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Effectiveness of evolutionary algorithms for optimization of heat exchangers

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ABSTRACT

This paper comprehensively investigates performance of evolutionary algorithms for design optimization of shell and tube heat exchangers (STHX). Genetic algorithm (GA), firefly algorithm (FA), and cuckoo search (CS) method are implemented for finding the optimal values for seven key design variables of the STHX model. ϵ-NTU method and Bell-Delaware procedure are used for thermal modeling of STHX and calculation of shell side heat transfer coefficient and pressure drop. The purpose of STHX optimization is to maximize its thermal efficiency. Obtained results for several simulation optimizations indicate that GA is unable to find permissible and optimal solutions in the majority of cases. In contrast, design variables found by FA and CS always lead to maximum STHX efficiency. Also computational requirements of CS method are significantly less than FA method. As per optimization results, maximum efficiency (83.8%) can be achieved using several design configurations. However, these designs are bearing different dollar costs. Also it is found that the behavior of the majority of decision variables remains consistent in different runs of the FA and CS optimization processes.

1. Introduction

Shell and tube heat exchangers (STHX) play a critical role in operation of many industrial plants including oil refineries, power stations, and manufacturing sites. By far, they are the most widely used type of heat exchanger used in different industries. Optimal design of STHX is a challenging engineering task. Several criteria such as efficiency and capital, operating, and energy costs can be considered in the design. As mentioned in [1], the design process has an iterative nature and includes several trials for obtaining a reasonable configuration that fulfills the design specifications and satisfies the trade-off between pressure drops and thermal exchange transfers. No doubt, this process is massively time-consuming and expert expensive. Furthermore, there is no guarantee that the final design is optimal in terms of considered criteria due to the limited capability of the design engineers in consideration and evaluation of all admissible designs. Budget constrains during the design phase even worsen this. So it is not surprising to see real world STHX that their designs is far away from being optimal.

Fig. 1 displays the layout and fluid flows of a typical STHX. Baffles placed along the tube bundle force the fluid to flow through tubes [2]. Baffles simply intensify the turbulent level and improve the shell film coefficient of heat transfer. Detailed information about components of a STHX can be found in [3]. The existing literature on design optimization of STHX greatly deal with finding the optimal values for baffles (spacing and ratio) and the number, length, diameter, and arrangement of tubes. Also tube pitch ratio has been considered in some studies as well [4,5]. Two approaches are often used for design optimization. Some authors focus on simultaneous optimization of several variables [1,4], while others fix some less important variables and try to find the optimal values for the most important design variables [6,7].

Gradient descent optimization methods cannot be applied for optimal design of STHX. This is due to a high level of calculation complexity and discrete nature of decision variables making the objective function nondifferentiable. Also these algorithms are highly likely to be trapped in local optima due to the massiveness of variable search space. Evolutionary algorithms, in contrast, are able to efficiently explore the search space and find approximate optimal solutions in a short time. They are also global optimization methods and can avoid local optima using different mechanisms and operations. Therefore, using evolutionary algorithms has become a standard practice for design of heat exchangers in the last decade [8,9].

Despite many breakthroughs in the field of evolutionary optimization (mainly reported in publications handled by IEEE Computational Intelligence Society), genetic algorithm is the most used

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method by process engineering researchers for design optimization of heat exchangers [8–14]. Several optimization methods have been introduced in recent years that outperform genetic algorithm in term of optimization results. Also some of these methods are even computationally less demanding. Examples of these methods are particle swarm optimization [15,16], cuckoo search [17], imperialist competitive algorithm [18], bee colony optimization [19], and firefly algorithm [20]. These methods show different performances in different engineering applications. A conceptual comparison of these methods for several case studies can be found in [21]. A few of these algorithms have been recently employed for design and optimization of heat exchangers [22–28].

The purpose of this paper is to comprehensively compare performance of the genetic algorithm, firefly algorithm, and cuckoo search method for the design of STHXs. To the best of our knowledge, this is the first study where firefly algorithm and cuckoo search method are employed for optimal design of STHXs. Seven design variables are considered as part of the optimization process. These are tube arrangement, pitch ratio, diameter, length, quantity, baffle spacing ratio, and baffle cut ratio. Optimization is purely done for maximizing the efficiency. Cost implications of this optimization approach are then analyzed and discussed. Performance of optimization algorithms is compared on their ability to find permissible and optimal configurations. The behavior of the seven design variables are also studied in detail. Simulation experiments are done for an approximate thermal model of a real world STHX.

The rest of this paper is organized as follows. Section 2 briefly introduces the STHX model used in this study. Optimization algorithms investigated in this study are briefly described in Section 3. Section 4 represents simulations results. Finally, conclusions are provided in Section 5.

2. Modelling shell and tube heat exchanger

The efficiency of the TEMA E-type STHX is calculated as,

$$\epsilon = 2\left(1 + C^* + \sqrt{1+C^{*2}}\,\frac{1-e^{-NTU\sqrt{1+C^{*2}}}}{1+e^{-NTU\sqrt{1+C^{*2}}}}\right)^{-1} \tag{1}$$

where the heat capacity ratio (C^*) is calculated as,

$$C^* = \frac{C_{min}}{C_{max}} = \frac{min(C_s, C_t)}{max(C_s, C_t)} = \frac{min((\dot{m}c_p)_s, (\dot{m}c_p)_t)}{max((\dot{m}c_p)_s, (\dot{m}c_p)_t)} \tag{2}$$

where subscripts s and t stand for shell and tube respectively. The number of transfer units is defined as,

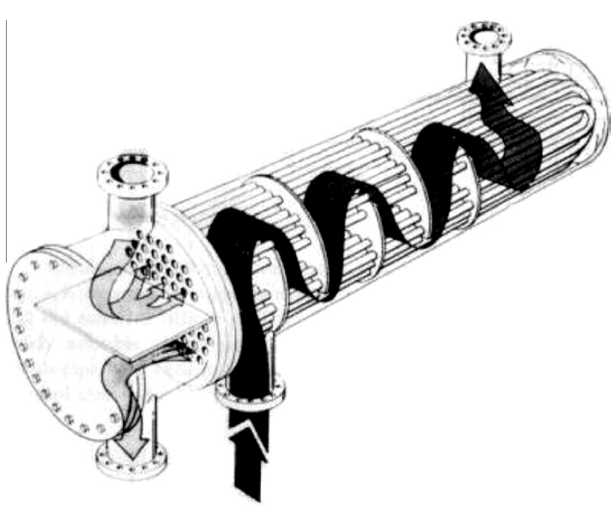

Fig. 1. The layout of a STHX with shell and tube fluid flows [2].

$$NTU = \frac{U_o\,A_t}{C_{min}} \tag{3}$$

where C_{min} is,

$$C_{min} = min(C_h, C_c) \tag{4}$$

where C_h and C_c are the hot and cold fluid heat capacity rates, i.e., $C_h = (\dot{m}c_p)_h$ and $C_c = (\dot{m}c_p)_c$. $\dot{m}$ is the fluid mass flow rate. Specific heats c_p are assumed to be constant.

The overall heat transfer coefficient (U_o) in (3) is then computed as,

$$U_o = \left(\frac{1}{h_o} + R_{o,f} + \frac{d_o\,ln(d_o/d_i)}{2k_w} + R_{i,f}\,\frac{d_o}{d_i} + \frac{d_o}{h_i d_i}\right)^{-1} \tag{5}$$

where $L, N_t, d_i, d_o, R_{i,f}, R_{o,f}$, and k_w are the tube length, number, inside and outside diameter, tube and shell side fouling resistances and thermal conductivity of tube wall respectively. h_i and h_o are heat transfer coefficients for inside and outside flows, respectively.

The total tube outside heat transfer area is calculated as.

$$A_t = \pi\,L\,d_o\,N_t \tag{6}$$

where L and d_o are the tube length and outside diameter.

The tube side heat transfer coefficient (h_i) is calculated as,

$$h_i = 0.024\,\frac{k_t}{d_i}\,Re_t^{0.8}\,Pr_t^{0.4} \tag{7}$$

for $2500 < Re_t < 124,000$. k_t and Pr_t are tube side fluid thermal conductivity and Prandtl number respectively. The tube flow Reynold number (Re_t) is also defined as,

$$Re_t = \frac{m_t\,d_i}{\mu_t A_{o,t}} \tag{8}$$

where m_t is the tube mass flow rate and $A_{o,t}$ is the tube side flow cross section area per pass,

$$A_{o,t} = 0.25\pi d_i^2\,\frac{N_t}{n_p} \tag{9}$$

where n_p is the number of passes.

The average shell side heat transfer coefficient is calculated using the Bell–Delaware method correlation,

$$h_s = h_k\,J_c\,J_l\,J_b\,J_s\,J_r \tag{10}$$

where h_k is the heat transfer coefficient for an ideal tube bank,

$$h_k = j_i\,c_{p,s}\left(\frac{\dot{m}_s}{A_s}\right)\left(\frac{k_s}{c_{p,s}\mu_s}\right)^{\frac{2}{3}}\left(\frac{\mu_s}{\mu_{s,w}}\right)^{0.14} \tag{11}$$

where j_i is the Colburn j-factor for an ideal tube bank. A_s is also the cross flow area at the centerline of the shell for one cross flow between two baffles. $\frac{\mu_s}{\mu_{s,w}}$ is the viscosity ratio at bulk to wall temperature in the shell side. J_c, J_l, J_b, J_s, and J_r in (10) are the correction factors for baffle configuration (cut and spacing), baffle leakage, bundle and pass partition bypass streams, bigger baffle spacing at the shell inlet and outlet sections, and the adverse temperature gradient in laminar flows.

The STHX total cost is made up of capital investment (C_{inv}) and operating (C_{opr}) costs [1],

$$C_{total} = C_{inv} + C_{opr} \tag{12}$$

There are several methods for determining the price of STHX. Here we use the Halls method for estimation of the investment cost as detailed in [29] (alternative cost estimation methods can be found in [30]). C_{inv} as a function of the total tube outside heat transfer surface area (A_t) is defined as,

$$C_{inv} = 8500 + 409\,A_t^{0.85} \tag{13}$$