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# Economic Optimization of a Shell-and-Tube Heat Exchanger (STHE) based on New Method by Grasshopper Optimization Algorithm (GOA)

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**Abstract:** Today, minimizing the cost of heat exchangers (HEs) is a major goal for the designer. In this study, a fast and reliable method is used to simulate, optimize design parameters and evaluate heat transfer enhancement and the economic optimization of STHE. Taking into account the importance of STHEs in industrial applications and the complexity in their geometry, the GOA methodology is adopted to obtain an optimal geometric configuration. The GOA is a metaheuristics search algorithm based on the mimics and simulates grasshoppers 'behavior in nature and the grasshoppers ' move to food sources. The total annual cost (TAC) (including the capital investment cost and the total operating cost consumption to overcome the pressure drop) is chosen as the objective function, and the design variables include number of tubes, number of tube passes, length of tubes, the arrangement of tubes, size and percentage of baffle cut, tube diameter, tubular step ratio have been considered. The developed algorithm is applied to two case studies and the results are compared with the original design and other optimization methods available in literature such as Genetic Algorithm (GA), Hybrid Genetic-Particle Swarm Optimization (GA-PSO), Gravitational Search Algorithm (GSA), Falcon Optimization Algorithm (FOA), Artificial Bee Colony (ABC), Bio- geography Based Optimization (BBO), Cuckoo Search Algorithm (CSA) and Firefly Algorithm (FFA). In order to investigate the feasibility of the proposed method, two case studies have been presented that show a significant TAC reduction of up to 30% with respect to traditional designed STHEs.

**Keywords:** Economic Optimization**,** Shell-and-Tube Heat Exchanger**,** Total Annual Cost**,** Grasshopper Optimization Algorithm

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## 1 Introduction

Nowadays, with the expansion of science and the advancement of industries, human needs have gone beyond problem solving to optimization. In many engineering issues, we usually have a cost or utility function or in general an objective function that we want to improve by improving the value of a system by optimizing its value. In the case of heat exchangers (HEs), we should also look for a plan to reduce the cost of HEs according to consumer needs. Among the different types of HEs, the STHEs is considered the most common of them due to the wide range of their application. The schematic view of the common STHE is shown in Fig. 1 [1].

According to scientific research developed in recent decades, optimization of STHEs has been largely based on minimizing system costs. Chauduri et al. [2] investigated these types of HEs with Simulated Annealing (SA) by applying and minimizing the heat transfer are and TAC. The major contribution of the present study can be identified in identifying better outcomes with more decision variables.

Almost a decade later, Salbas et al. [3] minimized the TAC using GA with variables such as outside diameter of the tubes, flow arrangement of the tubes, number of tube passages, outside shell diameter and baffles spacing, and achieved better results for the same study than previous studies. Caputo et al. [4], also using GA and the same objective function, evaluated three different cases in which shell diameter, outside tube diameter and baffles spacing were selected as design variables and resulted in a significant reduction in TAC.

Fesanghary et al. [5], studied the STHEs by the Harmony Search Algorithm (HSA) to minimize TAC. Then, Patel and Rao [6] and Sahin et al. [7], In order to optimize the cost of STHE for the case study of Caputo et al. [4], Respectively by PSO and ABC algorithm to obtain the results Better the simulation. Hadidi and Nazari [8] developed the BBO algorithm to minimize the TAC by modifying various geometric parameters such as baffle spacing, tube length, tube outer diameter, pitch size, etc. Hajibollahi et al. [9] optimized the design of STHEs using GA and PSO. They minimized their objective function TAC by selecting tube number, number of tube passes, inlet and outlet tube diameters, tube pitch ratio and tube arrangement parameters, and compared GA and PSO results to show that PSO yields better results.

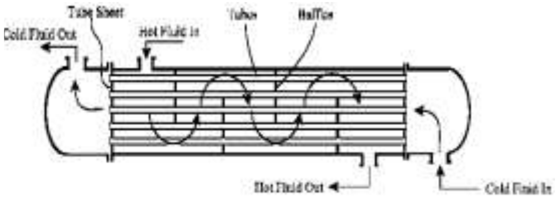


**Fig. 1** Diagram of a typical STHE [1].

Karimi et al. [10] studied a mixed materials heat exchanger network. They used two methods for minimizing the TAC such as total and partial decomposition. In these methods, networks divided to two separated subsystems including corrosive and anti-corrosive flows and investigated separately. The TAC is decreased using the partial decomposition more than another method. Rao and Patel [11] optimized the exchanger using a modified teaching–learning-based optimization algorithm (TLBO). They minimize their objective function by selecting decision variables for each STHE and plate-fin. Rao and Patel presented their results for two examples and compared them with the GA results and showed that the TLBO achieved better results than the GA. Karimi et al. [12] minimized the total annual cost in the shell and tube heat exchanger by ant colony optimization.

Different researches have been done on STHEs design with various objectives including minimization of TAC [13-29], maximization of effectiveness [11,13,30-32], minimization of pumping power [33-35], maximization of heat transfer rate [34-38], minimization of pressure loss [39], minimization of entropy generation units[35, 40,41], minimization of total weight or volume of the STHE [42,43], minimization of heat transfer area [33], and maximization of thermal efficiency [44].

In this study, the GOA suggested by Saremi et al. [45] is selected for optimization of a STHE. The objective function is the minimization of TAC. This study has provided a new and powerful methodology in the optimization of STHEs. Based on the proposed method, a complete computer program has been developed in MATLAB Version 2017a for the design of STHEs and reports of two different case studies are shown to demonstrate the effectiveness and accuracy of the proposed algorithm. This paper is organized as follows; Section 2 explains the mathematical formulation for the thermo-hydraulic design of a STHE, Section 3 renders the overview of GOA algorithm, Section 4 defines the target function, Section 5 describes the constraints and ranges of design variables. Subsequently Sections 6 depicts the computational results and analysis while Section 7 yields the final remarks.

## 2 Design Formulations of a STHE

### 2.1 Shell and Tube side heat transfer coefficient (HTC)

HTCs depend on the fluid velocity and the flow regime, and many equations have been proposed to determine the HTC for the flow inside the pipes. In terms of the in-tube flow phenomena and according to the flow regime of the HTCs as well as pressure drop (PD) calculations, the flow inside the pipe is divided in-to three regions including relaxation, transition and developed turbulent. The criterion of separation of these three states is Reynolds dimensionless number with the concept of mobility factor. The fluid is a deterrent to its movement. To calculate the laminar HTC we use Seider-Tate correlation [46,47], for transient conditions the Hausen correlation [48] and for the fully developed turbulent (turbulent area) flow conditions the well-known Dittus-Boelter [49] correlation; which is suitable for both gas or liquid fluid and is widely used in HTC in tubes.

$$h_t = K_{ht1} V_t^{\frac{1}{3}} \text{ for laminar flow } (Re_t \leq 2100) \tag{1}$$

$$h_t = K_{ht2} V_t^{\frac{2}{3}} - K_{ht3} \text{ for transition flow } (2100 < Re_t \leq 10^4) \tag{2}$$

$$h_t = K_{ht4} V_t^{\frac{4}{5}} \text{ for fully developed turbulent flow} (Re_t \geq 10^4) \tag{3}$$

In the above equations, the coefficients are calculated according to equations 4 to 7 and the fluid velocity inside the tube as per equation 8 [1]:

$$K_{ht1} = 1.86 \frac{k_t}{d_i} \left[ \left( \frac{\rho_t d_i}{\mu_t} \right) Pr_t \left( \frac{d_i}{L} \right) \right]^{\frac{1}{3}} \tag{4}$$

$$K_{ht2} = 0.116 \frac{k_t}{d_i} \left( \frac{\rho_t d_i}{\mu_t} \right)^{\frac{2}{3}} Pr_t^{\frac{1}{3}} \left[ 1 + \left( \frac{d_i}{L} \right)^{\frac{2}{3}} \right] \tag{5}$$

$$K_{ht3} = 14.5 \frac{k_t}{d_i} Pr_t^{\frac{1}{3}} \left[ 1 + \left( \frac{d_i}{L} \right)^{\frac{2}{3}} \right] \tag{6}$$

$$K_{ht4} = C \frac{k_t}{d_i} Pr_t^{\frac{2}{5}} \left( \frac{\rho_t d_i}{\mu_t} \right)^{\frac{4}{5}} \tag{7}$$

$(C = 0.024 \text{ for heating}. 0.023 \text{ for cooling}$

The fluid velocities in the tube, Reynolds number & Prandtl are calculated from the following equations [1]:

$$V_t = \frac{m_t}{\frac{\pi}{4} d_i^2 \rho_t} \left( \frac{N_P}{N_t} \right) \tag{8}$$

$$Re_t = \frac{\rho_t V_T d_i}{\mu_t} \tag{9}$$

$$Pr_t = \frac{\mu_t C_{pt}}{k_t} \tag{10}$$

The HTCs for the shell side is calculated from the following equations [1]:

$$h_s = K_{hs1} V_s^2 + K_{hs2} V_s + K_{hs3} \text{ for } Re_s \leq 250 \tag{11}$$

$$h_s = K_{hs4} V_s^{0.6633} \text{for } 250 < Re_s \leq 250000 \tag{12}$$

In equations 11 and 12, the coefficients are calculated as follows [1]:

$$K_{hs1} = -3.722 \times 10^{-5} \frac{F_P F_L J_S K^{\frac{2}{3}} c_P^{\frac{1}{3}} \rho^2 d_O}{\mu^{\frac{5}{3}}} \tag{13}$$

$$K_{hs2} = 0.03843 \frac{F_P F_L J_S K^{\frac{2}{3}} c_P^{\frac{1}{3}} \rho}{\mu^{\frac{2}{3}}} \tag{14}$$

$$K_{hs3} = \rho \left( C_{NS.inlet} V_{NS.inlet}^2 + C_{NS.outlet} V_{NS.outlet}^2 \right) \tag{15}$$

$$K_{hs4} = 0.08747 \frac{F_P F_L J_S K^{\frac{2}{3}} c_P^{\frac{1}{3}} \rho^{0.6633}}{\mu^{0.33} d_O^{0.3367} B_C^{0.5053}} \tag{16}$$

Where $F_P$ is the step coefficient for triangular and square rotation (45º, 30º and 60º) and the one for square 90º (linear square) 0.85, $F_L$ leakage coefficient with fixed tube 0.9, shell and tube U Fig 0.85 and the HE with a floating head is considered to be 0.8. $J_S$, the correction factor for the distance between the baffles is determined from the following relation [1]. $J_S$ is the correction factor for in coordinate baffle spacing [50]:

$$J_S = \begin{cases} \frac{(N_B - 1) + (B_{in}/B)^{2/3} + (B_{out}/B)^{2/3}}{(N_B - 1) + (B_{in}/B) + (B_{out}/B)} & \text{for } Re_S < 100 \\ \frac{(N_B - 1) + (B_{in}/B)^{2/5} + (B_{out}/B)^{2/5}}{(N_B - 1) + (B_{in}/B) + (B_{out}/B)} & \text{for } Re_S \geq 100 \end{cases} \tag{17}$$

The Reynolds Number based on tube outside diameter and $V_S$ fluid velocity for the shell-side are determined as follows [1]:

$$V_s = \frac{m_S}{\rho B \left[ (D_S - D_B) + \frac{(D_B - d_O)(P_T - d_O)}{P_{CF}} \right]} \qquad P_{CF} \begin{cases} 1 & \text{. for } 30^\circ \text{and } 90^\circ \text{ layouts} \\ \sqrt{2}/2 & \text{. for } 45^\circ \text{ layouts} \\ \sqrt{3}/2 & \text{. for } 60^\circ \text{ layouts} \end{cases} \tag{18}$$

$$Re_s = \frac{\rho d_O V_S}{\mu} \tag{19}$$

### 2.2 Shell and Tube side PD

Designing a HE is not enough just for the sake of heat transfer issues and the governing equations. The PD on either side of a fluid in a HE is an important indicator that should be calculated and optimized for the design geometry of the exchanger and the flow conditions. A low-PD often means not using the potential of the HE and having a low OHTC, and on the other hand, a high-PD, although typically associated with a high HTC, generally requires the use of a stronger pump for moving and passing fluid through the exchanger and its associated paths. This inevitably increases the pump's power consumption and associated costs. The total tube-side PD, $\Delta P_T$ for single-shell comprises the PD in the straight tubes ($\Delta P_{TT}$), the PD in the tube entrances, exits and reversals ($\Delta P_{TE}$), and the PD in nozzles ($\Delta P_{TN}$) [47]:

$$\Delta P_T = \Delta P_{TT} + \Delta P_{TE} + \Delta P_{TN}$$
$$\Delta P_T = K_{PT1} N_P L V_T^{2+m_f} + K_{PT2} V_T^2 + K_{PT3} \tag{20}$$

Where

$$K_{PT1} = \frac{2F_C \left( \frac{\rho_t d_i}{\mu_t} \right)^{m_f} \rho}{d_i} \tag{21}$$

$$K_{PT2} = 0.5 \alpha_R \rho \tag{22}$$

$$K_{PT3} = \rho \left( C_{TN.inlet} V_{TN.inlet}^2 + C_{TN.outlet} V_{TN.outlet}^2 \right) \tag{23}$$

Where the constants are defined by the Reynolds number as follows [1]:

$$F_C = \begin{cases} 16 & Re \leq 2100 \\ 5.36 \times 10^{-6} & 2100 < Re < 3000 \\ 0.0791 & Re \geq 3000 \end{cases} \tag{24}$$

$$m_f=\begin{cases}-1 & Re\le 2100\\ 0.949 & 2100<Re<3000\\ -0.25 & Re\ge 3000\end{cases} \quad (25)$$

$$\alpha_R=\begin{cases}3.25N_P-1.5 & 500\le Re\le 2100\\ 2N_P-1.5 & Re>2100\end{cases} \quad (26)$$

$$C_{TN,inlet,outlet}=C_{SN,inlet,outlet}=\begin{cases}0.75 & 100\le Re_{TN,inlet,outlet}, Re_{SN,inlet,outlet}\le 2100\\ 0.375 & Re_{TN,inlet,outlet}, Re_{SN,inlet,outlet}>2100\end{cases} \quad (27)$$

$$Re_{TN,inlet}=\frac{\rho d_{TN,inlet}V_{TN,inlet}}{\mu} \quad (28)$$

$$V_{TN,inlet}=\frac{m_T}{\rho\left(\frac{\pi d_{TN,inlet}^2}{4}\right)} \quad (29)$$

$$Re_{TN,outlet}=\frac{\rho d_{TN,outlet}V_{TN,outlet}}{\mu} \quad (30)$$

$$V_{TN,outlet}=\frac{m_T}{\rho\left(\frac{\pi d_{TN,ioutlet}^2}{4}\right)} \quad (31)$$

The total PD, $\Delta P_S$ for the shell-side in one shell includes the PD in the straight section of the shell ($\Delta P_{SS}$) And the PD in nozzles ($\Delta P_{NS}$) [47]. The total PD for the shell-side per shell is found by:

$$\Delta P_S=\Delta P_{SS}+\Delta P_{NS}=K_{PS1}V_S^{1.875}+K_{PS2}V_S^{1.843}+K_{PS3} \quad (32)$$

Where

$$K_{PS1}=18\left(5\frac{B}{D_S}-1\right)\begin{pmatrix}N_B-1\\+R_S\end{pmatrix}\frac{aD_S\rho}{d_e}\left(\frac{B_C}{0.2}\right)^{m_{fo}}\left(\frac{\rho d_e}{\mu}\right)^{-0.12} \quad (33)$$

$$K_{PS2}=90\left(1-\frac{B}{D_S}\right)\begin{pmatrix}N_B-1\\+R_S\end{pmatrix}\frac{bD_S\rho}{d_e}\left(\frac{B_C}{0.2}\right)^{m_{fo}}\left(\frac{\rho d_e}{\mu}\right)^{-0.157} \quad (34)$$

$$K_{PS3}=\rho\left(C_{NS,inlet}V_{NS,inlet}^2+C_{NS,outlet}V_{NS,outlet}^2\right) \quad (35)$$

Where $R_S$, the correction factor for unequal baffle spacing, and $d_e$, shell-side equivalent diameter, $C_{De}$, the pitch configuration factor, are determined respectively by [1]:

$$R_S=\left(\frac{B}{B_{in}}\right)^{1.8}+\left(\frac{B}{B_{out}}\right)^{1.8} \quad (36)$$

$$d_e=C_{De}\frac{P_T^2}{d_O}+d_O \quad (37)$$

$$C_{De}=\begin{cases}\dfrac{4}{\pi} & \text{for square pitch}\\ \dfrac{2\sqrt{3}}{\pi} & \text{for triangular pitch}\end{cases} \quad (38)$$

$Re_{NS}$ is the nozzle side Reynolds number and given by [1]:

$$Re_{NS,inlet}=\frac{\rho d_{NS,inlet}V_{NS,inlet}}{\mu} \quad (39)$$

$$V_{NS,inlet}=\frac{m_S}{\rho\left(\frac{\pi d_{NS,inlet}^2}{4}\right)} \quad (40)$$

$$Re_{NS,outlet}=\frac{\rho d_{NS,inlet}V_{NS,outlet}}{\mu} \quad (41)$$

$$V_{NS,outlet}=\frac{m_S}{\rho\left(\frac{\pi d_{NS,outlet}^2}{4}\right)} \quad (42)$$

For HEs with multiple shells connected in series, the total PD on the tube and shell-side fluid are found by [1]:

$$\Delta P_{total}=N_{shells}\Delta P_S+N_{shells}\Delta P_T \quad (43)$$

$N_{shells}$ is the number of shells connected given by [1]:

$$N_{shells}=\frac{\ln\left(\frac{1-RP}{1-P}\right)}{\ln W}\ \text{for } R\ne 1$$
$$N_{shells}=\frac{\left(\frac{P}{1-P}\right)\left(1+\frac{\sqrt{2}}{2}-X_P\right)}{X_P}\ \text{for } R=1 \quad (44)$$

The rate of heat exchange between cold and hot currents is given by the following equation [1]:

$$Q=m_hC_{ph}(T_{hi}-T_{ho})=m_cC_{Pc}(T_{co}-T_{ci}) \quad (45)$$

The heat exchanger surface area is calculated from the following relation [1]:

$$A=\frac{Q}{UF\Delta T_{LMTD}} \quad (46)$$

In the above relation, the logarithmic mean temperature difference (LMTD) is calculated by considering the cross-flow of relation (3) [1]:

$$\Delta T_{LMTD}=\frac{(T_{hi}-T_{co})-(T_{ho}-T_{ci})}{\mathrm{Ln}\left[\frac{T_{hi}-T_{co}}{T_{ho}-T_{ci}}\right]} \quad (47)$$

The shell-side fluid flow in the STHEs relative to the tube-side is a complex combination of diagonally aligned, incongruent and transverse cross currents. Therefore, in the HE equation, the LMTD must be corrected by the correction factor F described in Equation 4, about the complex dependence of the HEs depending on the four inlet and outlet temperatures of the two fluids [51].

$$F=\sqrt{R^2+1}\times\frac{\mathrm{Ln}\,\frac{1-P}{1-PR}}{(R-1)\mathrm{Ln}\left(\frac{2-P(R+1-\sqrt{R^2+1})}{2-P(R+1+\sqrt{R^2+1})}\right)} \quad (48)$$

$$R=\frac{T_{hi}-T_{ho}}{T_{co}-T_{ci}} \quad (49)$$

$$P=\frac{T_{co}-T_{ci}}{T_{hi}-T_{ci}} \quad (50)$$

The overall heat transfer coefficient (OHTC) is one of the most important indices in HEs. Detection of the main resistance to heat transfer within a HE is of great importance for the design and evaluation of its performance. Equation 7 is used to calculate the OHTC [51]:

$$U=\frac{1}{\frac{1}{h_S}+R_{sf}+\frac{d_o}{2k}ln\left(\frac{d_o}{d_i}\right)+\frac{d_o}{d_i}Rtf+\frac{d_o}{d_i}\frac{1}{h_T}} \quad (51)$$

**3 Optimization technique**

The GOA Algorithm is used to optimize and solve the problem. Grasshopper is a destructive insect known as a pest of crops. There are eleven thousand locust species [45]. As a grasshopper reaches its adult stage, it undergoes egg, nymph and adult stages, as shown in Fig2.

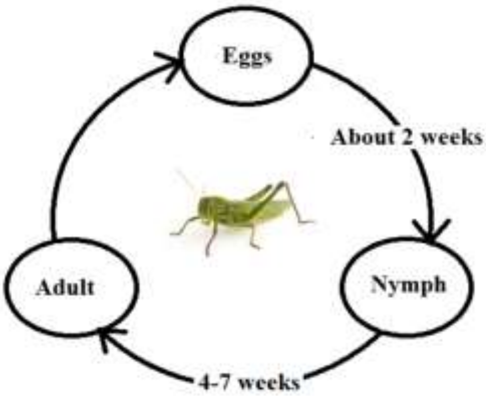


**Fig. 2** The Lifecycle of a Grasshopper [52].

The GOA is a metaheuristic optimization method and, like the PSO algorithm, falls under the category of swarm intelligence and population based algorithms. The mathematical model used to simulate the behavior of the grasshoppers is as follows [45]:

$$\mathrm{X_i} = \mathrm{S_i} + G_\mathrm{i} + \mathrm{A_i} \tag{52}$$

In the above relation $\mathrm{X_i}$, $\mathrm{S_i}$, $G_\mathrm{i}$, and $\mathrm{A_i}$ represent the position of the i[th] grasshopper, the social interaction of i[th] grasshopper, the gravity force on the i[th] grasshopper, and the wind advection of i[th] grasshopper.

To create the random behavior, the Eq. (52) can be expressed as follows:

$$\mathrm{X_i} = r_1\mathrm{S_i} + r_2 G_\mathrm{i} + r_3\mathrm{A_i} \tag{53}$$

Where the coefficients $r_1$, $r_2$, $r_3$ are random numbers between zero and one. Social interaction represents one of the main concepts of position and movement behavior of the grasshoppers which is expressed as:

$$\mathrm{S_i} = \sum_{\substack{\mathrm{i=1} \\ \mathrm{j \neq i}}}^{\mathrm{N}} s(\mathrm{d_{ij}})\ \hat{\mathrm{d}}_{\mathrm{ij}} \tag{54}$$

where $\mathrm{d_{ij}}$ denotes the distance between the ith and jth grasshoppers and is calculated as $\mathrm{d_{ij}} = |\mathrm{X_j} - \mathrm{X_i}|$. $\hat{\mathrm{d}}_{\mathrm{ij}} = \frac{(\mathrm{X_j - X_i})}{\mathrm{d_{ij}}}$ is a unit vector from ith grasshopper to the jth grasshopper. $s$ is a function that is used to determine the strength of the social forces and demonstrates how the social interaction (attraction and repulsion) of the grasshoppers is affected. $s$ is expressed as:

$$s(r) = f e^{\frac{-r}{l}} - e^{-r} \tag{55}$$

where f denotes the intensity of attraction and $l$ is the attractive length scale. Changing these parameters lead to social behaviors in synthetic grasshoppers and significantly change comfort, attraction, and repulsion zones. The gravity force in Eq. (52) is calculated as follows:

$$G_i = -g\hat{e}_g \tag{56}$$

where $g$ is the gravitational constant and $\hat{e}_g$ is the unit vector toward the center of the. For the direction of the wind advection, the following relation can be expressed:

$$A_i = \mathrm{u}\ \hat{e}_\omega \tag{57}$$

where u is the constant drift and $\hat{e}_\omega$ is the unit vector in the wind direction. Considering the above relations, Eq. (52) can be extended as follows:

$$X_i = \sum_{\substack{\mathrm{j=1} \\ \mathrm{j \neq i}}}^{N} s(|X_j - X_i|)\ \frac{(X_j - X_i)}{d_{ij}} - g\hat{e}_g + \mathrm{u}\ \hat{e}_\omega \tag{58}$$

where $N$ represents the number of grasshoppers. Eq. (58) is used to model the grasshoppers in the open space. Since the grasshoppers quickly reach the comfort zone and the group does not converge to one point, this model cannot be used directly to solve optimization problems. In this regard, the applicable model given in Eq. (58) is presented as follows:

$$X_i^d = c\left(\sum_{\substack{j=1 \\ j \neq i}}^{N} c\,\frac{ub_d - lb_d}{2}\, s(X_j^d - X_i^d|)\ \frac{(X_j - X_i)}{d_{ij}}\right) + \hat{T}_d \tag{59}$$

$lb_d$ and $ub_d$ represent the upper and lower limits in the dth dimension, and $\hat{T}_d$ is the best solution found in the dth dimension to the specified iteration. c is a decreasing constant and has a controlling role between extraction and exploration. Initially since the first term of Eq. (59) (discovery term) should be given more value, c is a large number. This constant decreases over time and is lead to the best answer. $c$ is updated by the following relation:

$$c = c_{max} - l\,\frac{c_{max} - c_{min}}{L} \tag{60}$$

In this work, $c_{max} = 1$ and , $c_{min} = 0.00001$. $l$ is the number of the current iteration and $L$is the maximum number of iterations of the algorithm. The implementation steps of GOA can be summarized as follows:

*Step1: Initialize the parameters of algorithm ;*
*Step2: Produce the population of grasshopper randomly ;*
*Step3: Assess the position of each grasshopper and calculate its merit ;*
*Step4: Identify the best grasshopper as the target ;*
*Step5: Repeat Steps 6 to 12 until the stop condition is established ;*
*Step6: Repeat steps 7 to 11 for each grasshopper ;*
*Step7:* $c = c_{max} - l\,\frac{c_{max} - c_{min}}{L}$ *;*
*Step8: Update the value of c ;*
*Step9: Update it for each grasshopper ;*
*Step10: Calculate the merit of the new grasshopper ;*
*Step11: If the new grasshopper's merit is better than the target, set the new grasshopper as the target ;*
*Step12: If the stop condition is not met, go to step 5, otherwise go to end ;*
*Step13: End*

The GOA, like PSO, ant and bee optimization algorithms are in the category of swarm intelligence and population base algorithms.

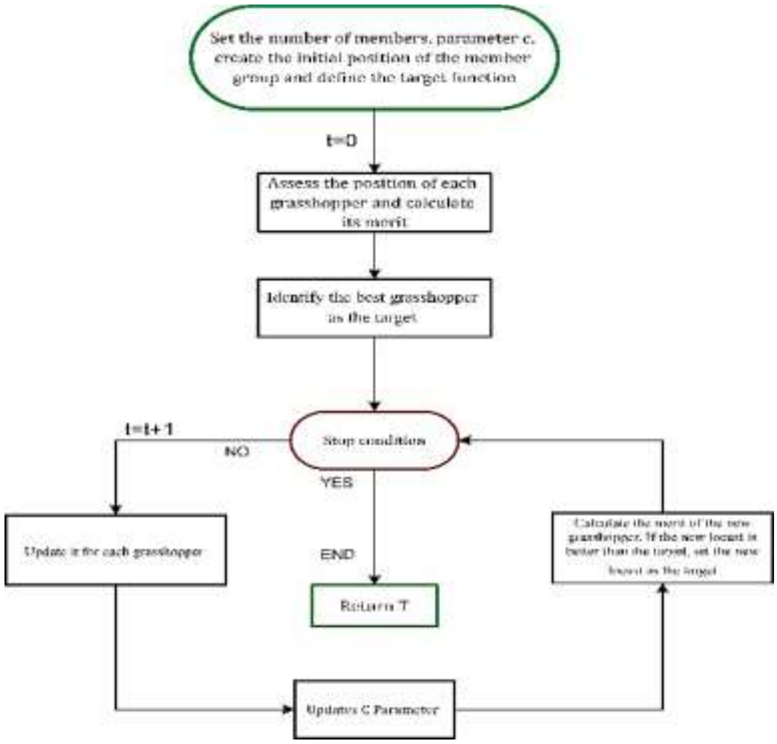


**Fig. 3** Flowchart of the GOA algorithm.

and unlike genetic algorithm, the generation is not produced. First, an initial generation is formed and corrected by considering the social interaction between the members of the generation. In the GOA, optimization begins by creating a random set of solutions. Then, the search factors (grasshoppers) update their position based on the first term of Eq. (59). Accordingly, the second term of Eq. (59) is updated as the best solution in each iteration. In addition, factor c is calculated according to the Eq. (60).

The distance between the grasshoppers in the interval of [1,4] is mapped in each iteration. Updating the position is repeated until the termination conditions are satisfied. The position and competence of the best objective are ultimately given as the best approximation to the output. The flowchart of the proposed GOA algorithm is presented in Fig. 3.

## 4 Cost Function

In this work, the TAC ($C_{tot}$) has been defined as the objective function. The TAC is calculated taking into account the capital investment cost $C_i$, energy cost $C_e$, annual operating cost $C_o$ and the total discounted operating cost $C_{od}$ as considered by Caputo et al. [4].

$$C_{tot} = C_i + C_{od} \tag{61}$$

The capital investment is computed as a function of heat exchanger surface area according to Tall et al. [53].

$$C_i = a_1 + a_2 A^{a_3} \tag{62}$$

Where $a_1$ is installation factor, $a_2$ is material factor and $a_3$ is the exponent. By assuming some of the streams will later be corrosive, requiring special materials of construction. For example, for heat exchanger made with stainless steel (SS) for both shell-and-tubes, these coefficients are $a_1 = 8000$, $a_2 = 259.2$ and $a_3 = 0.93$ [4]. The total operating cost for pumping power to dominate friction losses is computed from the following equation [4].

$$C_o = P C_e H \tag{63}$$

$$P = \frac{1}{\eta}\left(\frac{m_T}{\rho_T}\Delta P_T + \frac{m_S}{\rho_S}\Delta P_S\right) \tag{64}$$

$$C_{od} = \sum_{x=1}^{ny} \frac{C_o}{(1+i)^x} \tag{65}$$

In all of the cases, all of the values of discounted operating costs are calculated with ny =10 yr, annual discount rate i = 10%, energy cost Ce =0.12 V/kW h and work hours annual H= 7000 yr/h [23].

According to objective function, in eq. (61) has been shown relation between objective function and other decision variables such as area and pressure drops shell and tube sides and this equation can be considered as follows:

$$C_{tot} = f(d_i, L, N_t, P_T, B_C, N_{tp}, B) \tag{66}$$

As seen, objective function is depended on inner diameter tubes, length tubes, number of tubes, tube pitch ratio, number of tube pass, baffle spacing, baffle cut and arrangement of tubes. However, there is no any analytical solution for objective function. In another words, objective function isn't as differentiable function. Therefore, it should be use optimization algorithm for achieving global optimum that it used try and error method.

## 5 Bounds for design parameters

In this study, for higher accuracy, eight geometrical parameters of the STHE-are considered as design variables. The range of design variables of the geometric parameters is given in Table 1.

## 6 Results and Discussion

Given the importance of STHEs in industrial applications and the use of meta-heuristic algorithms, the GOA approach outlined in Section 3 uses an economic perspective to find the optimal design of STHEs. The code was developed in MATLAB version 2017a and simulated using a laptop with an AMD FX-7600P RadeonR7 12 Compute Cores 4C+8G, 2.7 GHz and 8 GB of RAM. To estimate the suitability and reliability of the proposed method, two case studies with valid comparisons have been considered.

### 6.1 Case Study

This case study is adapted from Sinnott et al. [51]. The main design for this case involves single shell pass and two tube passes in which methanol is used as hot fluid in the shell and brackish water as the cold side fluid stream. The heat duty of the heat exchanger is 4.34 MW. The physical thermal properties of the flows, including their flow rate and temperature, are shown in Table 2.

Many researchers have studied this case for the economic optimization design of a STHE. Caputo et al. [4] used GA, Segundo et al. [54] used the FOA, Karimi et al.[55] used GA-PSO, Hadidi and Nazari[8] used the BBO, Sahin et al.[15] used the ABC algorithm, Mohanty[38] used FFA, Asadi et al.[17] used the CSA, Mohanty[56] used GSA, For the STHE design. Table 3 and fig 4 shows the optimum results for case study 1 comparing results for the Ref.[4, 8, 15, 17, 38, 51, 54-56]. According to the results, using the GOA technique for TAC shows a decrease of 30.5, 18.64, 3.46, 4.1, 11.4, 11.78, 2.12, 6 and 10.6% respectively, compared to previous studies.

Fig. 5 shows the convergence of the objective function obtained using GOA and etc. It can be observed that the objective function converges within about 30 iteration for this case.

**Table1** Variables for optimization of the STHE.

| Variables | Values | Increment | Number of Solutions |
|---|---|---|---|
| Tube layout patern | Triangular(30º); Rotated triangular(60º); Square(90º); Rotated square(45º) | - | 4 |
| Number of tube passes | 1; 2; 4; 8 | - | 4 |
| Tube length (m) | 1.83; 2.44; 2.5; 3; 3.66; 4; 4.88; 6; 6.10; 7.32; 8.53; 9.75; 10.7; 11.58 | - | 14 |
| Tube diameter (mm), (di : do) | (12.8:16) ; (14.10:15.88) ; (14.8:18) ; (16.8:20) ; (17.27:19.05) ; (18.8:22) ; (21.8:25) ; (23.62:25.40) ; (26.8:30) ; (28.8:32) ; (29.97:31.75) ; (34.8:38) ; (36.8:40) ; (66.8:70) | - | 14 |
| Baffle spacing (m) | 0.20 to 0.50 | 0.001 | 3000 |
| Baffle cut (%) | 0.10 to 0.50 | 0.001 | 4000 |
| Tube number | 50 to 2000(step 1) | 1 | 1950 |
| Tube pitch ratio | 1.25 to 2(step 0.001) | 0.001 | 750 |

**Table2** Process input and physical properties for bath case studies.

| Process Parameters | Case study 1 | | Case study 2 | |
|---|---|---|---|---|
| | Shell side: methanol | Tube side: Sea- Water | Shell Side: Kerosene | Tube side: Crude oil |
| Mass flow rate (kg/s) | 27.80 | 68.90 | 5.52 | 18.80 |
| $T_i$(ºC) | 95 | 25 | 199 | 37.8 |
| $T_O$(ºC) | 40 | 40 | 93.3 | 76.7 |
| $\rho$(kg/m$^3$) | 750 | 995 | 850 | 995 |
| $C_p$(KJ/kgK) | 2.84 | 4.2 | 2.47 | 2.05 |
| $\mu$(Pa.s) | 0.00034 | 0.0008 | 0.0004 | 0.00358 |
| $\mu_W$(Pa.s) | 0.00038 | 0.00052 | 0.00036 | 0.00213 |
| K(W/m.K) | 0.19 | 0.59 | 0.13 | 0.13 |
| $R_f$(m$^2$.K/W) | 0.00033 | 0.0002 | 0.00061 | 0.00061 |

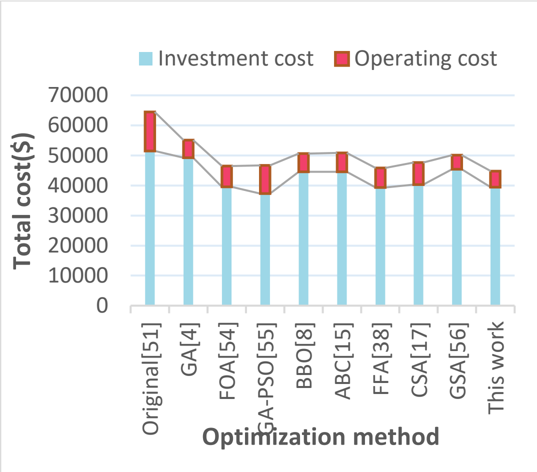


**Fig. 4** Total cost comparison for case study 1.

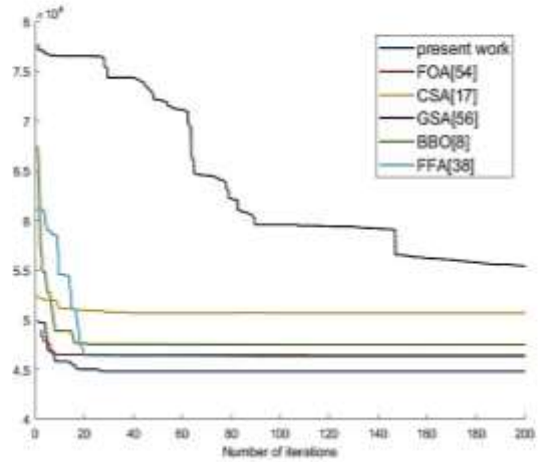


**Fig. 5** Convergence of GOA for case study 1.

**Table 3** Optimal geometry and thermal performance for case 1.

| Parameters | Original design[51] | GA[4] | FOA[54] | GA-PSO [55] | GOA Present work |
|---|---|---|---|---|---|
| Area(m2) | 278.6 | 262.8 | 195.5 | 201.7 | 173.91 |
| $\upsilon_T(m/s)$ | 0.75 | 0.69 | 0.995 | 0.543 | 0.81 |
| $Re_t$ | 14925 | 10936 | 10000 | 10805.7 | 14205 |
| $\Delta P_T(pa)$ | 6251 | 4298 | 6093 | 4525.7 | 6996.4 |
| $h_T\,(W/m^2C)$ | 3812 | 3762 | 9961 | 2759.8 | 4050.5 |
| $\upsilon_S(m/s)$ | 0.58 | 0.44 | 0.54 | 0.42 | 0.57072 |
| $Re_s$ | 18381 | 11075 | 8677 | 13307.8 | 19992 |
| $\Delta P_S(pa)$ | 35789 | 13267 | 17496 | 25911 | 9499.8 |
| $h_S\,(W/m^2C)$ | 1573 | 1740 | 2381 | 1138.8 | 3009.9 |
| $U_{overall}\,(W/m^2C)$ | 615 | 660 | 888.5 | 445.2 | 842.8 |
| $N_T$ | 918 | 1567 | 2715 | 1267 | 1095 |
| $L_T(mm)$ | 4.83 | 3.379 | 2.269 | 3.49 | 1.83 |
| $D_S(mm)$ | 0.894 | 0.830 | 0.6822 | 0.6472 | 0.90033 |
| $d_i(mm)$ | 16 | 12.8 | 8.08 | 11.6 | 14.10 |
| $d_o(mm)$ | 20 | 16 | 10.1 | 14.5 | 15.88 |
| $P_T\,(m)$ | 0.025 | 0.020 | - | - | 0.0246 |
| $C_i$ | 51507 | 49259 | 39513 | 37291 | 39416 |
| $C_o$ | 2111 | 947 | | 1528 | 878.43 |
| $C_{od}$ | 12973 | 5818 | 6906 | 9392 | 5397.6 |
| $C_{tot}$ | 64480 | 55077 | 46419 | 46683 | 44813 |

**6.2 Case Study 2**

This case study was taken from Kern [57]. The heat changer is a kerosene-crude oil exchanger. In this case the heat duty was 1.44 MW. The original design assumed the exchanger with four tube-side passages, one shell-side passage, and with square pitch pattern. The process parameters and the physical properties of the both shell and tube side fluids are summarized in Table 2. Table 4 shows the design parameters of the HE compared to the references [4, 38, 56, 57].

The comparison of the results of the second case study with the reports(fig 5) of Kern[57], Caputo et al.[4], Mohanty[38], Mohanty[56], Sahin et al.[15], Asadi et al.[17] and Hadidi and Nazari[8] respectively, showed a decrease by 29.16, 5.73, 1.5, 0.8, 5.36, 3.2 and 3.38 percent. Fig. 7 shows the convergence of the objective function obtained using GOA and etc. It can be observed that the objective function converges within about 50 iteration for this case.

**Table 4** Optimal geometry and thermal performance for case 2.

| Parameters | Original design [31] | GA [21] | FFA [28] | GSA [30] | GOA Present work |
|---|---|---|---|---|---|
| Area(m2) | 61.5 | 52.9 | 56.6 | 54.98 | 45.104 |
| $\upsilon_T(m/s)$ | 1.44 | 0.87 | 0.677 | 0.75 | 2.7191 |
| $Re_t$ | 8227 | 4068 | 2408 | 3102 | 9673.4 |
| $\Delta P_T(pa)$ | 49245 | 14009 | 9374 | 8449 | 13775 |
| $h_T\left(W/m^2C\right)$ | 619 | 1168 | 1262 | 1488 | 1555.2 |
| $\upsilon_S(m/s)$ | 0.47 | 0.43 | 0.4 | 0.476 | 0.74762 |
| $Re_s$ | 25281 | 18327 | 14448 | 15004 | 25419 |
| $\Delta P_S(pa)$ | 24909 | 15717 | 12768 | 17962 | 12100 |
| $h_S\left(W/m^2C\right)$ | 920 | 1034 | 1156 | 1512 | 2741 |
| $U_{overall}\left(W/m^2C\right)$ | 317 | 376 | 347.6 | 348 | 387.48 |
| $N_T$ | 158 | 391 | 924 | 718 | 54 |
| $L_T(m)$ | 4.88 | 2.153 | 1.64 | 1.317 | 1.3 |
| $D_S(m)$ | 0.539 | 0.63 | 0.7276 | 0.62 | 0.17 |
| $d_i(mm)$ | 20 | 16 | 12.6 | 12 | 12.8 |
| $d_o(mm)$ | 25 | 20 | 15.75 | 15 | 16 |
| $P_T$ (m) | 0.031 | 0.025 | 0.01968 | 0.0187 | 0.02048 |
| $C_i$ | 19007 | 17599 | 18202 | 17639 | 16955 |
| $C_o$ | 1304 | 440 | 210.2 | 290.1 | 355.79 |
| $C_{od}$ | 8012 | 2704 | 1231 | 1642 | 2186.2 |
| $C_{tot}$ | 27020 | 20303 | 19433 | 19281 | 19141 |

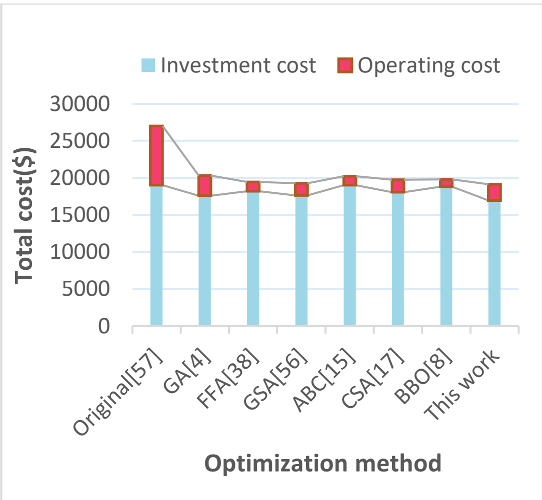


**Fig.6** Total cost comparison for case study 2.

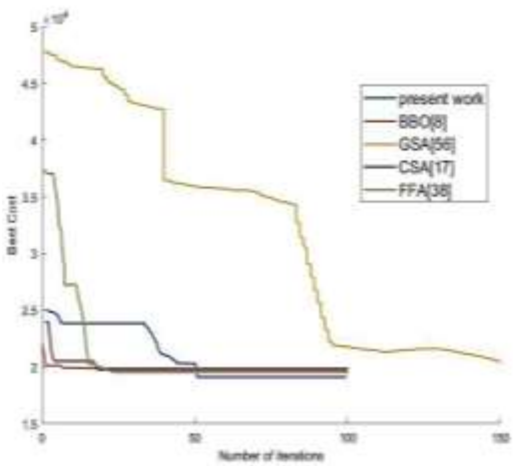


**Fig. 7** Convergence of GOA for case study 2.

## 7 Conclusion

In the present work, the GOA algorithm is applied to obtain an optimal design for a STHE with the aim of cost minimization. In the present work, the GOA algorithm is applied to obtain an optimal design for a STHE with the purpose of cost minimization. The aim of the optimization acts is minimization of the TAC which includes both the investment cost and the total discounted annual operating cost. The proposed methodology is cover to two case studies and the results are compared with other design optimization methods and the original design.

As observed in both the case studies, it is found that the application of the present methodology can provide a suitable design for a STHE from economic point of view. The results show that a reduction of HE area by 37.6%, operating cost 58.4% and total cost by 30.5% can be achieved by the present method for a HE of heat duty 4.34MW. Similarly, the reduction in area by 26.7%, operating cost 72.7% and total cost by 29.16% can be achieved for a HE of heat duty 1.44MW. Simultaneously the comparison of the results with other methods available in literature such as GA, GSA, ABC, GA-PSO, FOA, BBO, CSA and FFA show that the present method using GOA algorithm is an effective method of design optimization for a STHE from economic point of view. Hence this way can be adopted as a proper and reputable tool for design of a STHE in industrial applications.

**Nomenclature**

| | |
|---|---|
| $A$ | *Heat exchanger surface area ($m^2$)* |
| $A_f$ | *Annualized factor for capital costs* |
| $B$ | *Central baffle spacing (m)* |
| $B_C$ | *Baffle cut as percent of inside shell diameter* |
| $B_{in}$ | *Inlet baffle spacing* |
| $B_{out}$ | *Outlet baffle spacing* |
| $C$ | *Constant (0.024 for heating, 0.023 for cooling)* |
| $C_{cap}$ | *Heat exchanger capital cost due to heat transfer area* |
| $C_{De}$ | *Pitch configuration factor* |
| $C_{exe}$ | *Heat exchanger cost* |
| $C_P$ | *Fluid specific heat at constant pressure and average temperature (J/kg K)* |
| $C_{pump}$ | *Capital cost for pumps* |
| $C_{pumping}$ | *Operating cost for pumps* |
| $D_B$ | *Outside diameter of the tube bundle* |
| $d_e$ | *Shell-side equivalent diameter* |
| $d_i$ | *Tube inside diameter (m)* |
| $d_{NS,inlet}$ | *Inner diameter of the inlet nozzle for the shell-side fluid* |
| $d_{NS;outlet}$ | *Inner diameter of the outlet nozzle for the shell-side fluid* |
| $d_O$ | *Tube outside diameter (m)* |
| $D_S$ | *Internal diameter of the shell(m)* |
| $d_{TN,inlet}$ | *Inner diameter of the inlet nozzle for the tube-side fluid* |
| $d_{TN;outlet}$ | *Inner diameter of the outlet nozzle for the tube-side fluid* |
| $F$ | *Correction factor for logarithmic mean temperature difference* |
| $F_L$ | *The leakage factor to allow for all the stream leakage, which is a function of bundle configuration* |
| $F_P$ | *The pitch factor, which depends on the tube layout of the bundle* |
| $F_{SC}$ | *Correction factor for the shell construction* |
| $F_{TC}$ | *Tube count constant that accounts for the incomplete coverage of the shell diameter by the tubes, due to necessary clearances between the shell and the tube bundle and tube omissions due to the location of pass partition plates for multiple pass designs* |
| $hs$ | *Shell-side heat transfer coefficient ($W/m^2$ K)* |
| $h_t$ | *Tube-side heat transfer coefficient ($W/m^2$ K)* |
| $H_y$ | *Annual plant operation time (h/yr)* |
| $J_S$ | *Correction factor for unequal baffle spacing* |
| $k$ | *Thermal conductivity* |
| $K_S$ | *Shell-side constant for pressure drop relationship* |
| $K_T$ | *Tube-side constant for pressure drop relationship* |
| $L$ | *Tubes length (m)* |
| $LMTD$ | *Logarithmic mean temperature difference (K)* |
| $M$ | *Fluid mass velocity* |
| $m_s$ | *Mass flow rate on the shell-side (kg/s)* |
| $m_t$ | *Mass flow rate on the tube-side (kg/s)* |
| $N_b$ | *Number of baffles* |
| $N_s$ | *Number of sealing strips* |
| $N_{shells}$ | *Number of shells* |
| $N_T$ | *Total number of tubes* |
| $N_{tp}$ | *Number of tube passes* |
| $P$ | *The thermal effectiveness of the exchanger* |
| $P_c$ | *Pitch configuration factor* |
| $P_{CF}$ | *Pitch correction factor for flow direction* |
| $Pr_s$ | *Shell side Prandtl number* |
| $Pr_t$ | *Tube side Prandtl number* |
| $P_T$ | *Tube pitch(m)* |
| $Q$ | *Heat transferred per unit time(watts)* |
| $R$ | *The ratio of the two heat capacity flowrates* |
| $Re$ | *Reynolds number* |
| $R_S$ | *Correction factor for unequal baffle spacing* |
| $R_{sf}$ | *Shell side fouling resistance ($m^2$ K/W)* |
| $R_{tf}$ | *Tube side fouling resistance ($m^2$ K/W)* |
| $TAC$ | *Total annual cost ($)* |
| $T_{ci}$ | *Cold fluid inlet temperature (K)* |
| $T_{co}$ | *Cold fluid outlet temperature (K)* |
| $T_{hi}$ | *Hot fluid inlet temperature (K)* |
| $T_{ho}$ | *Hot fluid outlet temperature (K)* |
| $U$ | *Overall heat transfer coefficient ($W/m^2$ K)* |
| $v_s$ | *Shell side fluid velocity (m/s)* |
| $v_t$ | *Tube side fluid velocity (m/s)* |
| *Greek Symbols* | |
| $\Delta_{LMTD}$ | *Logarithmic mean temperature difference* |
| $\Delta P$ | *Pressure drop for fluid stream* |
| $\eta$ | *Efficiency of pump* |
| $\mu$ | *Viscosity of fluid stream (Pa s)* |
| $\rho$ | *Density of fluid stream ($kg/m^3$)* |
| *Subscripts* | |
| $c$ | *Cold stream* |
| $h$ | *Hot stream* |
| $i$ | *Inlet* |
| $o$ | *Outlet* |
| $S, s$ | *Shell side* |
| $T, t$ | *Tube side* |

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