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Optimal Design of Shell-and-Tube Heat Exchanger Based on Particle Swarm Optimization Technique

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Abstract
The paper studies optimization of shell-and-tube heat exchangers using the particle swarm optimization technique. A total cost function is formulated based on initial and annual operating costs of the heat exchangers. Six variables – shell inside diameter, tube diameter, baffle spacing, baffle cut, number of tube passes and tube layouts (triangular or square) – are considered as the design parameters. The particle swarm optimization selects the parameters so that the system has minimum total cost. Although generalization is not possible for any case, for minimization of cost functions of the three different cases studied in this research, larger tube outer diameter, triangular layout, baffle cut equalling 0.25 of shell diameter and one pass for each tube result in optimum designs. The other two parameters show no fixed trend.

Keywords: *heat exchanger, shell and tube, particle swarm optimization.*

1. Introduction
Heat exchangers, in general, are devices containing two streams with different temperatures while the heat is transferred between them [1]. Although there are various kinds of these devices with different applications, the shell-and-tube heat exchanger has wide uses in industries, especially in chemical processing, refineries, air-conditioning systems and power plants. This type of heat exchanger possesses advantages such as high pressure endurance and easy maintenance. Due to its advantages, the shell-and-tube heat exchanger has found extensive applications in comparison to other types. Thus, its optimal design can help save greatly on costs and energy.

The problem to be investigated in this paper relates to one of the fundamental factors affecting the applicability of shell-and-tube heat exchangers – optimization of the cost function depending on discrete variables, explicitly and implicitly. As seen with other functions studied in the literature, using analytic methods for optimization is a complicated process, and may even be impossible in some cases. Researchers have proposed many objective functions for shell-and-tube heat exchanger optimization, including total cost, pressure drop, exergy, entropy generation and heat transfer area [1–9]. Heuristic and meta-heuristic techniques have also been proposed. For example, Selbas et al. presented a total cost function for shell-and-tube heat exchangers which was a function of certain variables, and then applied genetic

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algorithm (GA) for its optimization [1]. In another study, Muralikrishana and Shenoy determined a feasible domain for design of heat exchangers by plotting geometrical and operational constraint curves on the pressure drop diagram, and designed the shell-and-tube heat exchangers optimally by trial and error [2]. Kara and Guraras developed a computer program calculating the heat transfer area by trying out all possible configurations for the design [3]. Serna and Jimenz related pressure drop to the tube and shell heat transfer rates using an analytical method [4]. Meanwhile, Eryner optimized exact baffle spacing by using a thermo-economic analysis method and compared the results with those yielded by the classical design method [5]. Ozcelik considered the total cost of the heat exchanger as the sum of the initial and exergy costs, then used GA for its optimization and obtained optimum design parameters [6]. Babu and Munawar used a differential evolution method to optimize the shell-and-tube heat exchangers using an improved version of GA [7]. They concluded that the proposed method was faster than the traditional GA. In another study, Costa and Queiroz reported an algorithm to optimize the heat transfer area of a shell-and-tube heat exchanger [8]. This algorithm eliminated designs which were not suitable for application and reduced algorithm repetition by searching along the tube count table and considering all of the design constraints. Guo et al. optimized a shell-and-tube heat exchanger using GA and computed optimum quantities for design variables by considering entropy generation as an objective function [9].

The particle swarm optimization (PSO) algorithm is a powerful optimization approach [10] and is applied in this paper. Here, a cost function is formulated for the shell-and-tube heat exchangers including their initial and operating costs. The operating cost includes power-loss costs of the shell and tube sides. PSO is applied to optimize the presented function. Some of the obtained results via application of PSO are compared with those computed using GA to verify the methods and results.

The proposed cost function will be a function of six variables: shell inside diameter, tube diameter, baffle spacing, baffle cut, tube pass and tube layout. Tube diameter is selected from the standard table presented by BWG [11]. In addition to a more comprehensive analysis in comparison to similar research, this paper presents the application of PSO for design optimization of the shell-and-tube heat exchangers to reach the minimum possible cost. Obviously, it uses the advantages of PSO such as extra operations like crossover being unnecessary (unlike GA) and having fewer parameters to adjust. In addition, convergence of PSO to the optimum region of the cost functions, with many variables, is better and faster than for other algorithms such as GA and evolutionary algorithms. Lastly, constraining the variables in PSO is easier.

2. Mathematical modelling

To design the shell-and-tube heat exchanger, its duty should first be defined by determining some constant values for mass flow rates of the tube and shell sides as well as their inlet and outlet temperatures. Then, one can select appropriate quantities for design variables and calculate the heat transfer area. By calculating the pressure drop in the shell and tube sides and estimating the power losses, the operating cost of the heat exchanger can be evaluated. The PSO algorithm is used to calculate the total cost of the heat exchanger and to select optimum values for design parameters.

To calculate the heat transfer area, a logarithmic mean temperature difference (LMTD) method is applied, as presented in Equation (1):

$$A = \frac{Q}{K\ F\ LMTD} \tag{1}$$

where A, Q, K and F are the heat transfer area, heat transfer rate, total heat transfer coefficient and dimensionless correction factor considered for correction of the flow configuration, respectively.

The heat transfer rate, Q, can be calculated by balancing heat transfer rates between the fluid flows as follows:

$$Q = \dot{m}_h C_{p,h}\left(T_{hi} - T_{ho}\right) = \dot{m}_c C_{p,c}\left(T_{co} - T_{ci}\right) \tag{2}$$

where $\dot{m}$, C_p, T_i and T_o denote the mass flow rate, specific heat capacity in constant pressure, inlet temperature and outlet temperature, respectively. The subscripts "*h*" and "*c*" stand respectively for the hot and cold flows.

The correction factor, F, is a function of temperature efficiency (P), ratio of heat capacity of the hot flow to cold flow (R), and flow configuration. The above parameters for single and multi-pass configurations of the shell-and-tube heat exchangers are presented in Equation (3) [1].

$$F = \frac{\sqrt{R^2+1}}{R-1} \frac{ln\, ln\left(\frac{1-P}{1-PR}\right)}{ln\, ln\left[\frac{2-P\left(R+1-\sqrt{R^2+1}\right)}{2-P\left(R+1+\sqrt{R^2+1}\right)}\right]} \tag{3}$$

where

$$R = \left(T_{h_i} - T_{h_o}\right) \Big/ \left(T_{c_o} - T_{c_i}\right) \tag{4}$$

$$P = T_{c_o} - T_{c_i} \Big/ T_{h_i} - T_{c_o} \tag{5}$$

The LMTD is obtained from Equation (6):

$$LMTD = \frac{\left(T_{h_i} - T_{c_o}\right) - \left(T_{h_o} - T_{c_i}\right)}{ln\, ln\left(\frac{T_{h_i} - T_{c_o}}{T_{h_o} - T_{c_i}}\right)} \tag{6}$$

The total heat transfer coefficient, K, depends on some parameters of both shell and tube. Thus, the design of shell-and-tube heat exchangers includes two separate sections. The first is for the design of the tubes and the next is for the design of the shells. In this paper, the former is conducted first.

2.1. Tube side design

The flow velocity in the tube can be calculated from the following Equation:

$$V_t = \frac{\frac{\dot{m}}{\pi\left(d_i^2/4\right)\rho}}{\frac{N_T}{s}} \tag{7}$$

where V_t, d_i, ρ , N_T and s represent the flow velocity through the tube, tube inner diameter, density, number of tubes and number of tube passes, respectively. The subscript "t" denotes the tube parameters. The number of tubes is obtained from Equation (8):

$$N_T = C\left(\frac{D_G - 0.02}{d_o}\right)^n \tag{8}$$

where D_G and d_o are shell diameter and outer diameter of the tube. C and n are two constants obtained from Table 1.

The distance between the two adjacent tube centres, St, is illustrated in Figure 1 for two common tube layouts (triangle and square):

Table 1. Values of C and n based on number of tube passes [1]

Number of tube passes	1	2	4	6	8
Triangular pitch St=1.25 d_o					
C	0.319	0.249	0.175	0.0743	0.0365
n	2.142	2.207	2.285	2.499	2.675
square pitch St=1.25 d_o					
C	0.215	0.156	0.158	0.0402	0.0331
n	2.207	2.291	2.263	2.617	2.643

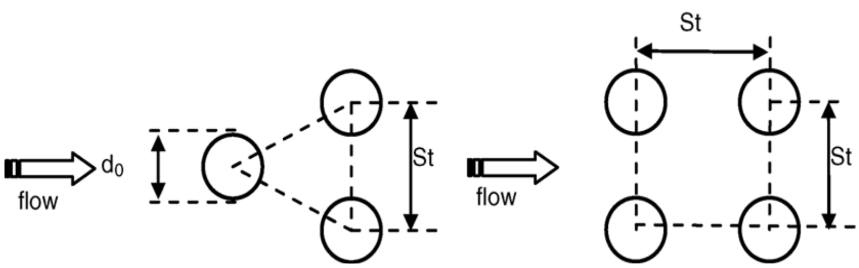


Fig. 1. Two common layouts of tubes in shell

Reynolds and Prandtl numbers to be utilized in the following equations are presented in Equations (9) and (10):

$$Re = \frac{\mathrm{V d_i}}{\nu} \tag{9}$$

$$Pr = \frac{\mu \mathrm{C_p}}{\mathrm{k}} \tag{10}$$

where ν, μ and k are the kinematic viscosity, dynamic viscosity and heat conductivity coefficient, respectively.

The convection heat transfer coefficient in the tube is obtained from Equations (11) to (13) based on the value of the Reynolds number [12, 13]:

$$h_t = \frac{\mathrm{k_t}}{\mathrm{d_i}}\left[3.657 + \frac{0.0677\left(Re_t Pr_t \frac{d_i}{L}\right)^{1.3}}{1 + 0.1 Pr_t \left(Re_t + \frac{di}{L}\right)^{0.3}}\right] \tag{11}$$

$Re_t < 2300$

$$h_t = \frac{k_t}{d_i}\left\{\frac{\frac{\lambda}{8}(Re_t - 1000)\, pr_t}{1 + 12.7\sqrt{\frac{\lambda}{8}}\left(pr_t^{0.67} - 1\right)}\left[1 + \left(\frac{d_i}{L}\right)^{0.67}\right]\right\} \tag{12}$$

$2300 < Re_t < 1000$

$$h_i = \frac{k_t}{d_i} 0.027\, Re_t^{0.8}\, Pr_t^{\frac{1}{3}}\left(\frac{\mu_t}{\mu_{w,t}}\right)^{0.14} \tag{13}$$

$R_{et} > 10000$

where $\mu_{w,t}$ is the dynamic viscosity of the fluid at the wall temperature and λ is the Darcy friction coefficient formulated as in [14].

$$\lambda = \left(1.82\, log_{10}\, log_{10}\, Re_t - 1.64\right)^2 \tag{14}$$

2.2. Shell side design

The shell hydraulic diameter, D_e, is presented in Equations (15) and (16) for square and triangular tube layouts, respectively [1].

$$D_e = \frac{1.27}{d_o}\left(S_t^2 - 0.785 d_o^{\,2}\right) \tag{15}$$

where S_t is the distance between two adjacent tubes equalling $S_t = 1.25 d_o$.

$$D_e = \frac{1.10}{d_o}\left(St^2 - 0.197 d_o^{\,2}\right) \tag{16}$$

The cross-section normal to the flow direction, A_s, can be computed from Equation (17). The subscript "*s*" represents the shell parameters.

$$A_s = \frac{(S_t - d_o) e D_G}{S_t} \tag{17}$$

where e and D_G denote baffle space and shell diameter, respectively.

The flow velocity in the shell side, V_s, and the related Reynolds number, Re_s, are presented in Equations (18) and (19).

$$V_s = \frac{\dot{m}_s}{\rho_s A_s} \tag{18}$$

$$Re_s = \frac{V_s D_e}{\nu_s} \tag{19}$$

The convection coefficient in the shell side can be estimated from Equation (20) [15].

$$h_s = \frac{k_s}{D_e}\, j\, Re_s\, Pr_s^{\frac{1}{3}}\left(\frac{\mu_s}{\mu_{w,s}}\right)^{0.14} \tag{20}$$

where j is a non-dimensional constant obtained from the Kern diagram presented in Figure 2.

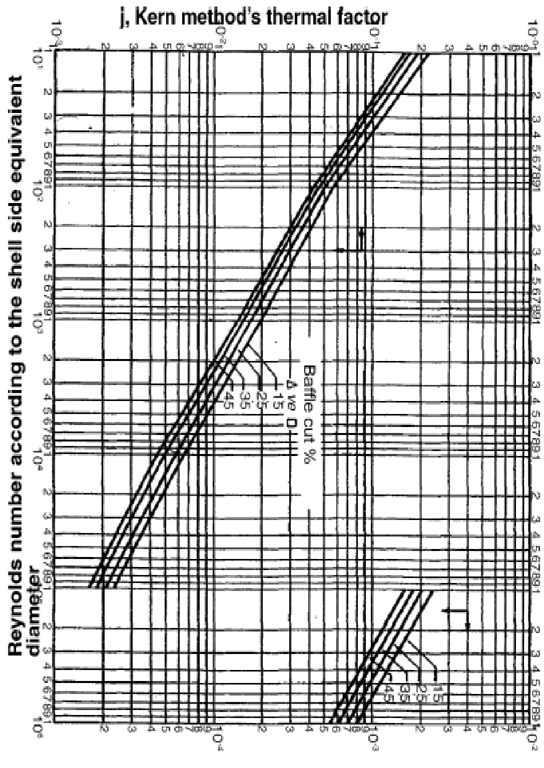


Fig. 2. Non-dimensional thermal constant *j* based on Kern method [13]

Hence, the total heat transfer coefficient can be determined using Equation (21).

$$\frac{1}{K} = \frac{d_o}{h_t d_i} + R_{f_i}\frac{d_o}{d_i} + \frac{X_w}{K_w} + R_{f_o} + \frac{1}{h_s} \tag{21}$$

where R_{f_i}, X_w, K_w and R_{f_o} are the thermal fouling resistance in the tube, thickness of the tube, thermal conductivity of the tube and thermal fouling resistance out of the tube, respectively.

Knowing the heat transfer area from Equation (1), the required length for the heat exchanger can be calculated from the following equation:

$$L = \frac{A}{\pi d_o N_t} \tag{22}$$

2.3. Effect of pressure loss

In all heat exchangers, there are strict physical and economic relations between heat transfer and pressure loss. For a certain heat capacity, increased flow velocity leads to increased heat transfer coefficient and pressure loss. The former is desirable, whereas the latter is not. The reason is that the pressure loss should be compensated by more powerful pumps, requiring additional cost. Thus, in design of the heat exchangers, both the heat transfer and the pressure loss should be considered.

The pressure loss in the tube side is formulated as below [1].

$$\Delta P_t = s\left[\lambda \frac{L}{d_i}\left(\frac{\mu_t}{\mu_{w,t}}\right)^{-0.14} + 2.5\right]\frac{\rho V_t^{\,2}}{2} \tag{23}$$

Meanwhile, the pressure loss in the shell side is obtained from Equation (24).

$$\Delta P_s = 8 j_{f,k} \frac{D_G}{D_e}\frac{L}{e}\frac{\rho_s V_s^{\,2}}{2}\left(\frac{\mu_s}{\mu_{w,s}}\right)^{-0.14} \tag{24}$$

where $j_{f,k}$ is the non-dimensional pressure constant based on the Kern method. The total pressure loss may be calculated from Equation (25).

$$\Delta P = \frac{1}{\eta}\left(\frac{\dot{m}_t}{\rho_t}\Delta P_t + \frac{\dot{m}_s}{\rho_s}\Delta P_s\right) \tag{25}$$

where η is the pump efficiency.

The goal of this research is to optimize the shell-and-tube heat exchanger from an economic point of view, maximizing the heat transfer while minimizing the costs. For this purpose, a total cost function, C_{tot}, including two initial, C_{in}, and operational, C_{op}, costs is considered [16].

$$C_{tot} = C_{in} + C_{op} \tag{26}$$

The initial and operational costs are formulated in Equations. (27) to (29):

$$C_{in} = a_1 + a_2 A^{a_3} \tag{27}$$

where α_1, α_2 and α_3 are the constants;

$$C_{\text{op}} = \sum_{k=1}^{n_y} \frac{C_0}{(1+i)^k} \tag{28}$$

$$C_0 = \Delta P C_E H \tag{29}$$

where n_y, i, C_E and H denote number of years, interest rate, cost of 1 kW energy and number of working hours per year, respectively.

3. Particle Swarm Optimization

In this algorithm, a number of particles are considered randomly. The position of each particle in n-dimensional space is indicated by an n-dimensional vector, where n equals the number of variables of the cost function. The particles move in the space with speed v. The positions and velocities of the particles are updated in each iteration. The criterion of updating the velocity is distance of each particle to local minimums and global minimum, computed by Equation (30a). The position of the particles can be calculated from Eqution (30b) [17].

$$\begin{aligned} v_{m,n}^{new} &= v_{m,n}^{old} + \Gamma_1 \times r_1 \times (p_{m,n}^{local\ best} - p_{m,n}^{old}) + \\ &\quad \Gamma_2 \times r_2 \times (p_{m,n}^{global\ best} - p_{m,n}^{old}), \end{aligned} \tag{30a}$$

$$p_{m,n}^{new} = p_{m,n}^{old} + v_{m,n}^{new}, \tag{30b}$$

where

$v_{m,n}$ = particle velocity,

$p_{m,n}$ = particle position,

r_1, r_2 = independent uniform random numbers,

$\Gamma_1 = \Gamma_2$ = learning factors = 2,

$p_{m,n}^{\text{localbest}}$ = best local solution,

$p_{m,n}^{\text{globalbest}}$ = best global solution.

In some respects, PSO is similar to continuous GA, but the former has some advantages [18-20]. As mentioned above, this algorithm does not require extra operations such as crossover and has fewer parameters to adjust. In addition, convergence of PSO to the optimum region of the cost functions, with many variables, is better and faster than with GA, and constraining the variables in PSO is easier. For these reasons, PSO was chosen to optimize the total cost of the shell-and-tube heat exchangers.

4. Results and discussion

In this paper, the optimal design of the heat exchanger uses six design variables, shell inside diameter (D_s), tube diameter (d), baffle spacing (B), baffle cut (B_c), number of tube passes (N_p) and tube layout (TL), in order to reach the minimum total cost for the heat exchanger. The considered ranges for the design variables are applied as in Table 2.

In addition to the constraints presented above, the pressure drop and tube and shell flow velocities should be in the limited ranges. Table 3 presents the constant values considered to conduct the research.

One of the most important factors in economic analysis is the interest rate. This is simply the rate at which interest is paid by a borrower (debtor) for the use of money that they borrow from a lender (creditor). For example, a small company borrows capital from a bank (or an investor) to provide new assets for its business, and in return the lender receives interest at a predetermined interest rate for deferring the use of the funds. Interest rates are normally expressed as a percentage of the principal for a period of one year. This is considered here to be 0.1 but higher interest rates generally increase the cost of borrowing, which can reduce investment and output and increase unemployment.

The following three cases, selected from [21], are taken as the case studies for optimization. Table 4 shows the design specifications and thermo-physical properties for each case study at $\frac{T_i + T_o}{2}$.

Table 2. Ranges of design parameters

Design parameter	Range of variation	Number of cases
Outer tube diameter	10–28 mm	10
Tube layout	Square & Triangle	2
Number of passes	1, 2, 4, 8	4
Shell diameter	150–530 mm	20
Baffle space	0.2–0.4 of shell diameter	5
Baffle cut	0.15, 0.25, 0.35, 0.45 of shell diameter	4

Table 3. Numerical constant considered for optimization [21]

Parameter	Numerical value
Tube thickness (mm)	2
Pump efficiency	0.7
Interest rate	0.1
Operational time (yr)	10
Operational hours per year (h/yr)	700
Numerical constant (€), a_1	8000
Numerical constant (€/m^2), a_2	259.2
Numerical constant, a_3	0.91
Energy cost (€/kW h), C_E	0.12
Population size	20

Table 4. Case studies for the optimization

Case specification	Mass flow rate ($\frac{kg}{s}$)	T_{in} ($^{\circ}C$)	T_{out} ($^{\circ}C$)	$\rho(\frac{kg}{m^3})$	$c_p(\frac{kJ}{kg\ K})$	μ(Pa s)	$k(\frac{W}{m\ K})$	$R_{fouling}(\frac{m^2K}{W})$
				Case 1				
Shell side: Methanol	27.8	95	40	750	2.84	0.00034	0.19	0.00033
Tube side: Sea water	68.9	25	40	955	4.2	0.0008	0.59	0.0002
				Case 2				
Shell side: kerosene	5.52	199	93.3	850	2.47	0.0004	0.13	0.00061
Tube side: Crude oil	18.8	37.8	76.7	955	2.05	0.00358	0.13	0.00061
				Case 3				
Shell side: distilled water	22.07	33.9	29.4	955	4.18	0.0008	0.62	0.00017
Tube side: raw water	35.31	23.9	26.7	999	4.18	0.00092	0.62	0.00017

Figures 3 (a) and (b) show the cost functions obtained from GA and PSO. The former is not related to the goal of this study and is used only to demonstrate the effectiveness of the latter. GA results in the optimum total cost value of 35,615.86 in iteration 16, but this value calculated with PSO is 35,500.7 in iteration 7. The values reveal that PSO dominates GA in both cost values and iteration numbers.

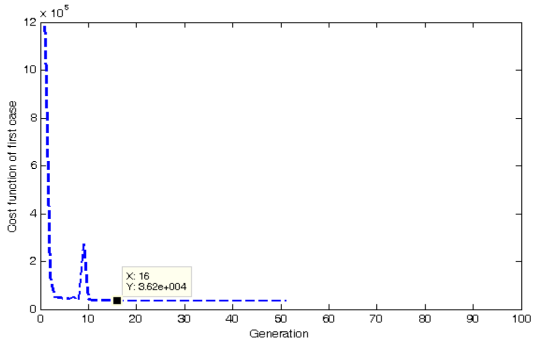
In order to obtain the best results, the developed codes are executed 50 times. The results presented in Figures 4 (a) and (b) prove that once again PSO has better performance that GA. As depicted in the figures, both the minimum and average cost values calculated using PSO are smaller than the values computed via GA. The following relation is considered as a dispersal index:

$$\gamma = \frac{\sum_{i=1}^{n}\left(C_{\text{tot,i}} - C_{\text{tot,min}}\right)}{n} \tag{31}$$

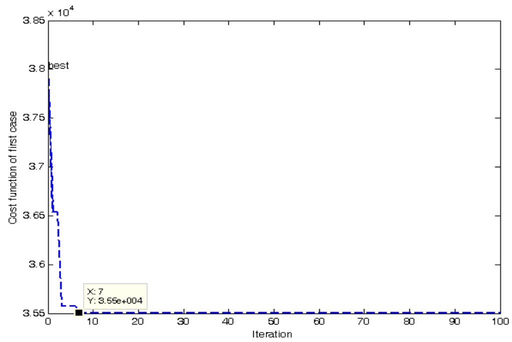
This index is $\gamma_{PSO} = 1570€$ for PSO and $\gamma_{GA} = 1939€$ for GA. The comparison between the two latter values is another piece of evidence for the effectiveness of PSO over GA.

The aforementioned figures and explanations related to case 1, shown in Table 4. The numerical values of the cost functions for the second and third cases vs. iteration computed using PSO are plotted in Figures 5 (a) and (b). The figures also reveal the capability of the applied optimization technique for the two other cases.

Table 5 presents the optimum values of the design parameters and other values formulated in this paper for the three case studies considered in Table 4. Using the obtained optimum design parameters, one can minimize the cost of the shell-and-tube heat exchangers.

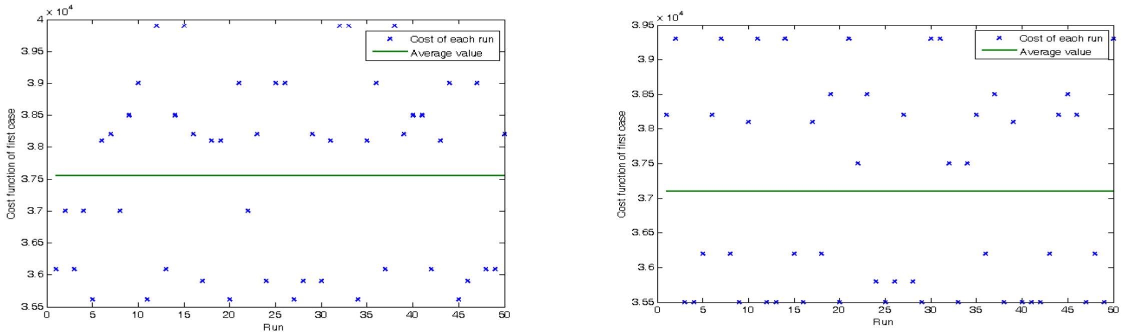


(a)



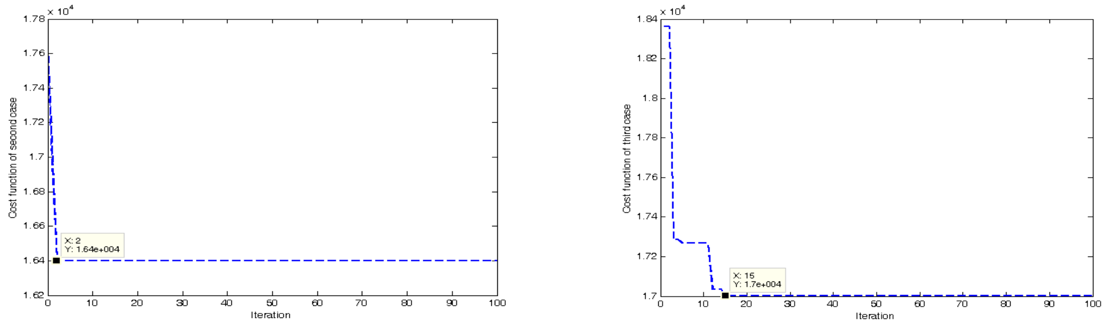
(b)

Fig. 3. Total cost functions vs. iteration obtained from (a) GA, (b) PSO



(a) (b)

Fig. 4. Comparison between the results obtained from (a) GA, (b) PSO in 50 runs



(a) (b)

Fig. 5. Results of total cost values vs. iteration of the (a) second and (b) third cost functions

Table 5. Optimum parameters and design values of the shell-and-tube heat exchanger

	Case 1	Case 2	Case 3
Parameters	Optimum value	Optimum value	Optimum value
Shell diameter, $\mathbf{D_s}$ (m)	0.33	0.15	0.27
Baffle space (m)	0.132	0.06	0.108
Tube outer diameter, d_o (m)	0.028	0.028	0.028
Tube layout	Triangle	Triangle	Triangle
Baffle space	25%	25%	25%
Number of tube passes	1	1	1
Number of tubes, N_t	55	9	35
Flow velocity in tube , v_t (m/s)	2.9	5.09	2.25
Reynolds number in tube, Re_t	8.3e4	3.26e4	5.93e4
Prandtl number in tube , Pr_t	5.7	56.45	6.14
Convection coefficient in tube, h_t (W/m^2 K)	11847	2764	9629
Pressure loss in tube, ΔP_t (kPa)	19.41	66.68	12.7
Shell cross section, A_s (m^2)	0.087	0.0018	0.0058
Shell equal diameter, De (m)	0.0199	0.019	0.0276
Flow velocity in shell, V_s (m/s)	4.255	3.61	3.963
Reynolds number in shell, Re_s	1.87e5	1.52e5	1.31e5
Prandtl number un shell, Pr_s	5.1	7.6	5.39
Convection coefficient in shell, h_s (W/m^2 K)	5990	4207	11859
Pressure loss in shell, ΔP_s (kPa)	51.22	41.74	40.68
Total heat transfer coefficient, K (w/m^2 K)	1142	493.1	1612
Heat transfer area, A (m^2)	152	38.78	43.19
Total pressure loss (kPa)	4.71	2.262	1.98
Initial cost, C_{in} (€)	33068	15232	15977
Operational cost, C_{op} (€)	2432	1168	1024
Total cost, C_{tot} (€)	35501	16400	17001

5. Conclusion

This paper has investigated the problem of optimization of shell-and-tube heat exchangers for three different case studies. The PSO method was applied to minimize the total cost. The shell inside diameter, tube diameter, baffle spacing, baffle cut, number of tube passes and tube layouts (triangular or square) were taken as the design parameters, chosen by the developed code to achieve the minimum cost. The results obtained from the PSO algorithm were partly compared with those calculated using GA; the outcomes revealed better performance of PSO in design optimization of the heat exchangers.

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