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# Investigating the effect of properties variation in optimum design of compact heat exchanger using segmented method

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ABSTRACT

A plate fin heat exchanger was optimally designed by selecting effectiveness and total annual cost as two simultaneous fitness functions using particle swarm optimization algorithm. Due to the variation of temperature and pressure in the exchanger passages, the non-uniform properties are occurred throughout the heat exchanger. To consider the effect of mentioned properties variation, the heat exchanger was segmented into sub-exchangers (for example 100) and properties were determined at the mean temperature for each small exchanger. The optimum results in the case of segmented (SEG) heat exchanger were compared with conventional properties evaluation or un-segmented (UN-SEG) heat exchanger in which the fluid properties are just determined at the mean temperature of outlet and inlet of heat exchanger. To generalize the results, the optimization was performed for the different hot stream inlet temperatures. The optimum results indicate that, the effectiveness decreases while the total annual cost increases in the case of SEG compared with UN-SEG. For example, total annual cost increases 2.8, 6.1, 11.7, 15.9 and 19.9%, respectively for the hot side inlet temperature of 400, 500, 600, 700 and 800 K, and for the heat exchanger with effectiveness of 0.8. Furthermore, effectiveness decreases about 1, 2, 3, 4.5 and 5.5%, respectively for the hot side inlet temperature of 400, 500, 600, 700 and 800 K.

## 1. Introduction

In the design of heat exchangers, especially the air side of compact heat exchanger, variation of fluid properties causes a flow misdistribution which decreases the heat exchanger performance (Shah and Sekulic, 2003). One of the common heat exchanger (HE) type is compact heat exchanger (CHE) with a large heat transfer surface area per unit volume. Plate fin heat exchanger (PFHE) is a type of CHE which is used in power and industrial plants (Fig. 1). Due to the low convection heat transfer coefficient of gases, the extended surface (fin) are used to increase the surface area (Kays and London, 1984). Rectangular striped fins with high compactness and convection heat transfer coefficient are used in this study (Fig. 1). Due to the variation of temperature in both sides of HE, all the fluid properties such as viscosity, Prandtl number, specific heat and density vary during the HE. The common strategy is evaluation of all the fluid properties at the average of inlet and outlet temperature for each side. It is worth noting, for the case of rating problem (no outlet temperature is specified), iterative try and error procedure is required. Furthermore, there are a lot of efforts for optimization of different types of HEs with various objectives and various decision variables using the various methods. For example, Hajabdollahi et al. performed the optimization of different types of HEs including the shell and tube (Hajabdollahi et al., 2012; Sanaye and Hajabdollahi, 2010a;

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**Nomenclature**

| | |
|---|---|
| $A$ | heat transfer surface area ($m^2$) |
| $af$ | annualized factor (–) |
| $b$ | fin height (m) |
| $c_p$ | specific heat (J/kg K) |
| $c$ | fin pitch (m) |
| $C_{min}$ | minimum of total heat capacity (W/K) |
| $C_{max}$ | maximum of total heat capacity (W/K) |
| $C^*$ | ratio of total heat capacity |
| $C_{in}$ | investment cost ($/year) |
| $c_{op}$ | operational cost ($/year) |
| $D_h$ | hydraulic diameter (m) |
| $f$ | Fanning friction factor (–) |
| $G$ | mass flux (kg/$m^2$ s) |
| $h$ | convection heat transfer coefficient (W/$m^2$ K) |
| $j$ | Culburn factor (–) |
| $k_f$ | fin conductivity (W/m K) |
| $L_c$ | length in the cold stream direction (m) |
| $L_h$ | length in the hot stream direction (m) |
| $L_n$ | no flow length (m) |
| $N$ | number of operating hours in a year |
| $N_p$ | number of passages (–) |
| $NTU$ | number of transfer units (–) |
| $Pr$ | Prandtl number (–) |
| $r$ | rate of interest (–) |
| $Re$ | Reynolds number (–) |
| $St$ | Stanton number (–) |
| $TAC$ | total annual cost ($/year) |
| $t_f$ | fin thickness (m) |
| $t_w$ | plate thickness (m) |
| $U$ | overall heat transfer coefficient (W/$m^2$ K) |
| $V$ | volumetric flow rate ($m^3$/s) |
| $x$ | length of fin (m) |
| $y$ | equipment life time (year) |

*Greek abbreviation*

| | |
|---|---|
| $\varepsilon$ | heat exchanger effectiveness (–) |
| $\alpha$ | is the heat transfer surface area per unit volume ($m^2/m^3$) |
| $\eta$ | compressor efficiency (–) |
| $\beta$ | ratio of hot and cold surface area (–) |
| $\mu$ | viscosity (Pa s) |
| $\nu$ | specific volume ($m^2$/kg) |
| $\Delta P$ | pressure drop (Pa) |
| $\sigma$ | ratio between $A_{flow}$ and $A_{front}$ ($A_{flow}/A_{front}$) |
| $\varphi_e$ | unit price of electrical ($MWh^{-1}$) |

*Subscripts*

| | |
|---|---|
| c | cold |
| flow | flow |
| h | hot |
| in | inlet |

Khosravi et al., 2015), condenser (Hajabdollahi et al., 2011a), plate fin (Sanaye and Hajabdollahi, 2010b; Ahmadi et al., 2010; Hajabdollahi et al., 2011b), fin tube (Hajabdollahi et al., 2011c; Nia et al., 2013), rotary regenerator (Sanaye and Hajabdollahi, 2009) as well as gasket plate (Hajabdollahi et al., 2013a) by using different algorithms including the Genetic Algorithm (Hajabdollahi et al., 2011a,b,c, 2012, 2013a; Sanaye and Hajabdollahi, 2009, 2010a,b; Khosravi et al., 2015; Ahmadi et al., 2010; Nia et al., 2013), Particle Swarm Algorithm (Hajabdollahi et al., 2011a), Firefly Algorithm (Khosravi et al., 2015) and by considering the different objective functions including total annual cost (Hajabdollahi et al., 2011a,c, 2012, 2013a; Sanaye and Hajabdollahi, 2010a,b; Khosravi et al., 2015; Ahmadi et al., 2010; Nia et al., 2013), effectiveness (Sanaye and Hajabdollahi, 2009, 2010a,b; Khosravi et al., 2015; Hajabdollahi et al., 2011b,c, 2013a), pressure drop (Hajabdollahi et al., 2011b; Sanaye and Hajabdollahi, 2009), exergy efficiency (Hajabdollahi et al., 2012; Ahmadi et al., 2010), entropy generation (Ahmadi et al., 2010) and temperature approach (Nia et al., 2013). Some other authors focused on optimization of PFHE by the different methods in different criteria. For example, some researchers performed second law optimization of PFHE by considering the unit number of entropy generation as fitness function (Mishra et al., 2009; Zhou et al., 2014; Rao and Patel, 2010; Zhang et al., 2010). Furthermore, some researchers used the Genetic Algorithm (Najafi et al., 2011; Chen and Chen, 2013), some other used the combination of Genetic Algorithm with other method (Peng and Ling, 2008; Guo et al., 2014) and some other authors used the other population base algorithm (Yousefi et al., 2012; Kotcioglu et al., 2013; Zarea et al., 2014).

In the all studied works, the fluid properties were considered constant or temperature dependent and evaluated at the mean temperature of heat exchanger. Usually, designers determine the fluid properties at the average temperature of HE. Although this seems a reasonable assumption, but it is still a raw assumption and make some differences for the results obtained from theoretical solution compared with that in the real problem especially for the high temperature variation or for the high effectiveness application. In addition, due to the variation of properties in the heat exchanger passages, the non-uniform fluid properties such as outlet temperature profile which causes the flow maldistribution. To track the effect of properties maldistribution, the heat exchanger is divided into a number of small heat exchangers. To illustrate this method, an unmixed–unmixed cross flow heat exchanger is considered. This heat exchanger is divided into $m \times n$ divisions, as illustrate in Fig. 2. Hot and cold fluid are segmented into $m$ and $n$ division, respectively. The heat exchanger relations such as energy balance, pressure drop and $\epsilon$-$NTU$ equations are solved for each segment separately and the outlet of each segment is determined.

In this paper, a PFHE is modeled and optimized using two simultaneous fitness functions including effectiveness and total annual cost. In addition, the exchanger is modeled and optimized by division of it into $m \times n$ segments (SEG) and their results are compared with common method in which the heat exchanger is not segmented (UN-SEG) and properties are evaluated at the average temperature of inlet and outlet. To generalize the results, the optimization is performed for different hot side inlet temperatures and results are reported.

## 2. Thermal modeling of each segment

In the heat exchangers the first law of thermodynamic in steady state yields:

$$\Delta H_c = \Delta H_h \tag{1}$$

where subscripts $c$ and $h$ denote the cold and hot stream. In this study, effectiveness-NTU method is applied to estimate the heat exchanger thermal performance. The best heat