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# Optimization of shell-and-tube heat exchangers conforming to TEMA standards with designs motivated by constructal theory

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ABSTRACT

A modified optimization design approach motivated by constructal theory is proposed for shell-and-tube heat exchangers in the present paper. In this method, a shell-and-tube heat exchanger is divided into several in-series heat exchangers. The Tubular Exchanger Manufacturers Association (TEMA) standards are rigorously followed for all design parameters. The total cost of the whole shell-and-tube heat exchanger is set as the objective function, 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 cost function by adjusting parameters such as the tube and shell diameters, tube length and tube arrangement. Three cases are studied which indicate that the modified design approach can significantly reduce the total cost compared to the original design method and traditional genetic algorithm design method.



## 1. Introduction

Shell-and-tube heat exchangers are widely used in various industrial fields such as the petro-chemical industry, manufacturing industry, food preservation, electrical power production and energy conservation systems due to their structural simplicity, relatively low cost and design adaptability. 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 significance to improve their thermal–hydraulic performance and reduce their cost as much as possible.

The design of shell-and-tube heat exchangers aims at selecting the suitable operational and geometric parameters such as fluid temperature, flow rate, flow arrangement, heat exchanger materials, tube length, shell and tube diameters, tube and baffle numbers. The traditional design approach is an iterative process based on the past experience and the constraints of working conditions, such as allowable fouling and pressure drops. In general, a reference geometric configuration is recommended at first. Then, the values of the working and geometric variables are adjusted based on the design specifications and requirements. The final design result is chosen after a significant amount of trial-and-error design until the heat transfer capacity and pressure drops are within the allowable values. The traditional design approach is not cost-effective due to the lack of evaluation criteria. In the pursuit of improved designs, considerable efforts using different optimization methodologies have been devoted to optimizing heat transfer processes, e.g., evolutionary algorithms [2–18] and mathematical programming [19–24]. The design procedure can be directed at optimizing different objectives such as total cost minimization [2–16], exergetic cost minimization [25], entropy generation minimization (EGM) [26], field synergy number maximization [27], or entransy generation minimization [28] by changing the design parameters. Other studies have also been dedicated, such as single geometric parameter optimization [29,30], optimization design considering maintenance [31], graphic tools for a preliminary design [32,33] and phase-changing optimization design [34].

Attention has been directed at designing shell-and-tube heat exchangers with single-objective function optimization over the past years. Selbas et al. [2] utilized a genetic algorithm (GA) for the design optimization of shell-and-tube heat exchangers. Their results demonstrated that a GA 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. Caputo et al. [3] applied a GA to the designing process as well and they analyzed this design approach from the economic point of view. They presented three cases, which demonstrated that the evident cost reductions are feasible with respect to the original design values. Later, other evolutional algorithms with single- or multi-objective function have been adopted. Patel and Rao [4] used particle swarm optimization (PSO) for designing shell-and-tube heat exchangers from the perspective of economics. They reported that the PSO method had

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**Nomenclature**

| | |
|---|---|
| $a_1$ | 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 (€ $KW^{-1}$ $h^{-1}$) |
| $C_o$ | 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) |
| $m$ | mass flow rate (kg $s^{-1}$) |
| $n$ | tube quantity (–) |
| $n_1$ | numerical constant (–) |
| $n_y$ | equipment life (year) |
| $N_p$ | tube passes number (–) |
| $P$ | pumping power (W) |
| $Pr$ | Prandtl number (–) |
| $P_t$ | tube pitch (m) |
| $Q$ | heat duty (J) |
| $Re$ | Reynolds number (–) |
| $R$ | fouling coefficient ($m^2$ K $W^{-1}$) |
| $T$ | temperature (K) |
| $v_t$ | tube-side fluid velocity (m $s^{-1}$) |

*Greek symbols*

| | |
|---|---|
| $\Delta T_{lm}$ | log mean temperature difference (K) |
| $\Delta P$ | pressure drop (Pa) |
| $\lambda$ | thermal conductivity (W $m^{-1}$ $K^{-1}$) |
| $\mu$ | dynamic viscosity (Pa $s^{-1}$) |
| $\rho$ | density (kg $m^{-3}$) |
| $\pi$ | numerical constant (–) |
| $\eta$ | pump efficiency (–) |

*Subscripts*

| | |
|---|---|
| $i$ | inlet |
| $o$ | outlet |
| $s$ | shell-side |
| $t$ | tube-side |
| $w$ | tube wall |

the features of simple concept, fast convergence and easy implementation to various thermal systems. Babu and Munawar [5] used differential evolution (DE), an improved and simpler GA, to design heat exchangers. They concluded that DE is much faster than a GA on convergence and this method can yield the global optimization. Fesanghary et al. [6] investigated the optimization design using global sensitivity analysis (GSA) and harmony search algorithm (HSA). Non-influential geometric parameters that have the least impact on the objective function of a heat exchanger were identified using GSA. After that, HSA was employed to adjust the most influential parameters in order to minimize the total cost. An illustrative case was studied which revealed that HSA can converge to the optimum solution with a higher accuracy in comparison to a GA. Shahin et al. [7] developed a method using artificial bee colony algorithm (ABC) for designing heat exchangers. It was shown that ABC can be successfully utilized for optimal design of shell-and-tube heat exchangers and it is the more accurate and quick method as compared to the conventional trial-and-error method and a conventional GA method. Hadidi et al. [8] proposed a design approach based on imperialist competitive algorithm (ICA) and came to a conclusion that the developed ICA technique reduced both the capital investment cost and operating cost, allowed for rapid solutions with good quality and gave the designer more freedom in the final choice with respect to the GA approach. Haddid and Nazari [9] later developed a design method based on biogeography-based optimization algorithm (BBO). The comparisons with GA, PSO, ABC and BBO demonstrated that all of the evolutionary algorithms showed a dramatic total cost reduction respect to the primary design, but the differences of the total cost between different algorithm applications were not remarkable for most case studies. Recently, Mariani et al. [10] presented a new quantum particle swarm optimization (QPSO) approach combining with Zaslavskii chaotic map sequences (QPSOZ) [35]. Two case studies displayed that better results were obtained using the QPSOZ method in comparison with those obtained by GA, PSO and classical QPSO. Azad and Amidpour [11] employed a new approach based on constructal theory to design and optimize shell-and-tube heat exchangers using GA. In this method, a double-branch constructal shell-and-tube heat exchanger was used to achieve the "tree-like network". A double-branch constructal shell-and-tube heat exchanger is a heat exchanger with two in-series sections and the tube number of the second section is twice the tube number of the first section [11]. It was demonstrated that a large reduction (more than 50%) in total cost was achieved as compared to the original design. Mizutani et al. [19] used generalized disjunctive programming (GDP) for optimization problem formulation and mixed-integer nonlinear programming (MINLP) for its solution. This study took the fluid allocation into consideration and the results indicated that the methodology properly accounted for the trade-offs between area and pumping costs. Other studies about designing shell-and-tube heat exchangers with multi-objective optimization approaches [12–17] have been conducted as well.

It should be noticed that the overwhelming majority of the above-mentioned efforts on design optimization used continuous values to determine mechanical parameters such as tube diameter, thickness and length. According to Smith [36], this type of approach provides just a preliminary specification for 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) [37] for the case of shell-and-tube heat exchangers. In open literature, limited data can be found on design procedure rigorously following the TEMA standards. Ravagnani et al. [18] solved a shell-and-tube heat exchanger design problem with PSO, strictly following the TEMA standards and respecting pressure drops and fouling limits. The results showed the benefits of avoiding local minima and finding the global optima as compared to previous approaches. Ravagnani et al. [20] presented a model based on GDP and optimized it with a MINLP formulation, following the TEMA standards. Three case studies proved that this model achieved more realistic results than other results reported in the literature. Onishi et al. [21] also applied a sequential model based on GDP and MINLP formulation into optimization designing with the industrial standards. The shell-side Reynolds number maximization, the shell-side pressures drop minimization, the tube-side Reynolds number maximization, the tube-side pressure drop minimization and the heat transfer area minimization were optimized in order. The proposed approach provided a good solution compared to other reports in literature, but it failed to guarantee a global optimal solution even in the case in which each of the sub-problems is solved to achieve a global optimization.

So far, various optimization design algorithms have been developed to optimize shell-and-tube heat exchangers. Although the optimized designs may vary from algorithm to algorithm, the differences between the results obtained by using various algorithms are generally not significant, as shown in [7–9]. Moreover, it is readily anticipated that the results using different evolutionary algorithms (GA, PSO, ABC, BBO, QPSOZ and so on) may remain unchanged if the TEMA standards are taken into account. In other words, design approaches using different evolutionary algorithms seldom differ in a practical sense. Therefore, it is of great importance to explore new approaches instead of new algorithms to design shell-and-tube heat exchangers accompanying with evident advantages on the reduction of economics or energy consumption. In this paper, a novel shell-and-tube heat exchanger based on constructal theory [11] is utilized to design heat transfer process. A modified optimization design approach for shell-and-tube heat exchangers motivated by constructal theory is proposed. The TEMA standards are followed. The genetic algorithm was applied to find the optimal values. Three cases were studied and they demonstrated an advantage in total cost minimization in comparison to data from the literature and with respect to results obtained by the conventional GA approach.

## 2. The modified design approach based on constructal theory

Constructal theory, proposed by Bejan [38], has been used to interpret some natural phenomena, such as the geometric features of rivers, clouds, veins and arteries, and trees [39–41], and it can be adopted as an optimization design method for engineering applications (such as fins of cooling devices, fuel cells, and plate heat exchangers) [42–44]. For an engineering system like a shell-and-tube heat exchanger, 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 fluids, heat) that flow through it. On this basis, Azad [11] proposed a novel heat exchanger, a 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, imitating a "tree-like network" configuration as shown in Fig. 1.

According to the definition, a constructal shell-and-tube heat exchanger [11] is a shell-and-tube heat exchanger with two or more bundles in-series sections, the tube number of the latter bundle being twice that of the former section as shown in Fig. 2. Azad

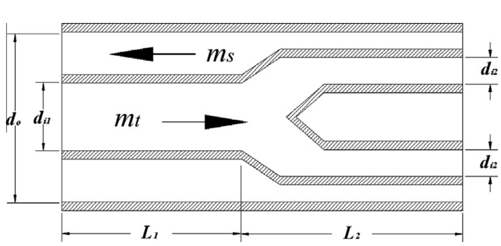


**Fig. 1.** Tree-shaped structure for a heat exchanger based on constructal theory.

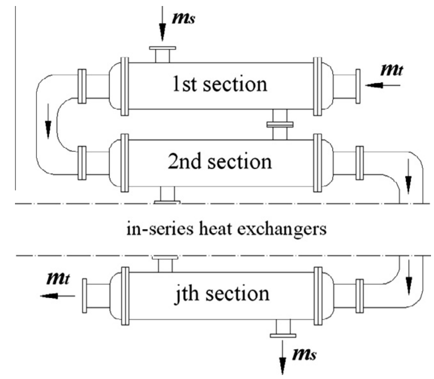


**Fig. 2.** The in-series shell-and-tube heat exchangers.

and Amidpour used this methodology in design optimization with a GA and the results demonstrated a total cost reduction compared to the original design. However, their design values did not take the TEMA standards into consideration. Furthermore, the requirement that the number of tubes in the second bundle must be twice that of the first bundle, which caters to constructal theory, is neither logical nor reasonable for realistic shell-and-tube heat exchanger design. In fact, their design approach yielded an unpractical shell-and-tube heat exchanger with the shell length of 0.161 m, shell diameter of 1.328 m and tube number of 3988. Furthermore, this design approach failed to provide a better solution than the conventional GA approach ([3]) from the economic point of view. Therefore, in order to design a heat exchanger in a better way, the concept of a constructal shell-and-tube heat exchanger, such that the heat exchanger is split into a series of two shell-and-tube heat exchangers, is retained, but there is no constraint on the number of tubes in each bundle. In other words, the shell-and-tube heat exchanger designed in this paper includes several sub-heat exchangers that are optimized together. Meanwhile, the TEMA standards are rigorously followed during the design procedure of each sub-unit, which is more realistic for an engineering design. The minimization of total heat exchanger cost, including the investment cost and operational cost, was taken as the objective function. A genetic algorithm was used to optimize the objective function by adjusting mechanical and flow parameters. In order to demonstrate the novelty and improvement of this modified design approach, the original design values and the values using the traditional GA approach, which meet the TEMA standards as well, were taken as the reference groups.

## 3. Heat exchanger calculation

### 3.1. Heat transfer, pressure drop and cost calculation

The energy balance equation for a shell-and-tube heat exchanger is shown in Eq. (1):

$$Q = (mc_p)_t(T_{t,i} - T_{t,o}) = (mc_p)_s(T_{s,o} - T_{s,i}) \tag{1}$$

where $Q$ is the heat duty, $m$ 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:

$$D_s = \left(\frac{n}{K_1}\right)^{1/n_1} \cdot d_o \tag{2}$$

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

**Table 1**
Values of $K_1$ and $n_1$ coefficients [45].

| Number of passes | Triangle tube pitch | | Square tube pitch | |
|---|---|---|---|---|
| | $K_1$ | $n_1$ | $K_1$ | $n_1$ |
| 1 | 0.319 | 2.142 | 0.215 | 2.207 |
| 2 | 0.249 | 2.207 | 0.156 | 2.291 |
| 4 | 0.175 | 2.285 | 0.158 | 2.263 |
| 6 | 0.0743 | 2.499 | 0.0402 | 2.617 |
| 8 | 0.0365 | 2.675 | 0.0331 | 2.643 |

number of the tubes, which can be found in Table 1 [45]. The tube pitch and inner diameter are calculated as follows [45]:

$$P_t = 1.25d_o \tag{3}$$

$$d_i = d_o - 2t \tag{4}$$

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

$$A = \frac{Q}{K\Delta T_{lm}F} \tag{5}$$

where $A$ is the heat transfer area based on the outer diameter of the tube, $\Delta T_{\rm lm}$ is the logarithmic mean temperature difference (LMTD), $F$ is the correction factor for LMTD according to the equipment architecture [46], and $K$ is the overall heat transfer coefficient based on the outer diameter of the tube. The LMTD and $F$ are computed through Eqs. (6) and (7), respectively [47]:

$$\Delta T_{lm} = \frac{(T_{s,i} - T_{t,o}) - (T_{s,o} - T_{t,i})}{\ln((T_{s,i} - T_{t,o})/(T_{s,o} - T_{t,i}))} \tag{6}$$

$$F = \begin{cases} 1 & tube\ pass = 1 \\ \frac{\sqrt{R^2+1}}{R-1} \cdot \frac{\ln\left(\frac{1-P}{1-PR}\right)}{\ln\left[\frac{2-P(R+1-\sqrt{R^2+1})}{2-P(R+1+\sqrt{R^2+1})}\right]} & tube\ pass = even\ number \end{cases} \tag{7}$$

where

$$R = \frac{T_{s,i} - T_{s,o}}{T_{t,o} - T_{t,i}} \tag{8}$$

and

$$P = \frac{T_{t,o} - T_{t,i}}{T_{s,i} - T_{t,i}} \tag{9}$$

The overall heat transfer coefficient is calculated using [46]:

$$K = \left[\frac{1}{h_t}\left(\frac{d_o}{d_i}\right) + R_t\left(\frac{d_o}{d_i}\right) + \frac{d_o \ln(d_o/d_i)}{2\lambda_w} + R_s + \frac{1}{h_o}\right]^{-1} \tag{10}$$

where $h_{\rm t}$ and $h_{\rm s}$ are the heat transfer coefficients, while $R_{\rm t}$ and $R_{\rm s}$ are fouling coefficients. $\lambda_{\rm w}$ is the thermal conductivity for tube wall at the bulk mean temperature of fluid. The tube number is calculated as:

$$n = \frac{A}{\pi d_o L} \tag{11}$$

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. (12) [46]:

$$h_t = 0.023\frac{\lambda_t}{d_i}\mathrm{Re}_t^{0.8}\mathrm{Pr}_t^{\frac{1}{3}}\left(\frac{\mu_t}{\mu_{tw}}\right)^{0.14} \tag{12}$$

where $\mu_{\rm t}$ is the fluid dynamic viscosity at the bulk temperature of tube-side $T_{\rm b,t}$, and $\mu_{\rm tw}$ is fluid dynamic viscosity at the inner tube wall temperature $T_{\rm tw}$, which can be estimated through Eq. (13) [45]:

$$h_t(T_{tw} - T_{b,t}) = K(T_{b,s} - T_{b,t}) \tag{13}$$

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

$$Re_t = \frac{\rho_t v_t d_i}{\mu_t} \tag{14}$$

The tube-side fluid velocity is calculated as:

$$v_t = \frac{N_p}{n} = \frac{m_t}{\pi(d_i^2/4)\rho_t} \tag{15}$$

where $N_p$ is the tube pass number. Kern's method is utilized to calculate the shell-side heat transfer coefficient expressed in Eq. (16) [47]:

$$h_s = 0.36\frac{\lambda_s}{D_e}\mathrm{Re}_s^{0.55}\mathrm{Pr}_s^{\frac{1}{3}}\left(\frac{\mu_s}{\mu_{sw}}\right)^{0.14} \tag{16}$$

where $D_e$ is the shell-side hydraulic diameter, $\mu_s$ is the dynamic viscosity coefficient at the bulk temperature of shell-side fluid, while $\mu_{\rm sw}$ is the dynamic viscosity coefficient at outer tube wall temperature, which can be estimated through Eq. (17) [45]:

$$h_s(T_{b,s} - T_{tw}) = K(T_{b,s} - T_{b,t}) \tag{17}$$

The shell-side hydraulic diameter and Reynolds number are calculated by Eqs. (18) and (19), respectively [45,47]:

$$F = \begin{cases} \frac{4(P_t^2 - \pi d_o^2/4)}{\pi d_o} & for\ square\ arrangement \\ \frac{4(P_t^2/2\times 0.87P_t - (0.5\pi d_o^2/4))}{\pi d_o} & for\ triangle\ arrangement \end{cases} \tag{18}$$

$$\mathrm{Re}_s = \frac{m_s \cdot D_e}{\mu_s \cdot A_s} \tag{19}$$

where $B$ is the baffle spacing, $A_s$ is the cross area of fluid flow which is calculated as below [45,47]:

$$A_s = \frac{D_s \cdot B(P_t - d_o)}{P_t} \tag{20}$$

The tube-side pressure drop can be obtained through Eq. (21) [45]:

$$\Delta P_t = N_p\left(4f_t\frac{L}{d_i} + 2.5\right)\frac{\rho_t v_t^2}{2} \tag{21}$$

Here $f_t$ is the friction factor for turbulent tube flow that is expressed in Eq. (22):

$$f_t = 0.046(\mathrm{Re}_t)^{-0.2} \tag{22}$$

The shell-side pressure drop is calculated by Eq. (23) [48]:

$$\Delta P_s = f_s \cdot \left(\frac{\rho_s v_s^2}{2}\right) \cdot \left(\frac{L}{B}\right) \cdot \left(\frac{D_s}{D_e}\right) \tag{23}$$

where $f_s$ is the friction factor for shell-side which is expressed in Eq. (24):

$$f_s = 2b_o\mathrm{Re}_s^{-0.15} \tag{24}$$

where $b_o = 0.72$ [48] is valid for $Re_s < 40{,}000$. The total power consumption is calculated through Eq. (25) [49]:

$$P = \frac{1}{\eta}\left(\frac{m_s}{\rho_s}\Delta P_s + \frac{m_t}{\rho_t}\Delta P_t\right) \tag{25}$$

Here $\eta$ is the pump efficiency and we give it a constant of 0.7. The total cost is obtained through Eq. (26) [3]:

$$C_{tot} = C_i + C_{\rm od} \tag{26}$$

where $C_i$ and $C_{\rm od}$ are the capital investment cost and the total discounted operating cost [50] which can be calculated through Eqs. (27) and (28), respectively.

$$C_i = a_1 + a_2 A^{a3} \tag{27}$$

$$C_{od} = \sum_{k=1}^{ny} \frac{C_o}{(1+i)^k} \tag{28}$$

Here, $a_1 = 8000$, $a_2 = 259.2$ and $a_3 = 0.91$ for shell-and-tube heat exchangers made of stainless steel [50]. $i$ 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. (29):

$$C_o = P \cdot C_E \cdot H \tag{29}$$

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.

### 3.2. Equations for in-series shell and tube heat exchangers

The overall heat transfer area of a shell-and-tube heat exchanger $A$ is the sum of several in-series heat exchangers:

$$A = \sum_{j=1}^{N} A_j \tag{30}$$

where $N$ is the number of in-series heat exchangers, $A_j$ is the heat transfer area of the $j$th section.

Similarly, the overall power consumption is calculated as follows:

$$P = \sum_{j=1}^{N} P_j \tag{31}$$

where $P_j$ is the power consumption for the $j$th section.

## 4. Optimization method

### 4.1. Genetic algorithm

Holland [51] first proposed the principles of the 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, and 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 of a shell-and-tube heat exchanger, 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. 3. Due to the length limitation of the context, the detailed information of genetic algorithm will not be given. Further information of GA utilization in heat transfer problems could be found in [52].

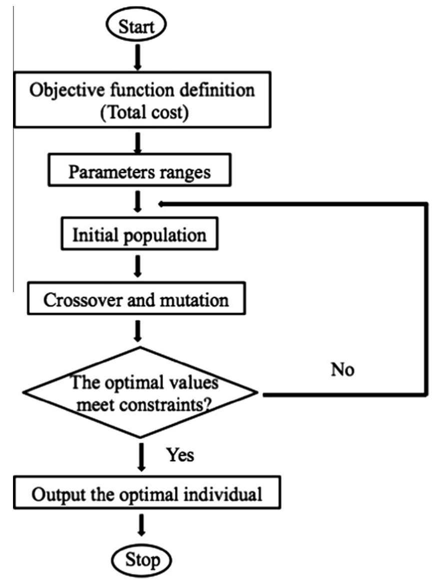


**Fig. 3.** Flow chart of the genetic algorithm using the total cost as objective function.

### 4.2. The design parameters, constraint conditions and objective function

The objective function of this modified optimization design approach for a shell-and-tube heat exchanger is the minimization of total cost including initial investment cost and power consumption cost. This study considers discrete decision variables according to the TEMA design standards, which are as follows:

1. The tube layout adopts two arrangements: triangular arrangement (30°) or square arrangement (90°).
2. The tube passes number ($N_p$) 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 $29.5 \times do^{0.75}$ 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. The range of thickness is presented in Table 2 [31].

In this paper, the allowable maximum pressure drops for shell-side and tube-side are both 70,000 Pa. The optimization was performed on a personal computer with Intel Xeon CPU E5630 of 2.53 GHz and 14.00 GB of RAM using the genetic algorithm toolbox solver in Engineering Equation Solver (EES). For the genetic algorithm parameters setting, the initial population, maximum generation and mutation probability were set to 50, 500 and 0.3265, respectively.

**Table 2**
Values of the BWG wall thicknesses [31].

| BWG | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|---|
| $t$ (mm) | 4.572 | 4.191 | 3.759 | 3.404 | 3.048 | 2.769 | 2.413 | 2.108 | 1.829 | 1.651 |

**Table 3**
Optimal parameters using three design approaches with continuous values [3,11,45].

| | Case study | | | |
|---|---|---|---|---|
| | Original values | GA values | Constructal-motivated design | |
| | | | 1st Part | 2nd Part |
| Pitch pattern | Triangular | Triangular | Triangular | Triangular |
| Tube passes | 2 | 2 | 1 | 1 |
| Shell passes | 1 | 1 | 1 | 1 |
| $D_s$(m) | 0.387 | 0.620 | 0.38994 | 0.48946 |
| $L$(m) | 4.880 | 1.548 | 1.4409 | 1.4135 |
| $B$(m) | 0.305 | 0.440 | 0.5 | 0.5 |
| $d_o$(m) | 0.019 | 0.016 | 0.0166 | 0.01508 |
| $P_t$(m) | 0.023 | 0.020 | 0.02077 | 0.01886 |
| $t$(m) | 0.0038 | 0.0032 | 0.00519 | 0.0047 |
| $Cl$(m) | 0.004 | 0.004 | 0.0041 | 0.00377 |
| $N_t$ | 160 | 803 | 275 | 550 |
| $v_t$(m/s) | 1.76 | 0.68 | 0.92612 | 0.5614 |
| $Re_t$ | 36,400 | 9487 | 13,368 | 7359.6 |
| $Pr_t$ | 6.2 | 6.2 | 6.2 | 6.2 |
| $h_t$(W/m$^2$ K) | 6558 | 6043 | 4625.7 | 2903.4 |
| $f_t$ | 0.023 | 0.031 | 0.0290 | 0.0343 |
| $\Delta P_t$(Pa) | 62,812 | 3673 | 2420.4 | 1026.2 |
| $a_s$(m$^2$) | 0.0236 | 0.0541 | 0.03914 | 0.04905 |
| $D_e$(m) | 0.013 | 0.015 | 0.01181 | 0.01072 |
| $v_s$(m/s) | 0.94 | 0.41 | 0.56883 | 0.45317 |
| $Re_s$ | 16,200 | 8039 | 8357.2 | 6046.8 |
| $Pr_s$ | 5.4 | 5.4 | 5.4 | 5.4 |
| $h_s$(W/m$^2$ K) | 5736 | 3476 | 4784.1 | 4408.8 |
| $f_s$ | 0.337 | 0.374 | 0.37158 | 0.39006 |
| $\Delta P_s$(Pa) | 67,684 | 4365 | 5696 | 5140.1 |
| $U$(W/m$^2$ K) | 1471 | 1121 | 1306.8 | 1097.4 |
| $A$(m$^2$) | 46.6 | 62.5 | 20.706 | 36.858 |
| $C_i$(€) | 16,549 | 19,163 | 18,360 | |
| $C_o$(€) | 4466 | 272 | 435 | |
| $C_{oD}$(€) | 27,440 | 1671 | 2671 | |
| $C_{total}$(€) | 43,989 | 20,834 | 21,031 | |

## 5. Results and discussion

The original design [45], that obtained using the GA [3], and the design approach based on constructal shell-and-tube heat exchanger [11] can be compared in Table 3, without imposing TEMA standards. The table demonstrates that the GA design approach and the constructal design approach can reduce the total cost compared to the original design approach. Quantitatively, the total costs are about 47.4% and 47.8% of the original one for the GA method and the design suggested by constructal theory, respectively. It is seen that the total cost of heat exchanger using GA is slightly lower than that using the design suggested by the constructal method. Therefore the design approach suggested by constructal theory failed to offer advantages over that found by the GA method when the design parameters take continuous values.

Three case studies [45,47] 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 4. The modified design approach is now implemented with two and more than two tube bundles (sections) in series. Fig. 4 gives the total costs of optimized heat exchangers with different section numbers. As shown in Fig. 4, the total cost increases with the number of series sections for each case. The

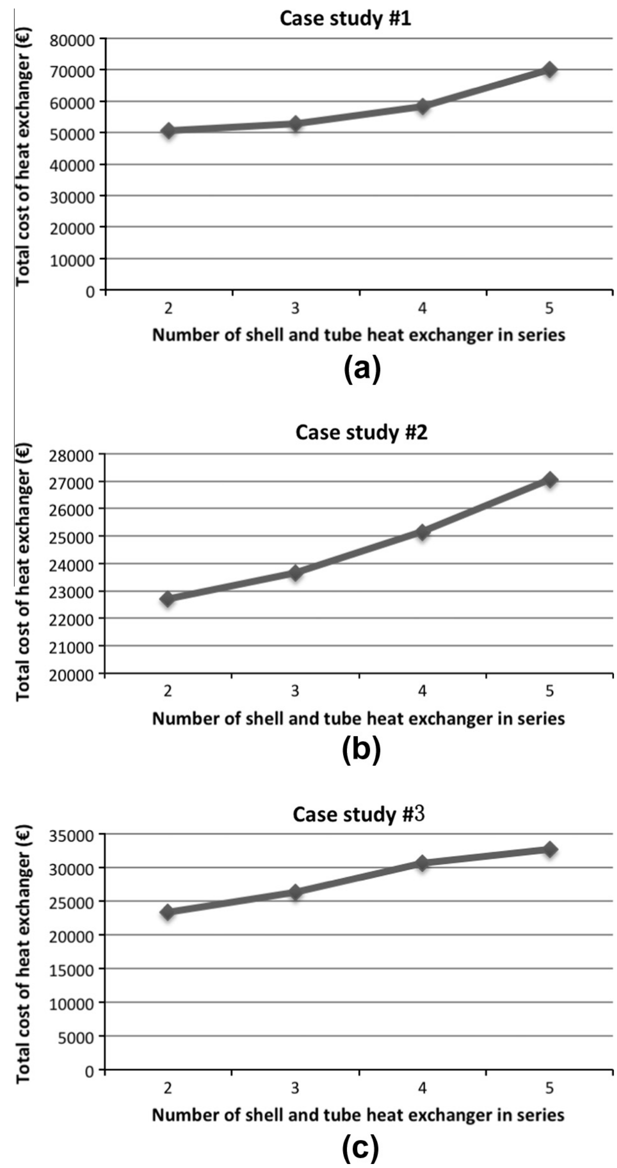


**Fig. 4.** Total costs of optimized in-series shell-and-tube heat exchangers with different numbers of sections: (a) the first case study, (b) the second case study and (c) the third case study.

**Table 4**
Case studies specifications [45,47].

| | Case study #1 | | Case study #2 | | Case study #3 | |
|---|---|---|---|---|---|---|
| | Shell-side: methanol | Tube-side: sea water | Shell-side: kerosene | Tube-side: crude oil | Shell-side: distilled water | Tube-side: 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 |
| $\rho$(kg/s) | 750 | 995 | 850 | 995 | 995 | 999 |
| $C_p$ (kJ/kg·K) | 2.84 | 4.20 | 2.47 | 2.05 | 4.18 | 4.18 |
| $\mu$(Pa s) | 0.00034 | 0.00080 | 0.00040 | 0.000358 | 0.00080 | 0.00092 |
| $\lambda$(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 |

**Table 5**
Case study #1: Optimal parameters of using three design approaches with discrete values.

| | Case study #1 | | | |
|---|---|---|---|---|
| | Original values | GA values | New values | |
| | | | 1st Part | 2nd Part |
| Pitch pattern | Triangular | Triangular | Triangular | Triangular |
| Tube passes | 2 | 2 | 1 | 1 |
| Shell passes | 1 | 1 | 1 | 1 |
| $D_s$(m) | 0.894 | 0.8229 | 0.6519 | 0.5115 |
| $L$(m) | 4.748 | 3.658 | 3.658 | 2.438 |
| $B$(m) | 0.356 | 0.5 | 0.777 | 0.768 |
| $d_o$(m) | 0.020 | 0.01588 | 0.01588 | 0.01588 |
| $P_t$(m) | 0.025 | 0.01985 | 0.01985 | 0.01985 |
| $t$(m) | 0.002 | 0.001651 | 0.001651 | 0.001651 |
| $Cl$(m) | 0.005 | 0.00397 | 0.00397 | 0.00397 |
| $N_t$ | 918 | 1514 | 911 | 542 |
| $v_t$(m/s) | 0.7507 | 0.7366 | 0.612 | 1.029 |
| $Re_t$ | 14939 | 11523 | 9575 | 16093 |
| $Pr_t$ | 5.7 | 5.7 | 5.7 | 5.7 |
| $h_t$(W/m$^2$ K) | 3878 | 4008 | 3456 | 5236 |
| $f_t$ | 0.006728 | 0.007087 | 0.007354 | 0.006629 |
| $\Delta P_t$(Pa) | 5880 | 5800 | 2060 | 4022 |
| $a_s$(m$^2$) | 0.05883 | 0.08229 | 0.1013 | 0.07857 |
| $D_e$(m) | 0.0142 | 0.01128 | 0.01128 | 0.01128 |
| $v_s$(m/s) | 0.6301 | 0.4504 | 0.3659 | 0.4717 |
| $Re_s$ | 19739 | 11203 | 9101 | 11734 |
| $Pr_s$ | 5.082 | 5.082 | 5.082 | 5.082 |
| $h_s$(W/m$^2$ K) | 1903 | 1755 | 1565 | 1800 |
| $f_s$ | 0.3266 | 0.3556 | 0.3669 | 0.3531 |
| $\Delta P_s$(Pa) | 37733 | 14445 | 5013 | 4245 |
| $U$(W/m$^2$ K) | 634.3 | 628.6 | 584.8 | 665.5 |
| $A$(m$^2$) | 273.7 | 276.1 | 166.1 | 65.9 |
| $C_i$(€) | 50,812 | 51,159 | 44,839 | |
| $C_o$(€) | 2167 | 1124 | 917.2 | |
| $C_{oD}$(€) | 13,315 | 6909 | 5636 | |
| $C_{total}$(€) | 64,127 | 58,069 | 50,475 | |

**Table 6**
Case study #2: Optimal parameters of using three design approaches with discrete values.

| | Case study #2 | | | |
|---|---|---|---|---|
| | Original values | GA values | New values | |
| | | | 1st Part | 2nd Part |
| Pitch pattern | Square | Square | Triangular | Triangular |
| Tube passes | 4 | 4 | 1 | 1 |
| Shell passes | 1 | 1 | 1 | 1 |
| $D_s$(m) | 0.539 | 0.765 | 0.2901 | 0.2932 |
| $L$(m) | 5.983 | 2.438 | 6.096 | 3.048 |
| $B$(m) | 0.127 | 0.138 | 0.304 | 0.367 |
| $d_o$(m) | 0.025 | 0.01905 | 0.01588 | 0.01905 |
| $P_t$(m) | 0.031 | 0.0238125 | 0.01985 | 0.0238125 |
| $t$(m) | 0.0025 | 0.001651 | 0.001651 | 0.001651 |
| $Cl$(m) | 0.006 | 0.0047625 | 0.00397 | 0.0047625 |
| $N_t$ | 158 | 673 | 161 | 111 |
| $v_t$(m/s) | 1.523 | 0.5768 | 0.9461 | 0.9120 |
| $Re_t$ | 8468 | 2525 | 3308 | 3901 |
| $Pr_t$ | 56.45 | 56.45 | 56.45 | 56.45 |
| $h_t$(W/m$^2$ K) | 1086 | 524.1 | 814.4 | 759.5 |
| $f_t$ | 0.007537 | 0.009601 | 0.009096 | 0.008801 |
| $\Delta P_t$(Pa) | 53,195 | 5594 | 8965 | 3920 |
| $a_s$(m$^2$) | 0.01344 | 0.02111 | 0.01764 | 0.02152 |
| $D_e$(m) | 0.02469 | 0.01881 | 0.01128 | 0.01353 |
| $v_s$(m/s) | 0.4831 | 0.3076 | 0.3682 | 0.3018 |
| $Re_s$ | 25344 | 12294 | 8823 | 8674 |
| $Pr_s$ | 7.6 | 7.6 | 7.6 | 7.6 |
| $h_s$(W/m$^2$ K) | 978.9 | 862.9 | 1199 | 990.6 |
| $f_s$ | 0.3146 | 0.3507 | 0.3686 | 0.3695 |
| $\Delta P_s$(Pa) | 25344 | 10134 | 10954 | 2575 |
| $U$(W/m$^2$ K) | 268.1 | 202.6 | 257.2 | 241.6 |
| $A$(m$^2$) | 74.21 | 98.18 | 48.87 | 20.31 |
| $C_i$(€) | 21,054 | 24,842 | 20,247 | |
| $C_o$(€) | 1452 | 205.8 | 397.6 | |
| $C_{oD}$(€) | 8920 | 1265 | 2443 | |
| $C_{total}$(€) | 29,974 | 26,106 | 22,690 | |

optimization results for this heat exchanger using a genetic algorithm indicates that a shell-and-tube heat exchanger with two in-series sections is preferred. Tables 5–7 present detailed comparisons of optimal parameters obtained by using different design methods. In all the tables, the first column gives the original design results ([45,47]), the second column gives the optimal result using the traditional genetic algorithm approach, and the third column gives the optimal result obtained by using the modified design approach suggested by constructal theory (see Fig. 5).

Case study #1: methanol-brackish water heat exchanger. This case study was taken from [45]. The original design assumed a heat exchanger with two tube-side passes (triangular arrangement) and one shell-side pass. The same architecture was used in the GA approach. Via the reduction of tube length and the increase of tube number, the GA approach decreased the total cost by 9.4% as compared with the original design method. Through the modified design approach, both the heat transfer area and total pressure drop were reduced, leading to a reduction of the total cost by 13.1% in comparison to the GA approach.

Case study #2: kerosene-crude oil heat exchanger. This case study was taken from [47]. The original design assumed a heat exchanger with four tube-side passes (square arrangement) and one shell-side pass. The same architecture was adopted in the GA approach. In this case, a reduction of total cost by 12.9% was observed using the GA approach resulting from the tube number increase and tube length decrease. For the modified approach, the total cost decreased by 24.3% compared to the original design.

Case study #3: distilled water-raw water heat exchanger. This case study was also taken from [47]. The original design assumed a heat exchanger with two tube-side passes (triangular arrangement) and one shell-side pass. The same architecture was applied in the GA approach. In this case, a remarkable reduction in the total cost was achieved using the GA approach (by 52.9%) and the modified approach (by 53%) as compared to the original design. This significant reduction was mainly caused by the significant increase of baffle spacing and tube number, and by the decrease of tube length. Consequently, a very high reduction of pressure drops for both shell- and tube-side flows allows about a reduction of about 91% in the operating cost. Overall, the total cost using the modified approach is slightly small than that using the GA approach.

From the analyses of the above three cases, it is reasonably concluded that the modified design approach demonstrates an advantage in optimization (i.e., the total cost minimization in this case), as compared to the original design and the GA design approach when discrete values are adopted. For the optimization of heat exchanger design in this work, there are two tube arrangements, four tube pass numbers, ten tube lengths, seven outer tube diameters and corresponding tube wall thicknesses to choose from, which consists of 1360 combinations for the GA design approach. If the modified approach is utilized, the combination adds up to 1,849,600 for two sections, and up to 2,515,456,000 for three sections. For both design approaches, the optimal parameters cannot exactly equal to the best operational parameters because most likely the best parameters, which correspond to the global minimum total cost, do not meet the requirements of TEMA. However, because the modified approach contains more solutions, the possibility of approaching the optimal is better than for the GA approach. Obviously, the combination of operational parameters on a shell-and-tube heat exchanger will increase if the in-series heat exchangers number increases, but it should be noticed that a larger

**Table 7**
Case study #3: Optimal parameters of using three design approaches with discrete values.

| | Case study #3 | | | |
|---|---|---|---|---|
| | Original values | GA values | New values | |
| | | | 1st Part | 2nd Part |
| Pitch pattern | Triangular | Triangular | Triangular | Triangular |
| Tube passes | 2 | 2 | 1 | 1 |
| Shell passes | 1 | 1 | 1 | 1 |
| $D_s$(m) | 0.387 | 0.5368 | 0.4576 | 0.3203 |
| $L$(m) | 5.904 | 2.438 | 2.438 | 2.438 |
| $B$(m) | 0.305 | 0.580 | 0.807 | 0.817 |
| $d_o$(m) | 0.019 | 0.01588 | 0.01588 | 0.01905 |
| $P_t$(m) | 0.023 | 0.01985 | 0.01985 | 0.0238125 |
| $t$(m) | 0.0019 | 0.001651 | 0.001651 | 0.001651 |
| $Cl$(m) | 0.004 | 0.00397 | 0.00397 | 0.0047625 |
| $N_t$ | 160 | 590 | 427 | 135 |
| $v_t$(m/s) | 2.436 | 0.9651 | 0.6665 | 1.349 |
| $Re_t$ | 40207 | 13181 | 9103 | 23066 |
| $Pr_t$ | 6.2 | 6.2 | 6.2 | 6.2 |
| $h_t$(W/m$^2$ K) | 9799 | 4852 | 3608 | 6063 |
| $f_t$ | 0.005519 | 0.006899 | 0.007429 | 0.006168 |
| $\Delta P_t$(Pa) | 65657 | 7303 | 1832 | 5740 |
| $a_s$(m$^2$) | 0.0217 | 0.06227 | 0.07386 | 0.05233 |
| $D_e$(m) | 0.01349 | 0.01128 | 0.01128 | 0.01353 |
| $v_s$(m/s) | 1.022 | 0.3562 | 0.3003 | 0.4239 |
| $Re_s$ | 17,155 | 4995 | 4211 | 7131 |
| $Pr_s$ | 5.4 | 5.4 | 5.4 | 5.4 |
| $h_s$(W/m$^2$ K) | 6186 | 3755 | 3418 | 3807 |
| $f_s$ | 0.3336 | 0.4014 | 0.4118 | 0.3805 |
| $\Delta P_s$(Pa) | 88,520 | 5071 | 2264 | 2401 |
| $U$(W/m$^2$ K) | 1230 | 966.6 | 869.2 | 1043 |
| $A$(m$^2$) | 56.35 | 71.71 | 51.87 | 19.61 |
| $C_i$(€) | 18,162 | 20,653 | 20,617 | |
| $C_o$(€) | 5141 | 444.7 | 445.3 | |
| $C_{oD}$(€) | 31,589 | 2733 | 2736 | |
| $C_{total}$(€) | 49,751 | 23,386 | 23,353 | |

sub-units number would be accompanied by a larger local pressure drop on tube sheet resulting in a larger operational cost. Most important, the modified design approach has potential applicability in many other engineering fields involving discrete values, such as plate heat exchangers and louver fin heat exchangers.

## 6. Conclusion

In this paper a modified design approach is proposed for shell-and-tube heat exchanger optimization. The novel design adopts the perception that divides a whole shell-and-tube heat exchanger into several in-series shell-and-tube heat exchangers, and then optimizes and designs sub-HEs simultaneously. A genetic algorithm is used to optimize the in-series heat exchangers and the total cost minimization of the shell-and-tube exchangers is set as the objective function. The main conclusions are as follows:

1. A modified optimization design approach motivated by constructal theory is proposed. It is successfully applied for optimization design procedure of shell-and-tube heat exchanger and demonstrates potential application in other engineering fields.
2. The modified approach suggested by constructal theory fails to prove the advantageous over the traditional genetic algorithm design approach when the design parameters take continuous values after a comparison of original values, traditional genetic algorithm values and constructal-based values.
3. Three cases studies with the TEMA standards imposing a discrete parameter space were considered to compare the results obtained by original values, the conventional values, and the new values. The results demonstrated that the novel design approach could reduce total cost in comparison with the other two methods when discrete values were used.

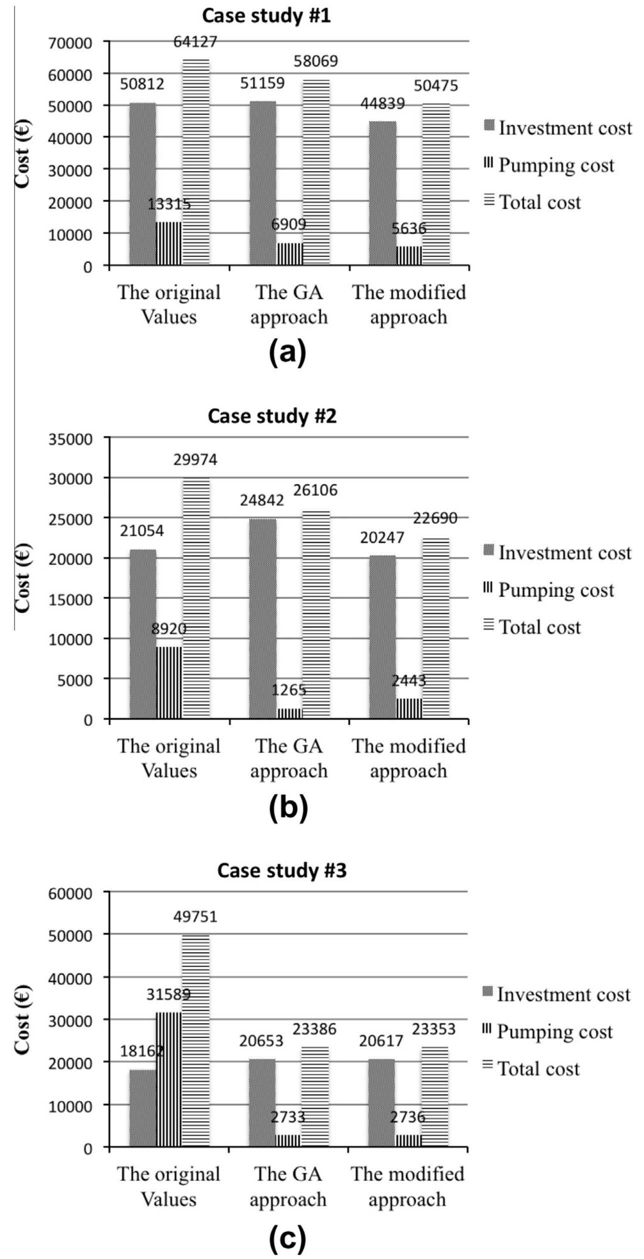


**Fig. 5.** Comparisons between the investment cost, pumping cost and total cost of the original values [45,47], the GA approach, and the modified approach: (a) the first case study, (b) the second case study and (c) the third case study.

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