Optimization of shell and tube heat exchanger using the water cycle algorithm

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Abstract. The shell and tube heat exchangers (STHEs) are used to transfer the thermal energy between two or more media. Due to its importance for conserving energy, numerous studies of heat exchanger design optimization have been carried out. The traditional design approaches are usually time-consuming, and may not guarantee to produce an economically optimal solution. A complete optimization procedure was conducted to find the minimum cost of the heat exchanger system. Heat exchanger design optimization was carried out using a recently proposed optimization algorithm called the water cycle algorithm (WCA). Three design variables are considered consisting of the shell inside diameter, tube outer diameter, and baffle spacing. The total annual costs which include the investment costs and the operating costs will be considered as the objective function.

Keyword: heat exchanger, thermal energy, water cycle algorithm, baffle spacing.

1. Introduction
A heat exchanger is a mechanical system used to transfer heat between two fluids stream at different temperatures, hot and cold streams, which are separated by a solid wall. The applications of heat exchangers are normally found in petrochemical industries as well as waste heat recovery, power generation, refrigeration, and air conditioning [1]. Among the different types of heat exchangers, shell and tube heat exchangers (STHEs) are the most widely used heat exchanger and have a wide range of operating pressures and temperatures. A typical STHE is shown in Figure 1.

![Shell and tube heat exchanger](image-url)
The design of STHE includes several geometric and operating variables to find a heat exchanger geometry that can satisfy the heat duty requirement with a particular set of design constraints. Normally, a geometric layout of the heat exchanger is chosen as a reference at first with a fixed allowable pressure drop value. Then, based on the design specifications and several assumptions of thermodynamic and mechanical parameters, the values of the design variable for the heat exchanger are defined [1, 2].

As the STHEs play an important role in conserving energy, several techniques of design optimization problem by considering different objective functions have been proposed such as mixed-integer nonlinear programming, genetic algorithm, simulated annealing, imperialist competitive algorithm, particle swarm optimization, and so on. Amongst the mentioned optimization technique, genetic algorithm (GA) received the most attention in the design optimization of STHEs for the past few years. Caputo et al. [3] proposed an optimization of heat exchanger using the GA from an economical perspective. They minimized the total cost of the heat exchanger which includes the total discounted operating expenditures and the capital investment. Three different case studies were carried out to validate the capability of the suggested method. A reduction in the total costs higher than 50% was obtained from the examined cases.

Guo et al. [4] conducted a study where they used GA along with entropy generation minimization to optimize the design of STHE. The modified entropy number in this case is the dimensionless entropy generation rate, set as the objective function. Multiple design variables were considered. The results showed a significant increase in heat exchanger effectiveness and also a reduction in pumping power. Selbas et al. [5] utilized GA by considering the use of a logarithmic mean temperature difference (LMTD) method which is used to identify the dimensions of the STHE characteristics. They included the maintenance cost and the capital recovery factor in their proposed cost estimation.

Patel and Rao [6] used particle swarm optimization (PSO) for the design optimization of STHE in terms of the economic point of view with the minimization of the total annual cost as its objective function. For the design variables, the outer tube diameter, shell internal tube diameter, and the baffle spacing was considered. The results of using PSO were then compared with the other four different case studies of using GA to show the effectiveness of the two methods.

A constructal theory, which was suggested by Bejan [7], has been used as an optimization design method for heat exchanger. For a system such as STHE, the constructal theory suggests that a design (geometry, flow pattern, configuration) with higher durability, stability, and conservation can be accomplished by increasing the access level of heat and working fluids that flow through it. Yang et al. [8, 9] performed an optimization design approach for STHE based on the constructal theory, with all the design parameters following the Tubular Exchanger Manufacturers Association (TEMA) standards. The objective function is the total cost of STHE by using genetic algorithm method.

Azad et al. [10] applied GA that is based on a constructal theory for the optimization design to reduce the total cost of the exchanger. It was shown that the overall heat transfer coefficient of the STHE increases by using the constructal theory and hence reducing the cost of manufacturing the heat transfer surface. Moreover, the constructal theory also helps to reduce the operational energy cost of the pump which was used to overcome the pressure loss. The results show a reduction of costs by more than 50%.

Another optimization technique which was applied for the cost minimization of STHE is based on the imperialist competitive algorithm (ICA) [11]. This algorithm is based on the human’s socio-political evolution that has been used in various engineering applications such as data clustering, thermal engineering, and industrial engineering. It has been shown that ICA has a better rate of convergence and also finding the global minima compared to the other evolutionary algorithm. Hadidi et al. [12] performed a design optimization using ICA by varying several important parameters of the STHE. The ICA method provides a faster solution to the design problem and able to explore many other different solutions of good quality. With the results showing a saving in operating cost as high as 94% and reduction of the total cost up to 53%, it proves that ICA method is more efficient than the traditional GA method.

Later, Hadidi et al. [13] also applied the biogeography-based optimization (BBO) algorithm to heat exchanger design optimization. BBO method was first proposed by Simon [14], a search algorithm based
on the geographical distribution of biological organisms. BBO algorithm mainly operates based on mutation and migration and has a similar way of sharing data between solutions similar to PSO and GA methods. The results produced by BBO method showed a slight improvement over the results given by the ICA with saving in operating cost as high as 96% and reduction of the total cost up to 56.1% [13].

Fettaka et al. [15] performed a multi-objective optimization, also called Pareto optimization, of the pumping power and heat transfer of STHE. The multi-objective optimization offers several Pareto-optimal solutions without degrading the values of the other objective function. In the research, they considered nine design variables which were the number of tube passes, tube layout pattern, baffle cut, baffle spacing, shell-to-baffle diametrical clearance, tube-to-baffle diametrical clearance, tube outer diameter, tube length, and tube wall thickness. The optimization was done via the fast and elitist non-dominated sorting genetic algorithm (NSGA-II). The results showed that multi-objective optimization offers a Pareto front with a larger extent of optimal design variables.

Turgut et al. [16] used an improved intelligent tuned harmony search (I-ITHS) algorithm, an improved form of the harmony search algorithm, for optimizing the total cost of STHE. Four design variables were considered such as the shell diameter, tube outer diameter, baffle spacing, and the number of tube passes. Results showed that I-ITHS algorithm can be applied to STHE design problem.

Asadi et al. [17] optimized the design of STHE using a cuckoo-search-algorithm (CSA). The objective function is the total annual cost of the heat exchanger. The obtained results by using CSA indicated the possibility of energy saving by 77%. A different kind of algorithm was used for the optimization of STHE by Mohanty [18] namely the firefly algorithm (FA) which was inspired by the nature of the flashing light of fireflies. Considering the total annual cost which includes the operating cost and investment cost as the objective function, a reduction in the operating cost by 77% and the total cost by 29% were obtained. Mohanty also performed a different algorithm called the gravitational search algorithm (GSA). The gravitational search algorithm is a heuristic type of search algorithm which is derived from the law of gravity and mass interactions. The GSA methodology was utilized to find the ideal geometric configuration by considering the complexity of the STHE design and its applications. The results obtained were then compared with several other types of the algorithm used in STHE [19]. In this study, heat exchanger design optimization was carried out by performing water cycle algorithm (WCA) simulations.

2. Shell and tube heat exchange model

2.1. Heat transfer coefficient

Depending on the system of the flow, the tube side heat transfer coefficient $h_t$ can be computed from the following correlation [2, 3]:

$$h_t = \frac{k_t}{d_t} \left[ 3.657 + \frac{0.0677(Re_t Pr_t \frac{d_t}{L})^{1.33}}{1 + 0.1 Pr_t (Re_t \frac{d_t}{L})^{0.3}} \right], \quad Re_t \leq 2300 \quad (1)$$

$$h_t = \frac{k_t}{d_t} \left[ \frac{0.127(Re_t - 1000) Pr_t}{1 + 12.7 \left( \frac{Pr_t}{Pr_t^*} \right)^{\frac{1}{2}}} \left( 1 + \frac{d_t}{L} \right)^{0.67} \right], \quad 2300 \leq Re_t \leq 10,000 \quad (2)$$

$$h_t = \frac{k_t}{d_t} \left[ 0.027 Re_t^{0.8} Pr_t^{\frac{1}{3}} \left( \frac{h_t}{\mu_{wt}} \right)^{0.14} \right], \quad Re_t \geq 10,000 \quad (3)$$
Nomenclature

\[ D_s \] shell inside diameter
\[ d_o \] tube outside diameter
\[ B \] baffles spacing
\[ N_t \] number of tubes
\[ S_t \] tube pitch
\[ v_t \] tube side fluid velocity
\[ Re_t \] tube side Reynolds number
\[ f_t \] tube side friction coefficient
\[ Pr_t \] tube side Prandtl number
\[ ν_s \] shell-side fluid velocity
\[ h_t \] tube side convective coefficient
\[ LMTD \] logarithmic mean temperature difference
\[ T_{hi} \] hot fluid inlet temperature
\[ T_{ci} \] cold fluid inlet temperature
\[ T_{ho} \] hot fluid outlet temperature
\[ T_{co} \] cold fluid outlet temperature

where \( k_t \) and \( f_t \) are the tube side thermal conductivity and the Darcy friction factor respectively. The tube inner diameter \( d_i \) is calculated as:
\[ d_i = 0.8d_o \] (4)
The Darcy friction factor can be calculated by [3, 20],
\[ f_t = (1.82 \log_{10} Re_t - 1.64)^{-2} \] (5)
The Prandtl number of the tube side \( Pr_t \) is given as:
\[ Pr_t = \frac{\nu_s c_p t}{k_t} \] (6)
The Reynolds number of the tube side \( Re_t \) is given as:
\[ Re_t = \frac{\rho_t V_t d_i}{\mu_t} \] (7)
The tube side flow velocity can be calculated using [21]:
\[ V_t = \frac{m_t}{\pi d_i^2} \frac{(n)}{\rho_t N_t} \] (8)
where \( m_t \) is the tube side mass flow rate, \( n \) is the number of tube passes and \( N_t \) is the number of tubes which can be determined from [1, 3]:
\[ N_t = K_1 \left( \frac{d_s}{d_o} \right)^{n_1} \] (9)

The \( D_s \) is the shell inside diameter, \( d_o \) is the tube outer diameter, \( K_1 \) and \( n_1 \) are the numerical coefficient depending on the number of passes and the flow arrangement. These coefficients are presented in Table 1.

| No. of passes | Triangular 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             |
The shell side heat transfer coefficient \( h_s \) is calculated using Kern’s formulation for segmental baffle STHE [21]:

\[
h_s = \frac{k_s}{D_e} \cdot 0.36Re_s^{0.55}Pr_s^{0.14} \left( \frac{\mu_s}{\mu_w} \right)^{0.14}
\]  

(10)

where \( D_e \) is the shell equivalent diameter which can be calculated by [2, 21]:

\[
D_e = 4\left[\frac{S_t^2 - (\pi d_o^2)}{\pi d_o}\right]; \quad \text{(for square pitch)}
\]  

(11)

\[
D_e = 4\left[0.43S_t^2 - 0.5\left(\frac{\pi d_o^2}{0.5\pi d_o}\right)\right]; \quad \text{(for triangular pitch)}
\]  

(12)

where \( S_t \) is the tube pitch as illustrated in Figure 2.

![Figure 2. Tube arrangements](image)

The Prandtl number for shell side \( Pr_s \) is given as:

\[
Pr_s = \frac{\mu c_p}{\lambda}
\]  

(13)

The Reynolds number for shell side \( Re_s \) is given as:

\[
Re_s = \frac{\rho_s V_s D_e}{\mu_s} = \frac{m_s D_e}{\rho_s A_s \mu_s}
\]  

(14)

The shell side flow velocity \( V_s \) can be calculated by taking into consideration the shell side cross-section area normal to flow direction \( A_s \):

\[
V_s = \frac{\mu_s}{\rho_s A_s} = \frac{m_s A_s}{D_e B C_l}
\]  

(15)

\[
A_s = \frac{S_t}{D_e B C_l}
\]  

(16)

where \( m_s \) is the shell side mass flow rate, \( B \) is the baffle spacing and \( C_l \) is the shell side clearance given as:

\[
C_l = S_t - d_o
\]  

(17)

After finding the heat transfer coefficient for both shell and tube side, the overall heat transfer coefficient \( U \) can be determined by considering the fouling resistance \( R_f \) on both sides [2]:

\[
U = \frac{1}{\left(\frac{1}{h_s} + R_f S_t + \frac{d_o}{A_s}\right)\left(R_f S_t + \frac{1}{R_f}ight)}
\]  

(18)

The fouling resistances are determined from the literature data based on the operating temperatures and the type of fluids. Considering the flow between the adjacent baffles as crossflow, the logarithmic mean temperature difference \( \text{LMTD} \) can be calculated by:

\[
\text{LMTD} = \frac{(T_{ho}-T_{co})-\left(T_{ho}-T_{ci}\right)}{\ln\left(T_{ho}-T_{ci}\right)\ln\left(T_{ho}-T_{co}\right)}
\]  

(19)
Based on the flow configuration involved, the temperature difference correction factor \( F \) is introduced where it is a function of dimensionless temperature ratio for most flow configuration of interest \([1, 2]\).

\[
F = \frac{(R^2 + 1)^{1/2}}{R - 1} \left[ \ln \left( \frac{1 - p}{1 - PR} \right) \right] \left[ \ln \left( \frac{2 - P R}{2 - P R + (R^2 + 1)^{1/2}} \right) \right] \] (20)

where \( R \) is the correction coefficient and \( P \) is the efficiency and can be found by the following equations:

\[
R = \frac{(T_{hi} - T_{ho})}{(T_{co} - T_{ci})} \] (21)

\[
P = \frac{(T_{co} - T_{ci})}{(T_{hi} - T_{ci})} \] (22)

Taking into consideration the overall heat transfer coefficient, \( LMTD \), and the correction factor, the heat exchanger surface area \( A \) is calculated by:

\[
A = \frac{Q}{U \cdot F \cdot LMTD} \] (23)

where \( Q \) is the heat transfer rate for sensible heat transfer expressed as:

\[
Q = m \cdot c_p \cdot (T_{hi} - T_{ho}) = m \cdot c_p \cdot (T_{co} - T_{ci}) \] (24)

The necessary tube length \( L \) can be determined based on the heat exchanger surface area:

\[
L = \frac{A}{\pi d_o N_t} \] (25)

2.2. Pressure drops
In many heat exchangers, heat transfer is closely related to the pressure drop in terms of economical and physical. The pressure-drop is the fluid pressure available to pump the fluid through the exchanger. For a heat exchanger that is going to be designed, under a constant heat capacity, increasing the flow velocity will also increase the heat transfer coefficient. In this matter, a compact with lower dimension heat exchanger can be made with lower investment cost.

However, high flow velocity results in the high-pressure drop which in return requires a more powerful pump and this mean additional running cost. Hence, the pressure drop must be taken into account along with the heat transfer when designing a heat exchanger to obtain the best solution for the system \([6, 21]\).

The tube side pressure drop is computed as the summation of the distributed pressure drop along the tubes length and the concentrated pressure losses in elbows and in the inlet and outlet nozzles \([21]\):

\[
\Delta P_t = \Delta P_{\text{tube,length}} + \Delta P_{\text{tube,elbow}} = \frac{p \cdot U \cdot L}{2} \left( \frac{L}{d_i} + p \right) n \] (26)

where \( p \) is a constant and has different values given in the literature. Kern \([21]\) assumed the value of \( p = 4 \) while Sinnott \([2]\) assumed the value of \( p = 2.5 \). Whereas the pressure-drop of the shell side can be calculated by:

\[
\Delta P_s = f_s \left( \frac{U \cdot L}{2} \right) \left( \frac{D_e}{d_o} \right)^{0.15} \] (27)

\[
f_s = 2 b_o R e_s^{-0.15} \] (28)

where \( f_s \) is the friction factor and \( b_o = 0.72 \) valid for \( R e_s < 40,000 \).

2.3. Costs estimations
The total annual cost \( C_{\text{tot}} \) is assessed as the objective function by considering the capital investment \( C_i \), the annual operating cost \( C_o \), the energy cost \( C_e \) and the total discounted operating cost \( C_{od} \) \([3]\):

\[
C_{\text{tot}} = C_i + C_o + C_e + C_{od} \] (29)
The capital investment is evaluated as a function of the heat exchanger surface area using Hall’s correlation [23]:

\[ C_i = a_1 + a_2 A + a_3 \] (30)

where \( a_1 = 8000 \), \( a_2 = 259.2 \) and \( a_3 = 0.93 \) for heat exchanger made of stainless steel for both shells and tubes.

The total discounted operating cost is influenced by the pumping power \( P \) to deal with the frictional losses and can be calculated as:

\[ C_{od} = \sum_{j=1}^{n_y} C_o \] (31)

\[ C_o = P C_e H \] (32)

\[ P = \frac{1}{\eta} \left( \frac{m_t}{\rho_t} \Delta P_t + \frac{m_s}{\rho_s} \Delta P_s \right) \] (33)

where the pumping efficiency \( \eta \) is taken as 80%.

From all of the above calculations, the total cost of the heat exchanger can be determined from Eq. (29).

3. Water cycle algorithm (WCA)

The WCA is a population-based metatheuristic algorithm based on the observation of the water cycle and how rivers and streams flow downhill towards the sea in the real world [24, 25, 26]. Rivers and streams are formed whenever water moves downhill from one location to another. Most rivers are formed at higher altitudes where snow or glaciers melt. Streams and rivers eventually end up to a sea. Water is also allowed to evaporate into the atmosphere to generate clouds which then condenses in the colder atmosphere, releasing the water back to the earth in the form of rain or precipitation.

The initial population generated randomly is called raindrops. Hence, a raindrop represents an individual or single solution comprising a number of design/decision variables. The best individual (best raindrop) is chosen as a sea to represent an optimal solution to a given problem. Further details of the WCA are given in [24, 25, 26].

4. Results and discussions

4.1. Case study

The case study was investigated in [3, 6, 18, 19] which was originally taken from [21]. The considered heat exchanger is having a heat duty of 1.44 MW. According to the original specification, the fluid flowing through the shell side is hot kerosene while the fluid flowing through the tube side is cold crude oil. The heat exchanger is a single shell pass and four tube passes exchanger with a square pitch arrangement.

The physical properties and the operating parameters for both shell and tube side fluids are summarized in Table 2.

| Process parameters | Case study |
|--------------------|------------|
|                    | Shell side: kerosene | Tube side: crude oil |
| Mass flow rate, \( m \) (kg/s) | 5.52 | 18.80 |
| Inlet temperature, \( T_i \) (°C) | 199.0 | 37.8 |
| Outlet temperature, \( T_o \) (°C) | 93.3 | 76.7 |
| Specific heat, \( c_p \) (m²K/W) | 2.47 | 2.05 |
| Density, \( \rho \) (kg/m³) | 850.0 | 995.0 |
| Viscosity at tube wall, \( \mu \) (Pa.s) | 0.0004 | 0.00358 |
| Viscosity at core flow, \( \mu_w \) (Pa.s) | 0.00036 | 0.00213 |
| Thermal conductivity, \( k \) (W/m.K) | 0.13 | 0.13 |
| Fouling resistance, \( R_f \) (m²K/W) | 0.00061 | 0.00061 |
The upper and lower bounds for the optimization design variables used in this work are similar to the ones considered in [3]. The shell internal diameter $D_s$ ranging between 0.1 m to 1.5 m, tube outer diameter $d_o$ ranging from 0.01 m to 0.051 m and baffle spacing $B$ ranging from 0.05 m to 0.5 m were considered. Likewise, all the values used for discounted operating costs calculation are energy cost $C_e = 0.12\,\text{€/kW\,h}$, $ny = 10$ years, annual discount rate $j = 10\%$ and an annual work hours $H = 7000\,\text{h/year}$, similar with the other researchers.

4.2. Discussions
The simulations for WCA were run 20 times by setting the number of populations as 20 and the number of maximum iterations as 20. This is because the objective function was found to be converged to its minimum value less than 12 generations as shown in Figure 3.

![Figure 3. Convergence of WCA](image)

The results obtained were compared with the original design reported in [21] and other optimization methods reported in the literature [3, 6, 18, 19]. The outcomes are presented in Table 3.

The results show WCA offers a reduction in the tube length compared to the original design and GA while the tube length given by WCA is longer than the lengths given by PSO, FA, and GSA. However, the number of tubes given by PSO, FA, and GSA is more than the number given by WCA. The heat transfer coefficients for both shell and tube sides given by WCA were found to be significantly lower than the ones given by other methods. Thus, a considerably lower overall transfer coefficient was observed. The heat exchanger area given by WCA is lowered by 29.2%, 10%, 15.9%, and 13.4% compared to the original design, GA, FA, and GSA respectively. Consequently, the investment cost given by WCA is lower than the other optimization methods except for PSO.

The tube side velocity causes the pressure drop on the tube side to be significantly higher than the values obtained from other optimization methods. The increase in the tube length also contributes to the increase in the pressure drop. On the contrary, the small shell side velocity results in a small pressure drop on the shell side as compared to the values obtained from other optimization methods. Based on the pressure drops on both sides, comparing with the other methods, the annual operating cost given by WCA is lower by 68.3%, 6.13%, and 21% for the original design, GA and PSO respectively while it is much higher than FA and GSA. Thus, making the total discounted annual operating cost to be lower compared to GA and PSO.
Table 3. Optimal STHE design using different optimization methods.

| Parameter | Original design [21] | GA [3] | PSO [6] | FA [18] | GSA [19] | WCA (Present work) |
|-----------|-----------------------|--------|---------|---------|---------|-------------------|
| $D_s$ (m) | 0.539                 | 0.63   | 0.59    | 0.7276  | 0.62    | 0.623             |
| $d_o$ (m) | 0.025                 | 0.02   | 0.015   | 0.01575 | 0.015   | 0.02              |
| $B$ (m)   | 0.127                 | 0.12   | 0.1112  | 0.1054  | 0.11    | 0.135             |
| $N_t$     | 158                   | 391    | 646     | 924     | 718     | 400               |
| $S_r$ (m) | 0.031                 | 0.025  | 0.0187  | 0.01968 | 0.0187  | 0.0244            |
| $v_t$ (m/s)| 1.44                  | 0.87   | 0.93    | 0.677   | 0.75    | 0.99              |
| $R_{et}$  | 8.227                 | 4.068  | 3.283   | 2.408   | 3.102   | 4.281             |
| $f_t$     | 0.033                 | 0.041  | 0.044   | 0.049   | 0.046   | 0.040             |
| $P_{rt}$  | 55.2                  | 55.2   | 55.2    | 55.2    | 55.2    | 56.5              |
| $D_e$ (m) | 0.025                 | 0.019  | 0.0149  | 0.0156  | 0.0148  | 0.0193            |
| $A_s$ (m$^2$)| 0.0148               | -      | -       | -       | 0.01687 |
| $R_{es}$  | 25.281                | 18.327 | 15.844  | 15.004  | 15.814  | 15.814            |
| $P_{rs}$  | 7.5                   | 7.5    | 7.5     | 7.5     | 7.6     | 7.6               |
| $v_s$ (m/s)| 0.47                  | 0.43   | 0.495   | 0.4     | 0.476   | 0.385             |
| $h_t$ (W/m$^2$ K)| 619                  | 1,168  | 1,205   | 1,262   | 1,488   | 584               |
| $h_s$ (W/m$^2$ K)| 920                  | 1,034  | 1,288   | 1,156   | 1,512   | 985               |
| $U$ (W/m$^2$ K)| 317                  | 376    | 409.3   | 347.6   | 348     | 221               |
| $A$ (m$^2$) | 61.5                 | 52.9   | 47.5    | 56.6    | 54.98   | 47.6              |
| $L$ (m)   | 4.88                  | 2.153  | 1.56    | 1.64    | 1.317   | 1.941             |
| $\Delta P_t$ (Pa) | 49.245            | 14.009 | 16.926  | 9374    | 8449    | 17.458            |
| $f_s$     | 0.315                 | 0.331  | 0.337   | 0.3422  | 0.34    | 0.3377            |
| $\Delta P_s$ (Pa) | 24.909            | 15.717 | 21.745  | 12.768  | 17.962  | 9.844             |
| $C_i$ (€) | 19.007                | 17.599 | 16.707  | 18.202  | 17.639  | 16.720            |
| $C_o$ (€/year) | 1.304             | 440    | 523.3   | 210.2   | 290.1   | 413               |
| $C_{od}$ (€) | 8.012             | 2.704  | 3.215.6 | 1.231   | 1.642   | 2.541             |
| $C_{tot}$ (€) | 27.020            | 20.303 | 19.922.6| 19.433  | 19.281  | 19.261            |

Therefore, based on the total investment cost and annual discounted operating cost, a reduction of total cost by 28.7% is associated with the original design and saving of 5.1%, 3.3%, and 0.9% by using GA, PSO, and FA optimization method respectively. The comparisons of the total costs for each optimization methods are shown in Figure 4.

Figure 4. Comparison of the total cost for each optimization methods.
5. Conclusions
This paper examined the application of a recently proposed optimization method called the water cycle algorithm (WCA) to shell and tube heat exchanger optimization to lower the cost of a heat exchanger. The objective function considered is the total annual cost which includes the total investment cost and operating cost. Comparing the results obtained from the WCA, against the results obtained from other optimizers reported in published literature shows that WCA produced better results in terms of a reduction of investment cost as much as 12% and in operating cost of 68.3%, causing a decrease in total cost up to 28.7%. This illustrates the efficiency of the WCA in finding better solutions. Furthermore, WCA obtained the best values in 12 generations since WCA uses more efficient mechanisms to search for terms of exploration and exploitation.

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