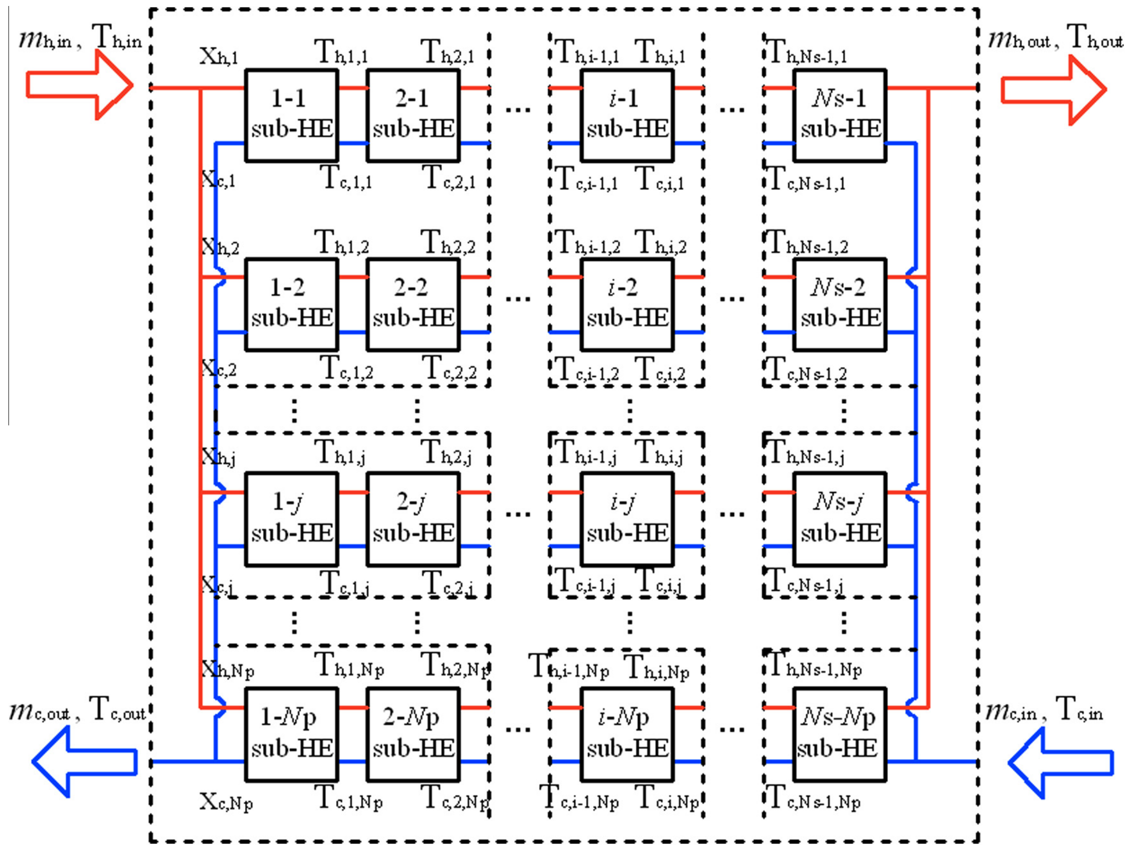

Fig. 3. The series-and-parallel arranged heat exchangers.

rigorously followed for each-HE. In order to demonstrate the novelty and improvement of this novel design method, the original design approach, the traditional genetic algorithm approach, and the simple constructal design approach proposed by our group were taken as the reference groups.

2.2. Problem formulation

2.2.1. Series-and-parallel arranged heat exchangers

The sub-HEs are displayed in the series-and-parallel arrangement, as demonstrated in Fig. 3. The optimization model presentation is illustrated as follows. The goal is to minimize or maximize the fitness function, which is expressed as:

$$\text{Minimize/maximize}\sum_{i=1}^{N_s}\sum_{j=1}^{N_p} f_{ij}(x) \quad x=[x_{ij,1},x_{ij,2},\ldots,x_{ij,k},\ldots,x_{ij,M}] \tag{1}$$

The constraints of variables with continuous values are stated as follows:

$$x_{ij,k}|_{\min} \leqslant x_{ij,k} \leqslant x_{ij,k}|_{\max} \tag{2}$$

The constraints of variables with discrete values are stated as follows:

$$\begin{cases} x_{ij,1}\in[C_{1,1},C_{1,2},\ldots,C_{1,L_1}] \\ x_{ij,2}\in[C_{2,1},C_{2,2},\ldots,C_{2,L_2}] \\ \ldots \\ x_{ij,k}\in[C_{k,1},C_{k,2},\ldots,C_{k,L_k}] \\ \ldots \\ x_{ij,M}\in[C_{M,1},C_{M,2},\ldots,C_{M,L_M}] \end{cases} \tag{3}$$

Other constraints for each sub-HE and the global-HE are stated as follows:

$$\begin{cases} g_{ij}(x)\big|_{\min} \leqslant g_{ij}(x) \leqslant g_{ij}(x)\big|_{\max} \\ \left.\sum_{i=1}^{N_s}\sum_{j=1}^{N_p} g_{ij}(x)\right|_{\min} \leqslant \sum_{i=1}^{N_s}\sum_{j=1}^{N_p} g_{ij}(x) \leqslant \left.\sum_{i=1}^{N_s}\sum_{j=1}^{N_p} g_{ij}(x)\right|_{\max} \\ x=[x_{ij,1},x_{ij,2},\ldots,x_{ij,k},\ldots,x_{ij,M}] \end{cases} \tag{4}$$

The fitness function is expressed as Eq. (1) and set as the total cost of all sub-HEs in this paper; $x_{ij,k}$ is the kth design variable (tube diameter, fin frequency or plate number) for the ith × jth sub-HE. For each sub-HE, the lower and upper bounds are expressed in Eq. (2) for the continuous values and in Eq. (3) for the discrete values. The constraints of each sub-HE such as maximum pressure drop, minimum heat transfer capacity and energy conservation, and the constraints of global-HE such as volume restriction are expressed in Eq. (4). N_s and N_p are the quantities of sub-heat exchangers in series and parallel respectively; M is the quantity of input variables for every sub-section. From the above formulation, it should be noticed that the series-and-parallel arranged HEs design is time-consuming due to complex arrangement pattern, large number of sub-HEs and variables. Therefore, it should be simplified into two simple forms.

2.2.2. Series-arranged heat exchangers or parallel-arranged heat exchangers

The previous optimization design method and problem formulation can provide highly general applicability; however, it is extremely time-consuming. From the perspective of resource-saving and fast convergence, the optimization design method and problem formulation can be simplified into two special cases: the series-arranged heat exchangers (when $N_p = 1$), and the

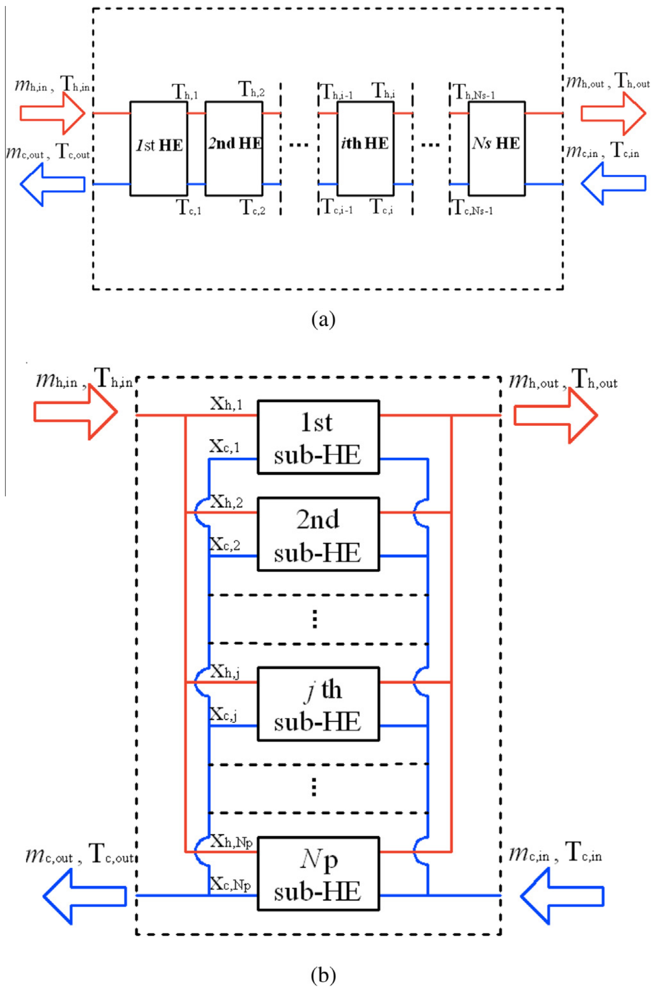

Fig. 4. The simplified heat exchangers design: (a) series-arranged heat exchanger; (b) parallel-arranged heat exchanger.

parallel-arranged heat exchangers (when $N_s = 1$) as shown in Fig. 4. For a series-arranged or parallel-arranged heat exchangers optimization problem, the fitness function is stated as follows:

$$\text{Minimize/maximize} \sum_{i=1}^{N_s} f_i(x) x = \lfloor x_{i1}, x_{i2}, \ldots, x_{ik}, \ldots, x_{iM} \rfloor \tag{5}$$

The constraints of variables with continuous values are stated as follows:

$$x_{ik}|_{\min} \leqslant x_{ik} \leqslant x_{ik}|_{\max} \tag{6}$$

The constraints of variables with discrete values are stated as follows:

$$\begin{cases} x_{i1} \in [C_{1,1}, C_{1,2}, \ldots, C_{1,L_1}] \\ x_{i2} \in [C_{2,1}, C_{2,2}, \ldots, C_{2,L_2}] \\ \ldots \\ x_{ik} \in [C_{k,1}, C_{k,2}, \ldots, C_{k,L_k}] \\ \ldots \\ x_{iM} \in [C_{M,1}, C_{M,2}, \ldots, C_{M,L_M}] \end{cases} \tag{7}$$

Other constraints for each sub-HE and the global-HE are stated as follows:

$$\begin{cases} g_i(x)|_{\min} \leqslant g_i(x) \leqslant g_i(x)|_{\max} \\ \left.\sum_{i=1}^{N} g_i(x)\right|_{\min} \leqslant \sum_{i=1}^{N} g_i(x) \leqslant \left.\sum_{i=1}^{N} g_i(x)\right|_{\max} \\ x = \lfloor x_{i1}, x_{i2}, \ldots, x_{ik}, \ldots, x_{iM} \rfloor \end{cases} \tag{8}$$

The fitness function is expressed as Eq. (5) and set as the total cost of all sub-HEs in this paper. x_{ik} is the kth design variable (tube diameter, fin frequency or plate number) for the ith heat exchanger. For each sub-HE, the lower and upper bounds are expressed in Eq. (6) for the continuous values and in Eq. (7) for the discrete values. The constraints of each sub-HE such as maximum pressure drop, minimum heat transfer capacity and energy conservation, and the constraints of global-HE such as volume restriction are expressed in Eq. (8). N is the quantity of sub-heat exchangers, and M is the quantity of input variables for every sub-section. It should be noticed that although all sub-HEs in Figs. 3 and 4 are counter-flow HEs, the proposed optimization design method is also applicable for heat exchangers with other flow pattern.

3. Application

3.1. Optimization approach

Holland [63] first proposed the principles of GA conceived from the mechanism of natural selection in a competitive environment. The GA, as one of the family of evolutionary algorithms (EA), is routinely applied for optimization and search problems in engineering design. The GA starts with an initial population of design candidates that represents "parents" to generate "offspring" with shared attributes from their parents. Then the most fit of the offspring parent another generation. As this process is repeated, complex combinations in the design space arise, and the best designs are retained. The process continues until the appearance of an individual with a predefined target fitness, or until a limiting generation is reached. For the optimization design procedure, the operational and geometric variables, objective function and each design solution are analogous to chromosomes, fitness values and individual. The flow chart of optimization design for a heat exchanger based on GA is presented in Fig. 5. For the readers' convenience, more information of GA utilization in heat transfer problems could be found in [36]. The optimization was performed on a computer with Intel Xeon CPU E5630 of 2.53 GHz and 14.00 GB of RAM using the genetic algorithm optimization toolbox *ga* solver in MATLAB. For

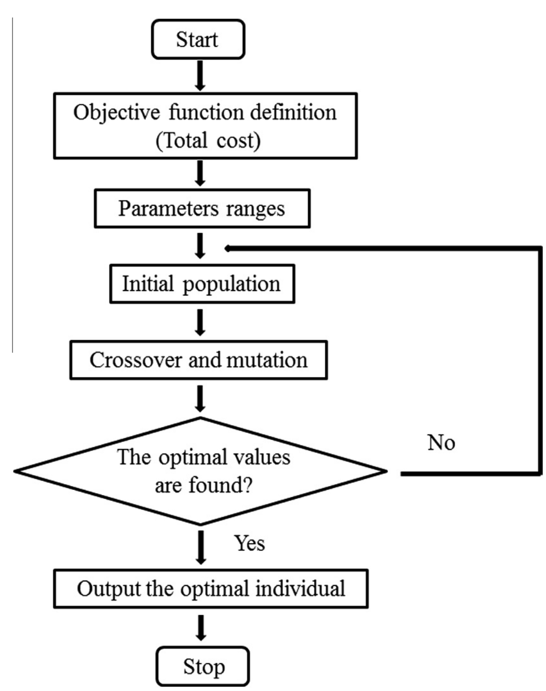

Fig. 5. Flow chart of the genetic algorithm using the total cost as objective function.

the genetic algorithm parameters setting, the initial population, maximum generation and mutation probability were set to 50, 500 and 0.3265, respectively. It should be noticed that other evolutionary algorithms (GA, PSO, ABC, BBO, QPSOZ and so on) could be easily utilized in the novel design approach.

3.2. Heat transfer, pressure drop and economic calculations

The energy balance equation for a shell-and-tube heat exchanger is shown in Eq. (9):

$$Q = (\dot{m}c_p)_t(T_{t,i} - T_{t,o}) = (\dot{m}c_p)_s(T_{s,o} - T_{s,i}) \tag{9}$$

where Q is the heat duty, $\dot{m}$ is the mass flow rate, c_p is the specific heat capacity, T is the temperature, the subscripts t and s stand for the tube-side and shell-side, respectively; and the subscripts i and o stand for the inlet and outlet of tube or shell side, respectively. The shell inner diameter is calculated as:

$$D_s = \left(\frac{n}{K_1}\right)^{1/n_1} \cdot d_o \tag{10}$$

where n is the tube number, d_o is the tube outer diameter, K_1 and n_1 are coefficients taken according to the arrangement and passes number of the tubes, which can be found in [54]. The tube pitch and inner diameter are calculated as follows [64]:

$$P_t = 1.25d_o \tag{11}$$

$$d_i = d_o - 2t \tag{12}$$

where t is the tube-wall thickness. The heat transfer area is calculated as:

$$A = \frac{Q}{K\Delta T_{lm}F} \tag{13}$$

where A is the heat transfer area based on the outer diameter of the tube, ΔT_{lm} is the logarithmic mean temperature difference (LMTD), F is the correction factor for LMTD according to the equipment architecture [65], and K is the overall heat transfer coefficient based on the outer diameter of the tube. The LMTD and F are computed through Eqs. (14) and (15), respectively [66]:

$$\Delta T_{lm} = \frac{(T_{s,i} - T_{t,o}) - (T_{s,o} - T_{t,i})}{\ln((T_{s,i} - T_{t,o})/(T_{s,o} - T_{t,i}))} \tag{14}$$

$$F = \begin{cases} 1 & \text{tube pass} = 1 \\ \frac{\sqrt{R^2+1}}{R-1} \cdot \frac{\ln\left(\frac{1-P}{1-PR}\right)}{\ln\left[\frac{2-P(R+1-\sqrt{R^2+1})}{2-P(R+1+\sqrt{R^2+1})}\right]} & \text{tube pass} = \text{even number} \end{cases} \tag{15}$$

where

$$R = \frac{T_{s,i} - T_{s,o}}{T_{t,o} - T_{t,i}} \tag{16}$$

and

$$P = \frac{T_{t,o} - T_{t,i}}{T_{s,i} - T_{t,i}} \tag{17}$$

The overall heat transfer coefficient is calculated using [65]:

$$K = \left[\frac{1}{h_t}\left(\frac{d_o}{d_i}\right) + R_t\left(\frac{d_o}{d_i}\right) + \frac{d_o \ln(d_o/d_i)}{2\lambda_w} + R_s + \frac{1}{h_o}\right]^{-1} \tag{18}$$

where h_t and h_s are the heat transfer coefficients, while R_t and R_s are fouling coefficients. λ_w is the thermal conductivity for tube wall at the bulk mean temperature of fluid. The tube number is calculated as:

$$n = \frac{A}{\pi d_o L} \tag{19}$$

where L is the tube length. The tube side heat transfer coefficient, based on an assumption of turbulent, fully developed flow, is calculated using Eq. (20) [65]:

$$h_t = 0.023\frac{\lambda_t}{d_i}Re_t^{0.8}Pr_t^{\frac{1}{3}}\left(\frac{\mu_t}{\mu_t w}\right)^{0.14} \tag{20}$$

where μ_t is the fluid dynamic viscosity at the bulk temperature of tube-side $T_{b,t}$, and μ_{tw} is fluid dynamic viscosity at the inner tube wall temperature T_{tw}, which can be estimated through Eq. (21) [64]:

$$h_t(T_{tw} - T_{b,t}) = K(T_{b,s} - T_{b,t}) \tag{21}$$

where $T_{b,s}$ is the fluid bulk temperature of shell-side flow. The Reynolds number for the tube-side flow is calculated using:

$$Re_t = \frac{\rho_t v_t d_i}{\mu_t} \tag{22}$$

The tube-side fluid velocity is calculated as:

$$v_t = \frac{N_{pass}}{n} = \frac{\dot{m}_t}{\pi(d_i^2/4)\rho_t} \tag{23}$$

where N_{pass} is the tube pass number. Kern's method is utilized to calculate the shell-side heat transfer coefficient expressed in Eq. (24) [66]:

$$h_s = 0.36\frac{\lambda_s}{D_e}Re_s^{0.55}Pr_s^{\frac{1}{3}}\left(\frac{\mu_s}{\mu_{sw}}\right)^{0.14} \tag{24}$$

where D_e is the shell-side hydraulic diameter, μ_s is the dynamic viscosity coefficient at the bulk temperature of shell-side fluid, while μ_{sw} is the dynamic viscosity coefficient at outer tube wall temperature, which can be estimated through Eq. (25) [64]:

$$h_s(T_{b,s} - T_{tw}) = K(T_{b,s} - T_{b,t}) \tag{25}$$

The shell-side hydraulic diameter and Reynolds number are calculated by Eqs. (26) and (27), respectively [64,66]:

$$F = \begin{cases} \frac{4(P_t^2 - \pi d_o^2/4)}{\pi d_o} & \text{for square arrangement} \\ \frac{4(P_t^2/2 \times 0.87P_t - (0.5\pi d_o^2/4))}{\pi d_o} & \text{for triangle arrangement} \end{cases} \tag{26}$$

$$Re_s = \frac{\dot{m}_s \cdot D_e}{\mu_s \cdot A_s} \tag{27}$$

where B is the baffle spacing, A_s is the cross area of fluid flow which is calculated as below [64,66]:

$$A_s = \frac{D_s \cdot B(P_t - d_o)}{P_t} \tag{28}$$

The tube-side pressure drop can be obtained through Eq. (29) [64]:

$$\Delta P_t = N_{pass}\left(4f_t\frac{L}{d_i} + 2.5\right)\frac{\rho_t v_t^2}{2} \tag{29}$$

Here, f_t is the friction factor for turbulent tube flow that is expressed in Eq. (30):

$$F_t = 0.046(Re_t)^{-0.2} \tag{30}$$

The shell-side pressure drop is calculated by Eq. (31) [67]:

$$\Delta P_s = f_s \cdot \left(\frac{\rho_s v_s^2}{2}\right) \cdot \left(\frac{L}{B}\right) \cdot \left(\frac{D_s}{D_e}\right) \tag{31}$$

where f_s is the friction factor for shell-side which is expressed in Eq. (32):

$$f_s = 2b_o Re_s^{-0.15} \tag{32}$$

where $b_o = 0.72$ [67] is valid for $Re_s < 40{,}000$. The total power consumption is calculated through Eq. (33) [68]:

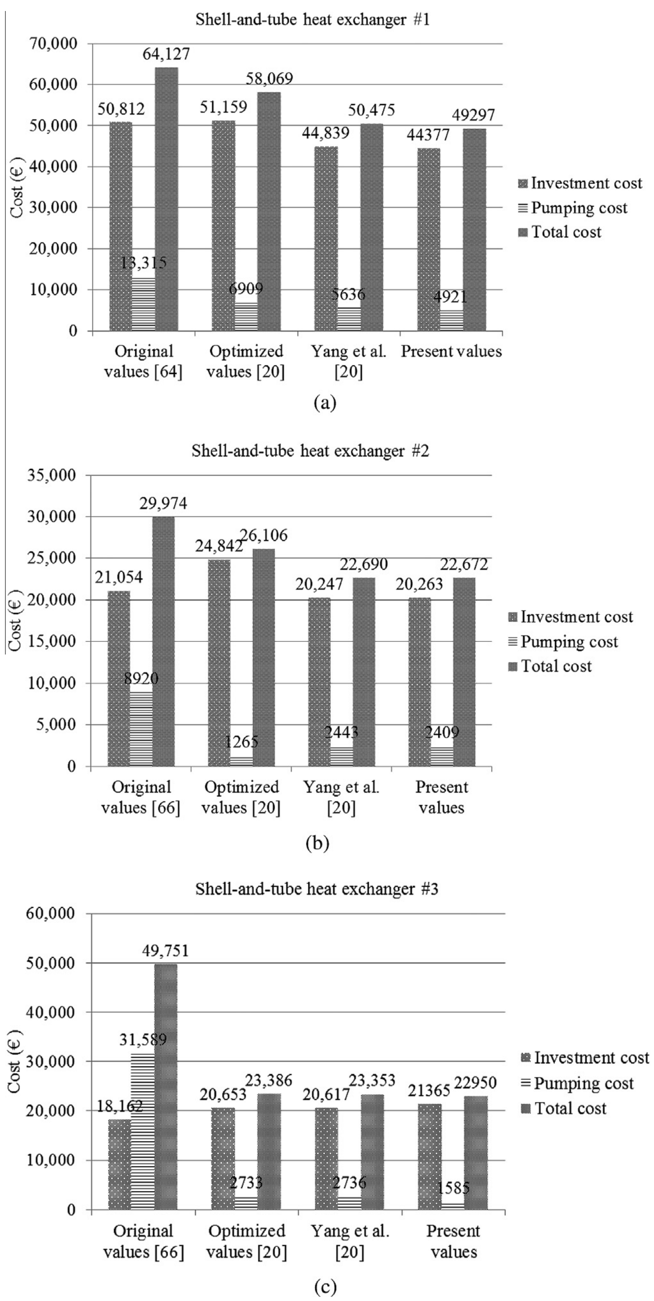

Fig. 6. Comparisons of investment cost, pumping cost and total cost for different design values: (a) the first case study; (b) the second case study; (c) the third case study.

$$P = \frac{1}{\eta}\left(\frac{\dot{m}_s}{\rho_s}\Delta P_s + \frac{\dot{m}_t}{\rho_t}\Delta P_t\right) \tag{33}$$

Here η is the pump efficiency and we give it a constant of 0.7. The total cost is obtained through Eq. (34) [65]:

$$C_{tot} = C_i + C_{od} \tag{34}$$

where C_i and C_{od} are the capital investment cost and the total discounted operating cost [69] which can be calculated through Eqs. (35) and (36), respectively.

$$C_i = a_1 + a_2 A^{a3} \tag{35}$$

$$C_{od} = \sum_{k=1}^{ny} \frac{C_o}{(1+i)^k} \tag{36}$$

Here, $a_1 = 8000$, $a_2 = 259.2$ and $a_3 = 0.91$ for shell-and-tube heat exchangers made of stainless steel [69]. i is the fractional interest rate per year which is set as 10% and ny is set as 10 years. C_o is the annual operating cost that can be calculated through Eq. (37):

$$C_o = P \cdot C_E \cdot H \tag{37}$$

where C_E is the energy cost which is set as 0.12 €/KW h, and H is the amount of working hours which is set as 7000 h per year.

3.3. Design parameters, constraint conditions and objective function

The objective function for the case studies is the minimization of total cost including initial investment cost and power consumption cost. The allowable maximum pressure drops for shell-side and tube-side are both 70,000 Pa. All sub-HEs meet energy conservation, i.e. the energy between hot stream and cold stream is in balance for each sub-HE. The input parameters (discrete values) are illustrated as follows according to the TEMA design standards:

1. The tube layout adopts two arrangements (ARR): triangular arrangement (30°) or square arrangement (90°).
2. The tube passes number (N_{pass}) adopts four discrete values: 1, 2, 4, or 8.
3. The baffle spacing (B) varies from the minimum baffle spacing of 0.0508 m to the maximum unsupported tube span of $29.5 \times d_o^{0.75}$ where d_o is in meters.
4. The tube length (L) adopts ten discrete values: 2.438 m, 3.048 m, 3.658 m, 4.877 m, 6.096 m, 7.32 m, 8.53 m, 9.75 m, 10.7 m or 11.58 m.
5. The tube outer diameter (d_o) adopts seven values: 0.01588 m, 0.01905 m, 0.02223 m, 0.0254 m, 0.03175 m, 0.0381 m or 0.0508 m.
6. The tube wall thickness (t) adopts discrete values based on the Birmingham Wire Gauge (BWG) according to the recommendations of TEMA [20]. The optimization problem is written in another form as follows.

Table 1
Case studies specifications [64,66].

Fluid allocation	Case study #1		Case study #2		Case study #3	
	Shell	Tube	Shell	Tube	Shell	Tube
	Methanol	Sea water	Kerosene	Crude oil	Distilled water	Raw water
Mass flow (kg/s)	27.80	68.90	5.52	18.80	22.07	35.31
T input (°C)	95.0	25.0	199.0	37.8	33.9	23.9
T output (°C)	40.0	40.0	93.3	76.7	29.4	26.7
ρ (kg/m^3)	750	995	850	995	995	999
C_p (kJ/kg K)	2.84	4.20	2.47	2.05	4.18	4.18
μ (Pa s)	0.00034	0.00080	0.00040	0.000358	0.00080	0.00092
λ (W/m K)	0.19	0.59	0.13	0.13	0.62	0.62
$R_{fouling}$ (m^2 K/W)	0.00033	0.00020	0.00061	0.00061	0.00017	0.00017

7. The sub-HE numbers N_s and N_p are larger than 1 and only take integer values.

The shell-and-tube heat exchanger design optimization problem formulation is given as follows:

$$\text{The objective function}: \sum_{i=1}^{N_s}\sum_{j=1}^{N_p} C_{tot,ij}\left(ARR_{ij}, N_{pass\,ij}, B_{ij}, L_{ij}, d_{oij}, t_{ij}\right) \tag{38}$$

$$\text{Variables constraints}: \begin{cases} ARR_{ij} \in [30^\circ, 90^\circ] \\ N_{pass\,ij} \in [1,2,4,8] \\ 0.0508 \leqslant B_{ij} \leqslant 29.5 \times d_o^{0.75} \\ L_{ij} = \in \begin{bmatrix} 2.438, 3.048, 3.658, 4.877, 6.096, \\ 7.32, 8.53, 9.75, 10.7, 11.58 \end{bmatrix} \\ D_{o,ij} \in \begin{bmatrix} 0.01588, 0.01905, 0.02223, \\ 0.0254, 0.03175, 0.0508 \end{bmatrix} \\ t_{ij} \in BWG \end{cases} \tag{39}$$

$$\text{Other constraints}: \begin{cases} \Delta P_{t,ij}(ARR_{ij}, N_{pass\,ij}, B_{ij}, L_{ij}, d_{oij}, t_{ij}) \leqslant 70{,}000 \\ \Delta P_{s,ij}(ARR_{ij}, N_{pass\,ij}, B_{ij}, L_{ij}, d_{oij}, t_{ij}) \leqslant 70{,}000 \\ m_{c,in} = m_{c,out}, \quad m_{h,in} = m_{h,out} \\ x_{h,1} + x_{h,2} + \cdots x_{h,N_p} = 1 \\ x_{c,1} + x_{c,2} + \cdots x_{c,N_p} = 1 \\ x_{h,j} m_{h,in} \cdot c_{ph} \cdot (T_{h,i-1,j} - T_{h,i,j}) = x_{c,j} m_{c,in} \cdot c_{pc} \\ \cdot (T_{c,i-1,j} - T_{c,i,j}) \\ i = 1,2,\ldots,N_s \;\; j = 1,2,\ldots,N_p \end{cases} \tag{40}$$

4. Results and discussion

Three case studies [64,66] were undertaken to further explore the relative advantages and disadvantages of the design approaches. The characteristics of three shell-and-tube heat exchangers are given in Table 1. Four different design values are compared, which are original design values, conventional GA design values, conventional constructal design values, and novel constructal design values. It should be noted that the conventional constructal design values [19] were obtained by using one of the simplified optimization design methods in this work.

Case study #1: methanol-brackish water heat exchanger. This case study was taken from [64]. The original design is a heat exchanger with two tube-side passes (triangular arrangement) and one shell-side pass. The same architecture was used in the conventional GA approach. Different designs are compared in Fig. 6 and Table 2. Quantitatively, the total cost for the four various results is 64127 €, 58069 €, 50475 €, and 49297 €, respectively. The heat exchanger number N_s is 1 and N_p is 2. The constructal design method proposed previously in the work, solved the design process with a dramatic economic cost reduction up to 23% compared to the original design. For the new design method, the global optimum (the minimum cost) is achieved using two sub-heat exchangers in parallel arrangement, instead of series arrangement which is obtained from the traditional constructal method proposed in [19].

Case study #2: kerosene-crude oil heat exchanger. This case study was taken from [66]. The original design assumed a heat exchanger with four tube-side passes (square arrangement) and one shell-side pass. The four different methods are compared in

Table 2
Case study #1: optimal parameters of using three design approaches.

	Original values [64]	GA values [20]	Yang et al. [20]		Present values	
			1st part	2nd par	1st part	2nd part
Sub-units	–	–	2 Sub-heat exchangers		2 Sub-heat exchangers	
HEs			Series arrangement		Parallel arrangement	
Intermediate			T_{s1} = 343 K		x_{s1} = 0.3948	x_{s2} = 0.6052
Pattern	Triangular	Triangular	Triangular	Triangular	Triangular	Triangular
Tube passes	2	2	1	1	1	1
Shell passes	1	1	1	1	1	1
D_s (m)	0.894	0.8229	0.6519	0.5115	0.3863	0.4719
L (m)	4.748	3.658	3.658	2.438	6.096	6.096
B (m)	0.356	0.5	0.777	0.768	0.466	0.5892
d_o (m)	0.020	0.01588	0.01588	0.01588	0.1588	0.1588
P_t (m)	0.025	0.01985	0.01985	0.01985	0.1985	0.1985
t (m)	0.002	0.001651	0.001651	0.001651	0.001651	0.001651
Cl (m)	0.005	0.00397	0.00397	0.00397	0.00397	0.00397
N_t	918	1514	911	542	297	456
v_t (m/s)	0.7507	0.7366	0.612	1.029	0.7412	0.74
Re_t	14939	11523	9575	16093	11595	11577
Pr_t	5.7	5.7	5.7	5.7	5.7	5.7
h_t (W/m^2 K)	3878	4008	3456	5236	4028	4023
f_t	0.006728	0.007087	0.007354	0.006629	0.007078	0.00708
ΔP_t (Pa)	5880	5800	2060	4022	4432	4420
a_s (m^2)	0.05883	0.08229	0.1013	0.07857	0.036	0.05561
D_e (m)	0.0142	0.01128	0.01128	0.01128	0.01128	0.01128
v_s (m/s)	0.6301	0.4504	0.3659	0.4717	0.4065	0.4034
Re_s	19739	11203	9101	11734	10110	10034
Pr_s	5.082	5.082	5.082	5.082	5.082	5.082
h_s (W/m^2 K)	1903	1755	1565	1800	1663	1656
f_s	0.3266	0.3556	0.3669	0.3531	0.3611	0.3615
ΔP_s (Pa)	37733	14445	5013	4245	10024	9552
U (W/m^2 K)	634.3	628.6	584.8	665.5	617	615.9
A (m^2)	273.7	276.1	166.1	65.9	90.25	138.6
C_i (€)	50,812	51,159	44,839		44,377	
C_o (€)	2167	1124	917.2		800.9	
C_{od} (€)	13,315	6909	5636		4921	
C_{total} (€)	64,127	58,069	50,475		49,297	

Table 3
Case study #2: optimal parameters of using three design approaches.

	Original values [66]	GA values [20]	Yang et al. [20]		Present values	
			1st part	2nd part	1st part	2nd part
Sub-units	–	–	2 Sub-heat exchangers		2 Sub-heat exchangers	
HEs			Series arrangement		Series arrangement	
Intermediate			T_{s1} = 436.4 K		T_{s1} = 436.6 K	
Pattern	Square	Square	Triangular	Triangular	Triangular	Triangular
Tube passes	4	4	1	1	1	1
Shell passes	1	1	1	1	1	1
D_s (m)	0.539	0.765	0.2901	0.2932	0.2924	0.2881
L (m)	5.983	2.438	6.096	3.048	6.096	3.048
B (m)	0.127	0.138	0.304	0.367	0.3321	0.3069
d_o (m)	0.025	0.01905	0.01588	0.01905	0.01588	0.01905
P_t (m)	0.031	0.0238125	0.01985	0.0238125	0.01985	0.0238125
t (m)	0.0025	0.001651	0.001651	0.001651	0.001651	0.001651
Cl (m)	0.006	0.0047625	0.00397	0.0047625	0.00397	0.0047625
N_t	158	673	161	111	164	107
v_t (m/s)	1.523	0.5768	0.9461	0.9120	0.9299	0.9468
Re_t	8468	2525	3308	3901	3251	4051
Pr_t	56.45	56.45	56.45	56.45	56.45	56.45
h_t (W/m^2 K)	1086	524.1	814.4	759.5	803.2	782.6
f_t	0.007537	0.009601	0.009096	0.008801	0.009128	0.008735
ΔP_t (Pa)	53,195	5594	8965	3920	8686	4202
a_s (m^2)	0.01344	0.02111	0.01764	0.02152	0.01942	0.01768
D_e (m)	0.02469	0.01881	0.01128	0.01353	0.01128	0.01353
v_s (m/s)	0.4831	0.3076	0.3682	0.3018	0.3344	0.3672
Re_s	25344	12294	8823	8674	8011	10556
Pr_s	7.6	7.6	7.6	7.6	7.6	7.6
h_s (W/m^2 K)	978.9	862.9	1199	990.6	1137	1104
f_s	0.3146	0.3507	0.3686	0.3695	0.3739	0.3588
ΔP_s (Pa)	25344	10134	10954	2575	8455	4352
U (W/m^2 K)	268.1	202.6	257.2	241.6	252.9	250.7
A (m^2)	74.21	98.18	48.87	20.31	49.71	19.57
C_i (€)	21,054	24,842	20,247		20,263	
C_o (€)	1452	205.8	397.6		392	
C_{od} (€)	8920	1265	2443		2409	
C_{total} (€)	29,974	26,106	22,690		22,672	

Fig. 6 and Table 3. Quantitatively, the total cost for different design methods is 29974 €, 26106 €, 22690 €, and 22672 €, respectively. The heat exchanger number N_s is 2 and N_p is 1. For the novel design approach, the total cost is decreased by 24.4% compared to the original values. The minimum total cost is obtained by the case of two sub-heat exchangers in series arrangement, the same with the results obtained from the old constructal method.

Case study #3: distilled water-raw water heat exchanger. This case study was also taken from [66]. The original design assumed a heat exchanger with two tube-side passes (triangular arrangement) and one shell-side pass. As shown in Fig. 6 and Table 4, the total cost of shell-and-tube heat exchanger for the original design, traditional GA design, simplified constructal design, and the global constructal design is 49751 €, 23386 €, 23353 €, and 22950 €, respectively. The heat exchanger number N_s is 1 and N_p is 2. The global constructal design method proposed in the present work can achieve a dramatic economic cost reduction up to 53.9% compared to the original design method. The minimum economic cost is 22950 € in the case of two sub-HEs in parallel, not in series obtained from [19].

For the three case studies of shell-and-tube heat exchange optimization design in the present work, there are five variables to decide according to the TEMA standards: two tube arrangements, four tube pass numbers, ten tube lengths, seven outer tube diameters and several tube wall thicknesses for different diameters. The five variables consist of 1360 design solutions. However, by the utilization of the constructal design method in this paper, the combination adds up to $1360^{N_s \times N_p}$ ($N_s \geqslant 1$, $N_p \geqslant 1$) solutions as the number of heat exchangers in series arrangement and parallel arrangement is N_s and N_p, respectively. On the other hand, when the simplified constructal design method is applied, the solution number is either 1360^{N_s} or 1360^{N_p}. Although part of solution domain is invalid due to the constraint conditions, i.e. maximum pressure allowance, minimum heat transfer capacity, local energy conservation of each sub-HE and global energy conservation, the feasible domain is still extended compared to the traditional methods. It should be noted that either the old or the new constructal method has a higher possibility on finding the global optimum than the other. In fact, both methods are capable of finding or approaching the global optimum by using the genetic algorithm or other evolutionary algorithms. It is the various solution domains of the two optimization methods that cause the difference between the values of the two design methods. From the analyses of the above three cases, it is concluded the constructal theory-based optimization design method proposed in the work has an obvious advantage in solving the MDNLP problems. It achieves a dramatic cost reduction compared to the past design methods by expanding the feasible solution domain. In addition, it can be converted to the simplified versions according to real design circumstance. More importantly, the present design methodology can be easily implemented in many other engineering fields such as plate-fin heat exchangers, tube-fin heat exchangers, and brazed plate heat exchangers as they contain large discrete variables (e.g. herringbone angle, tube diameter increment, and plate quantity).

5. Conclusion

In this paper, an optimization design method based on constructal theory is proposed for heat exchanger application. The novel design adopts the perception that divides a whole heat exchanger into several sub heat exchangers, arranges sub-HEs in

Table 4
Case study #3: optimal parameters of using three design approaches.

	Original values [66]	GA values [20]	Yang et al. [20]		Present values	
			1st part	2nd part	1st part	2nd part
Sub-units	–	–	2 Sub-heat exchangers		2 Sub-heat exchangers	
HEs			Series arrangement		Parallel arrangement	
Intermediate			T_{s1} = 305.5 K		x_{s1} = 0.5353	x_{s2} = 0.4647
Pattern	Triangular	Triangular	Triangular	Triangular	Triangular	Triangular
Tube passes	2	2	1	1	1	1
Shell passes	1	1	1	1	1	1
D_s (m)	0.387	0.5368	0.4576	0.3203	0.3273	0.2727
L (m)	5.904	2.438	2.438	2.438	4.877	8.53
B (m)	0.305	0.580	0.807	0.817	0.6095	0.5924
d_o (m)	0.019	0.01588	0.01588	0.01905	0.01905	0.02540
P_t (m)	0.023	0.01985	0.01985	0.0238125	0.0238125	0.03175
t (m)	0.0019	0.001651	0.001651	0.001651	0.001829	0.001829
Cl (m)	0.004	0.00397	0.00397	0.0047625	0.0047625	0.00635
N_t	160	590	427	135	141	52
v_t (m/s)	2.436	0.9651	0.6665	1.349	0.7248	0.8550
Re_t	40207	13181	9103	23066	12114	20185
Pr_t	6.2	6.2	6.2	6.2	6.2	6.2
h_t (W/m^2 K)	9799	4852	3608	6063	3706	3947
f_t	0.005519	0.006899	0.007429	0.006168	0.007016	0.006335
ΔP_t (Pa)	65657	7303	1832	5740	2990	4540
a_s (m^2)	0.0217	0.06227	0.07386	0.05233	0.0399	0.03231
D_e (m)	0.01349	0.01128	0.01128	0.01353	0.01353	0.01804
v_s (m/s)	1.022	0.3562	0.3003	0.4239	0.2976	0.3190
Re_s	17155	4995	4211	7131	0.00017	0.00017
Pr_s	5.4	5.4	5.4	5.4	5.4	5.4
h_s (W/m^2 K)	6186	3755	3418	3807	3136	2863
f_s	0.3336	0.4014	0.4118	0.3805	0.4013	0.3803
ΔP_s (Pa)	88520	5071	2264	2401	3423	4190
U (W/m^2 K)	1230	966.6	869.2	1043	855.7	873
A (m^2)	56.35	71.71	51.87	19.61	76.16	41.14
C_i (€)	18,162	20,653	20,617		21,365	
C_o (€)	5141	444.7	445.3		257.9	
C_{od} (€)	31,589	2733	2736		1585	
C_{total} (€)	49,751	23,386	23,353		22,950	

a certain pattern, and then optimizes sub-HEs simultaneously. A genetic algorithm is used to optimize the parameters of each sub-HE. The total cost minimization of all sub shell-and-tube exchangers is set as the objective function. The main conclusions are as follows:

1. A heat exchanger design approach motivated by constructal theory is proposed and simplified. The problem formulations for both non-simplified and simplified methods are given. The successful applications of shell-and-tube heat exchangers demonstrate potential applicability in other engineering fields.
2. Three cases studies with the TEMA standards imposing a discrete parameter space were considered to compare the results obtained by the original, the conventional GA, the old constructal, and the new constructal values. The results demonstrated that the novel design approach has a great advantage in solving the MDNLP problem and reduces total cost compared to the other methods.

Conflict of Interest

We wish to draw the attention of the Editor to the following facts which may be considered as potential conflicts of interest and to significant financial contributions to this work.

We confirm that the manuscript has been read and approved by all named authors and that there are no other persons who satisfied the criteria for authorship but are not listed. We further confirm that the order of authors listed in the manuscript has been approved by all of us.

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Acknowledgments

This work is supported by the National Natural Science Foundation of China (No. 51036003) and the National Key Basic Research Development Program of China (No. 2013CB228302) and the China Scholarship Council.

References

[1] B.I. Master, K.S. Chunangad, V. Pushpanathan, Fouling mitigation using helixchanger heat exchangers, in: Proceedings of the ECI Conference on Heat Exchanger Fouling and Cleaning: Fundamentals and Applications, Santa Fe, USA, 2003, pp. 317–322.
[2] K. Muralikrishna, U.V. Shenoy, Heat exchanger design targets for minimum area and cost, Trans. IChemE A Chem. Eng. Res. Des. 78 (2000) 161–167.
[3] M. Picón-Núñez, G.T. Polley, G. Martínez-Rodríguez, Graphical tool for the preliminary design of compact heat exchangers, Appl. Therm. Eng. 61 (2013) 36–43.
[4] P.D. Chaudhuri, U.M. Diwekar, An automated approach for the optimal design of heat exchangers, Ind. Eng. Chem. Res. 36 (1997) 3685–3693.

[5] V.K. Patel, R.V. Rao, Design optimization of shell-and-tube heat exchanger using particle swarm optimization technique, Appl. Therm. Eng. 30 (2010) 1417–1425.
[6] B.V. Babu, S.A. Munawar, Differential evolution strategies for optimal design of shell-and-tube heat exchangers, Chem. Eng. Sci. 62 (2007) 3720–3739.
[7] M. Fesanghary, E. Damangir, I. Soleimani, Design optimization of shell and tube heat exchangers using global sensitivity analysis and harmony search algorithm, Appl. Therm. Eng. 29 (2009) 1026–1031.
[8] V.C. Mariani, A.R.K. Duck, F.A. Guerra, L.S. Coelho, R.V. Rao, A chaotic quantum-behaved particle swarm approach applied to optimization of heat exchangers, Appl. Therm. Eng. 42 (2012) 119–128.
[9] M.A.S.S. Ravagnani, A.P. Silva, E.C. Biscaia, J.A. Caballero, Optimal design of shell-and-tube heat exchangers using particle swarm optimization, Ind. Eng. Chem. Res. 48 (2009) 2927–2935.
[10] J.M. Ponce-Ortega, M. Serna-González, A. Jiménez-Gutiérrez, Use of genetic algorithms for the optimal design of shell-and-tube heat exchangers, Appl. Therm. Eng. 29 (2009) 203–209.
[11] M. Ghazi, P. Ahmadi, A.F. Sotoodeh, A. Taherkhani, Modeling and thermo-economic optimization of heat recovery heat exchangers using a multimodal genetic algorithm, Energy Convers. Manage. 58 (2012) 149–156.
[12] B.K. Soltan, M.S. Avval, E. Damangir, Minimizing capital and operating costs of shell and tube heat condensers using optimum baffle spacing, Appl. Therm. Eng. 24 (2004) 2801–2810.
[13] D. Eryener, Thermoeconomic optimization of baffle spacing for shell and tube heat exchangers, Energy Convers. Manage. 47 (2006) 1478–1489.
[14] R. Selbas, O. Kizilkan, M. Reppich, A new design approach for shell-and-tube heat exchangers using genetic algorithms from economic point of view, Chem. Eng. Process. 45 (2006) 268–275.
[15] A.C. Caputo, P.M. Pelagagge, P. Salini, Heat exchanger design based on economic optimization, Appl. Therm. Eng. 28 (2008) 1151–1159.
[16] A.S. Sahin, B. Kilic, U. Kilic, Design and economic optimization of shell and tube heat exchangers using artificial bee colony (ABC) algorithm, Energy Convers. Manage. 52 (2011) 3356–3362.
[17] A. Hadidi, M. Hadidi, A. Nazari, A new design approach for shell-and-tube heat exchangers using imperialist competitive algorithm (ICA) from economic point of view, Energy Convers. Manage. 67 (2013) 66–74.
[18] A. Hadidi, A. Nazari, Design and economic optimization of shell-and-tube heat exchangers using biogeography-based (BBO) algorithm, Appl. Therm. Eng. 51 (2013) 1263–1272.
[19] A.V. Azad, M. Amidpour, Economic optimization of shell and tube heat exchanger based on constructal theory, Energy 36 (2011) 1087–1096.
[20] J. Yang, A.W. Fan, W. Liu, A.M. Jacobi, Optimization of shell-and-tube heat exchangers conforming to TEMA standards with designs motivated by constructal theory, Energy Convers. Manage. 78 (2014) 468–476.
[21] Y. Özçelik, Exergetic optimization of shell and tube heat exchangers using a genetic based algorithm, Appl. Therm. Eng. 27 (2007) 1849–1856.
[22] J.F. Guo, L. Cheng, M.T. Xu, Optimization design of shell-and-tube heat exchanger by entropy generation minimization and genetic algorithm, Appl. Therm. Eng. 29 (2009) 2954–2960.
[23] J.F. Guo, M.T. Xu, L. Cheng, The application of field synergy number in shell-and-tube heat exchanger optimization design, Appl. Energy 86 (2009) 2079–2087.
[24] J.F. Guo, M.T. Xu, The application of entransy dissipation theory in optimization design of heat exchanger, Appl. Therm. Eng. 36 (2012) 227–235.
[25] Q. Chen, Enstrasy dissipation-based thermal resistance method for heat exchanger performance design and optimization, Int. J. Heat Mass Transfer 60 (2013) 156–162.
[26] P. Wildi-Tremblay, L. Gosselin, Minimizing shell-and-tube heat exchanger cost with genetic algorithms and considering maintenance, Int. J. Energy Res. 31 (2007) 867–885.
[27] B. Allen, L. Gosselin, Optimal geometry and flow arrangement for minimizing the cost of shell-and-tube condensers, Int. J. Energy Res. 32 (2008) 958–969.
[28] Y. Li, X.M. Jiang, X.Y. Huang, J.G. Jia, J.H. Tong, Optimization of high-pressure shell-and-tube heat exchanger for syngas cooling in an IGCC, Int. J. Heat Mass Transfer 53 (2010) 4543–4551.
[29] A. Agarwal, S.K. Gupta, Jumping gene adaptations of NSGA-II and their use in multi-objective optimal design of shell and tube heat exchangers, Chem. Eng. Res. Des. 86 (2008) 123–139.
[30] S. Sanaye, H. Hajabdollahi, Multi-objective optimization of shell and tube heat exchangers, Appl. Therm. Eng. 30 (2010) 1937–1945.
[31] S. Fettaka, J. Thibault, Y. Gupta, Design of shell-and-tube heat exchangers using multiobjective optimization, Int. J. Heat Mass Transfer 60 (2013) 343–354.
[32] R.V. Rao, V. Patel, Multi-objective optimization of heat exchangers using a modified teaching–learning-based optimization algorithm, Appl. Math. Model. 37 (2013) 1147–1162.
[33] A. Gomez, L. Pibouleau, C. Azzaro-Pantel, S. Domenech, C. Latge, D. Haubensack, Multiobjective genetic algorithm strategies for electricity production from generation IV nuclear technology, Energy Convers. Manage. 51 (2010) 859–871.
[34] S. Belanger, L. Gosselin, Multi-objective genetic algorithm optimization of thermoelectric heat exchanger for waste heat recovery, Int. J. Energy Res. 36 (2012) 632–642.
[35] R. Hilbert, G. Janiga, R. Baron, D. Thévenin, Multi-objective shape optimization of a heat exchanger using parallel genetic algorithms, Int. J. Heat Mass Transfer 49 (2006) 2567–2577.
[36] L. Gosselin, M. Tye-Gingras, F. Mathieu-Potvin, Review of utilization of genetic algorithms in heat transfer problems, Int. J. Heat Mass Transfer 52 (2009) 2169–2188.
[37] A.L.H. Costa, E.M. Queiroz, Design optimization of shell-and-tube heat exchangers, Appl. Therm. Eng. 28 (2008) 1798–1805.
[38] F.T. Mizutani, F.L.P. Pessoa, E.M. Queiroz, Mathematical programming model for heat-exchanger network synthesis including detailed heat-exchanger designs. 1. Shell-and-tube heat-exchanger design, Ind. Eng. Chem. Res. 42 (2003) 4009–4018.
[39] M.A.S.S. Ravagnani, J.A. Caballero, A MINLP model for the rigorous design of shell and tube heat exchangers using the TEMA standards, Trans. IChemE A Chem. Eng. Res. Des. 85 (2007) 1423–1435.
[40] V.C. Onishi, M.A.S.S. Ravagnani, J.A. Caballero, Mathematical programming model for heat exchanger design through optimization of partial objectives, Energy Convers. Manage. 74 (2013) 60–69.
[41] J.M. Ponce-Ortega, M. Serna-Gonzalez, L.I. Salcedo-Estrada, A. Jimenez-Gutierrez, Minimum-investment design of multiple shell and tube heat exchangers using a MINLP formulation, Trans. IChemE A Chem. Eng. Res. Des. 84 (2006) 905–910.
[42] L. Lu, W.J. Cai, Y.S. Chai, L.H. Xie, Global optimization for overall HVAC systems-Part I problem formulation and analysis, Energy Convers. Manage. 46 (2005) 999–1014.
[43] S.S. Rao, Engineering Optimization, third ed., Wiley, New York, 1996. pp. 687–706.
[44] R. Smith, Chemical Process Design and Integration, first ed., Wiley, UK, 2005.
[45] TEMA, Standards of The Tubular Exchanger Manufacturers Association, ninth ed., Tubular Exchanger Manufacturers Association, New York, 2007.
[46] A. Bejan, Advanced Engineering Thermodynamics, second ed., Wiley, New York, 1997.
[47] A. Bejan, Shape and Structure, from Engineering to Nature, first ed., Cambridge University Press, Cambridge, 2000.
[48] C.J. Pennycuick, Animal Flight, first ed., Edward Arnold, London, 1976.
[49] H. Tennekes, The Simple Science of Flight, first ed., The MIT Press, Cambridge, 1996.
[50] A. Bejan, Constructal-theory network of conducing paths for cooling a heat generation volume, Int. J. Heat Mass Transfer 44 (799–811) (1997) 813–816.
[51] J.V.C. Vargas, J.C. Ordonez, A. Bejan, Constructal flow structure for a PEM fuel cell, Int. J. Heat Mass Transfer 47 (2004) 4177–4193.
[52] V.A.P. Raja, T. Basak, S.K. Das, Heat transfer and fluid flow in a constructal heat exchanger, in: Proceedings of Fifth International Conference on Enhanced, Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology, Hoboken, New Jersey, USA, 2005, pp. 147–153.
[53] A. Bejan, S. Lorente, Constructal theory of generation of configuration in nature and engineering, J. Appl. Phys. 100 (2006) 041301.
[54] A.H. Reis, Constructal theory: from engineering to physics, and how flow systems develop shape and structure, Appl. Mech. Rev. 59 (2006) 269–282.
[55] A. Bejan, S. Lorente, Design with Constructal Theory, first ed., Wiley, New Jersey, 2008.
[56] A. Bejan, J.H. Marden, The constructal unification of biological and geophysical design, Phys. Life Rev. 6 (2009) 85–102.
[57] A. Bejan, S. Lorente, A.F. Miguel, A.H. Reis, Constructal Human Dynamics, Security and Sustainability, first ed., IOS Press, Amsterdam, 2009.
[58] A. Bejan, S. Lorente, The constructal law and the evolution of design in nature, Phys. Life Rev. 8 (2011) 209–240.
[59] L.G. Chen, Progress in study on constructal theory and its applications, Sci. China Ser. E 55 (2012) 802–820.
[60] L.A.O. Rocha, S. Lorente, A. Bejan, Constructal Law and the Unifying Principle of Design, first ed., Springer, New York, 2013.
[61] A. Bejan, S. Lorente, Constructal law of design and evolution: physics, biology, technology, and society, J. Appl. Phys. 113 (2013) 151301.
[62] A. Bejan, S. Lorente, D.H. Kang, Constructal design of regenerators, Int. J. Energy Res. 37 (2013) 1509–1518.
[63] J.H. Holland, Adaptation in Natural and Artificial Systems, second ed., University of Michigan Press, Ann Arbor, 1975.
[64] R.K. Sinnot, J.M. Coulson, J.F. Richardson, Coulson and Richardson's Chemical Engineering: Chemical Engineering Design, vol. 6, Elsevier, Oxford, 2005. fourth ed.
[65] R.K. Shah, D.P. Sekulic, Fundamentals of Heat Exchanger Design, Wiley, New Jersey, 2003.
[66] D.Q. Kern, Process Heat Transfer, first ed., McGraw-Hill, New York, 1950.
[67] M.S. Peters, K.D. Timmerhaus, Plant Design and Economics for Chemical Engineers, fourth ed., McGraw-Hill, New York, 1991.
[68] G. Towler, R. Sinnott, Chemical Engineering Design: Principle, Practice and Economics of Plant and Process Design, first ed., Elsevier, Amsterdam, 2008.
[69] M. Taal, I. Bulatov, J. Klemes, P. Stehlik, Cost estimation and energy price forecast for economic evaluation of retrofit projects, Appl. Therm. Eng. 23 (2003) 1819–1835.