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Multi-Objective Optimization of a Discrete Plate Finned-Tube Evaporator Design Using Entropy Generation Minimization Method

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Abstract

In this article, a multi-objective design optimization of a discrete plate finned-tube evaporator by means of entropy generation minimization technique is presented. The objectives are to minimize the dimensionless entropy generation number associated with the heat transfer, with finite temperature difference, and the fluid friction with finite pressure drop using Non-dominated Sort Genetic Algorithm-II (NSGA-II) technique. The Pareto optimal frontier was obtained and a final optimal solution was selected. This multi-objective optimization resulted in a refrigerant flow length of 0.228 m (2.98% less than the refrigerant flow length before optimization) and an air flow length through the evaporator of 0.055 m (8.3% less than the air flow length through the evaporator before optimization). Thus, the heat transfer area of the optimized evaporator is less than the original evaporator, so the compact heat exchanger matrix becomes less than the original compact heat exchanger matrix.

Keywords: *discrete plate finned-tube heat exchanger, entropy generation minimization, multi-objective optimization, thermal-hydraulic design*

Introduction

A compact heat exchanger is a device which is used to transfer thermal energy, between two or more fluids, at different temperatures and in thermal contact, which are single-phase or two-phase (Choi, Kim, Lee, & Kim, 2010). The heat exchangers can be classified in several ways such as, according to the transfer process, number of fluids and heat transfer mechanism. The classification according to the surface compactness deals with one of the important class of heat exchanger called compact heat exchangers. Compact heat exchangers can be classified in several ways such as, according to the transfer process, the amount of liquid and the heat transfer mechanism (Lankinen, Suihkonen, & Sarkomaa, 2003). However, the dimensions of compact heat exchanger optimizations are very dependent on the irreversibilities of heat transfer and pressure drop, which are both conflicting with each other. Therefore multi-objective optimization is needed to get a set of optimal solutions that can provide flexibility to the designer to determine the dimensions as desired.

Multi-objective optimization is an optimization problem with more than one purpose function. The existence of this multi-objective optimization problem cannot be denied and avoided. There are so many problems, both in the real world and in mathematics, that naturally have more than one goal to be optimized. Among these objective functions, it is possible to have conflicts between one or several objectives with other objectives (Bui & Alam, 2008) and such problems give rise to a series of optimal solutions known as Pareto optimal solutions, and not a single optimal solution. So it is important to find not only Pareto-optimal solution, but as much as possible from them. This is because each of the two solutions is a trade-off between the objective (Deb, 2014).

Numerous experimental and numerical studies have been conducted on the heat transfer and resistance characteristics of finned tubes. These studies mainly focus on herringbone wavy finned tubes (Wang, Hwang, & Lin, 2002; Wang, Liaw, & Yan, 2011; Wang, & Liaw, 2012; Wongwises, & Chokeman, 2005), and plain finned tubes (Wang, & Chi, 2000; Hasan, & Siren, 2003; Kim, & Kim, 2005), while fewer studies exist, however, on the heat transfer and resistance characteristics of discrete plate finned tubes. Finned-Tube heat exchangers, having discrete plate fins, have been widely used to enhance the heat transfer performance of evaporator in refrigerators and freezers under dry and frosting conditions (Choi, Kim, Lee, & Kim, 2010). An evaporator is designed to accomplishing a heat transfer duty, with wall temperatures considered to be fairly uniform (Kiatsiriroat, Siriplubplala, & Nuntaphan, 1998). Two important issues in the design of evaporators are maximum efficiency and minimum cost for a particular application. A well-designed and highly efficient evaporator can help in saving large amount of energy (Sagia & Paignigiannis, 2003). The design processes involves decision making concerning both geometrical (e.g. heat transfer surface and free flow passage) and operational parameters (e.g. flow rates and temperatures of the stream) aimed at accomplishing a certain heat transfer duty at the penalty of pumping power (Hermes, 2012).

Entropy generation minimization (or thermodynamic optimization) is the method of modelling and optimization of real devices that owe their thermodynamic imperfection to heat transfer, mass transfer, and fluid flow irreversibilities (Bejan, 1996). Bejan (Bejan, 1982, 1996) developed the entropy generation minimization (EGM) approach to the heat exchanger optimization design. In this approach, Bejan (Bejan, 1982) took into account two types of the irreversibilities in a heat exchanger, namely, the transfer of heat across the stream-to-stream temperature difference and the frictional pressure drop that accompanies the circulation of fluid through the apparatus. Pussoli (Pussoli, Barbosa Jr, da Silva, & Kaviany, 2012) combined the experimentally validated modelling for the air-side heat transfer and pressure drop with entropy generation minimization (EGM) theory of Bejan (Bejan, 1982) to determine the optimum characteristics of peripheral finned-tube evaporators. Hermes (Hermes, 2013) assessment of the thermal-hydraulic design approaches was introduced by Hermes (Hermes, 2012), for designing condensers and evaporators for refrigeration system spanning from household to light commercial applications. Hermes (Hermes, 2012) proposed an algebraic formulation which expresses the dimensionless rate of entropy generation as a function of the number of transfer units, the fluid properties, the thermal-hydraulic characteristics, and the operating conditions for a heat exchanger with uniform temperature.

The purpose of this paper is to combine the basis of correlation proposed by Hermes (Hermes, 2012, 2013) for entropy generation minimization with the basis of empirical correlation for heat transfer and flow friction characteristics proposed by Wang, et. al (Yuan, Mei, Hu, Wang, & Yang, 2013) to determine the optimum characteristics of discrete plate finned-tube evaporator. The value of entropy generation was divided into two parts, i.e., entropy generation related to heat transfer and to fluid friction. Since the entropy generations related to heat transfer and pressure drop have conflicting objectives, in which increase in one objective leads to a decrease of another and vice versa, in order to find the best values of decision variables, these objectives were optimized simultaneously in a multi-objective optimization process which balanced the two conflicting objectives.

Thermal and Hydraulic Design

Thermo-hydraulic modelling is performed under the following assumption that an evaporator can be regarded as a heat exchanger with uniform wall temperature and the model was formulated based on energy, entropy and momentum balances in the secondary fluid flow (air) assuming the evaporator has an even lump, as follow (Hermes, 2012, 2013):

$$Q = \dot{m}c_p(T_o - T_i) = \varepsilon\,\dot{m}c_p(T_s - T_i) \tag{1}$$

where Q, $\dot{m}$, c_p, T_o, T_i and T_s denotes the rate of heat transfer, mass flow rate of air, specific heat of air, temperature air inlet, temperature air outlet and surface temperature of evaporator, respectively. The effectiveness (ε) of evaporator with both fluids unmixed is calculated from (Hermes, 2012):

$$\varepsilon = 1 - e^{-NTU} = \frac{T_o - T_i}{T_s - T_i} \tag{2}$$

with number of transfer units (*NTU*) is defined as (Hermes, 2012):

$$NTU = \frac{hA}{\dot{m} c_p} = \frac{UA}{\dot{m} c_p} \tag{3}$$

Here, A is outside the total heat transfer surface area, including fins and tubes, as follows:

$$A = A_p + A_f \tag{4}$$

where A_p and A_f are the heat transfer surface area of tube outside and fins, respectively. As follows (Shah & Sekulic, 2003):

$$A_p = \pi D_o (L_1 - \delta N_f L_1) N_t + 2\left(L_2 L_3 - \frac{\pi D_o^2}{4} N_t \right) \tag{5}$$

$$A_f = 2\left[L_2 L_3 - \left(\frac{\pi D_o^2}{4} \right) N_t \right] N_f L_1 + 2 L_3 \delta N_f L_1 \tag{6}$$

where N_t and N_f are the total number of tubes in an evaporator and the number of fins per unit length in the fin pitch direction; D_o is tube outside diameter; δ is the fin thickness; L_1, L_2 and L_3 are refrigerant flow length, air flow length, and no-flow height, respectively (Fig. 1).

The Fanning factor f and Colburn factor j are defined in Eq. (7) and (8) by Wang et al. (Wang, Chang, Hsieh, & Lin, 1996) for plate fin-and-tube heat exchanger having plane fins as follows:

$$j = 0.394\,\mathrm{Re}_{D_c}^{-0.392}\left(\frac{\delta}{D_c}\right)^{-0.0449} N^{-0.0897}\left(\frac{F_p}{D_c}\right)^{-0.212} \quad (7)$$

$$f = 1.039\,\mathrm{Re}_{D_c}^{-0.418}\left(\frac{\delta}{D_c}\right)^{-0.104} N^{-0.0935}\left(\frac{F_p}{D_c}\right)^{-0.197} \quad (8)$$

This correlations is valid for $800 \le \mathrm{Re}_{D_c} \le 7500$. The properties are evaluated at the average temperature of the air passing over the finned tubes in the heat exchanger.

In this paper, entropy generation minimization number was conducted on the basis of correlation proposed by Hermes (Hermes, 2012, 2013) as follows:

$$Ns = \Theta^2 NTU^{-1} + \frac{f}{2j}\dot{u}^2\,\mathrm{Pr}^{2/3}\,NTU \quad (9)$$

where $\dot{u} = u_c / \sqrt{c_p T_m}$ is a dimensionless core velocity, and $\Theta = (T_o - T_i)/T_s$ a dimensionless temperature difference with both T_o and T_i known from measurement. The first and second terms of the right hand side of equation (9) stand for the dimensionless entropy generation rates associated with the heat transfer with a finite temperature difference and a finite pressure drop, respectively. These two types of dimensionless entropy generation were considered as two separate conflicting objective functions. To gain more description about the entropy generation minimization approach please refer to the following Refs. (Hermes, 2012, 2013)

Objective Functions, Decision Variables and Constraints

The two objective functions defined in this paper were as follow:

Non-dimensioned entropy generation related to heat transfer:

$$Ns_{\Delta T} = \Theta^2 NTU^{-1} \quad (10)$$

Non-dimensioned entropy generation related to fluid friction:

$$Ns_{\Delta P} = \frac{f}{2j}\dot{u}^2 \Pr^{2/3} NTU \tag{11}$$

In the current study, decision variables were the design parameters of the evaporator as follows: (i) Θ: the dimensionless temperature difference; (ii) $\dot{m}$: air mass flow rate ; (iii) c_p: specific heat of air at constant pressure; (iv) U: overall heat transfer coefficient; (v) D_o: tube outside diameter; (vi) δ: fin thickness; (vii) N: number of tube row; (viii) F_p: fin pitch; (ix) N_t: total number of tubes; (x) N_f: number of fins per unit length in the fin pitch direction; (xi) L_1: refrigerant flow length; (xii) L_2: air flow length; (xiii) L_3: no-flow height; (xiv) $\Pr$; Prandtl number; (xv) u_c: maximum core velocity; and (xvi) Q: heat transfer rate. Table 1 shows the decision variables, bounds, and constraints that will be optimized simultaneously.

Results and Discussion

The two objective functions (Eqs (10) and (11) respectively), decision variables, bounds and constraints as introduced in Table 1 was optimized in a multi-objective optimization process using NSGA-II. Fig. 2 illustrates the Pareto frontier in the objective space.

Fig. 3 presented the normalized form of the Pareto frontier obtained in multi-objective optimization scenario. Commonly, it is better to work with the normalized data of the Pareto frontier instead of the real values. The horizontal and vertical axes in Fig. 3 are the normalized form of the objective functions that defined as follow respectively:

$$Ns^*_{\Delta T} = \frac{Ns_{\Delta T} - Ns_{\Delta T,\min}}{Ns_{\Delta T,\max} - Ns_{\Delta T,\min}} \tag{12}$$

and

$$Ns^*_{\Delta P} = \frac{Ns_{\Delta P} - Ns_{\Delta P,\min}}{Ns_{\Delta P,\max} - Ns_{\Delta P,\min}} \quad (13)$$

where $Ns_{\Delta T,\min}$ and $Ns_{\Delta T,\max}$ are the minimum and maximum values of $Ns_{\Delta T}$ in the Pareto frontier. In the same way $Ns_{\Delta P,\min}$ and $Ns_{\Delta P,\max}$ are the minimum and maximum values of $Ns_{\Delta P}$ in the Pareto frontier.

In the case of multi-objective optimization, the process of decision making is required for the selection of the final solution among the optimum points which exist on the Pareto frontier and this process is performed using a definition of an ideal point (also called equilibrium point) as shown in Fig. 3 by point 'I'. At the ideal point (point 'I'), both objective functions ($Ns_{\Delta T}$ and $Ns_{\Delta P}$) have minimum values and thus both $Ns^*_{\Delta T}$ and $Ns^*_{\Delta P}$ are zero. It is noted that point 'I' does not exist in actual practice and hence, merely an ideal point. In the present paper, the point of Pareto frontier which has minimum distance from the point 'I' is considered as a final optimum solution. The solution at this point attains the minimum possible value for both the objective functions ($Ns_{\Delta T}$ and $Ns_{\Delta P}$). Hence, values of $Ns^*_{\Delta T}$ and $Ns^*_{\Delta P}$ for the selected final optimal solution in the multi-objective optimized design are 0.481 and 0.487, respectively.

Table 2 shows the geometry values of the design parameters before optimization and subsequent optimization. This multi-objective optimization resulted the refrigerant flow length is 0.228 m (2.98% less than the refrigerant flow length before optimization) and the air flow length through the evaporator is 0.055 m (8.3% less than the air flow length through the evaporator before optimization). Thus, the heat transfer area of the optimized evaporator is less than the original evaporator, so the compact heat exchanger matrix becomes less than the original compact heat exchanger matrix.

Conclusions

In this study, a multi-objective optimization of a discrete plate finned-tube evaporator was presented based on entropy generation minimization proposed by Hermes (2012) and the basis of empirical correlations for heat transfer and flow friction characteristics proposed by Wang, et al. (1996). It was found that both entropy generation rates associated with finite temperature difference and the finite pressure drop of the discrete plate finned-tube evaporator were in optimum state. Therefore, multi-objective optimization of this type of heat exchanger led to the best trade-off between entropy generation, related to heat transfer, and entropy generation related to fluid friction. It was shown that this approach, in terms of separating entropy generation, enabled a better design of the evaporator with less total entropy generation and less heat transfer surface area than the original evaporator.

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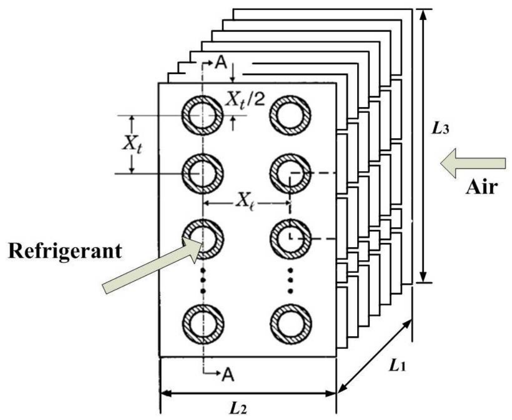
Table 1. Variables, bounds, and constraints

	Constraints	
Θ = -0.012	N = 2	$0.225 \le L_1 \le 0.235$ m
$\dot{m}$ = 0.11 kg/s	N_t = 14	$0.05 \le L_2 \le 0.06$ m
c_p = 1.0055 kJ/kg.K	N_f = 108/m	$2L_1 + 2L_2 \le 0.59$
U = 102.55	F_p = 9.24	L_3 = 0.207 m

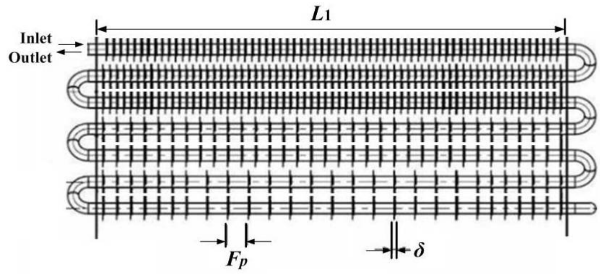
$W/m^2.K$	mm	
D_o = 8.15 mm	Pr = 0.712	Q = 400 W
δ = 0.12 mm	u_c = 2.5 m/s	

Table 2. The geometry values of the design parameters for the before optimization and the after optimization.

Parameters	**Before optimization**	**After optimization**
Tube outside diameter, D_o (mm)	8.15	8.15
Fin pitch, F_p (mm)	9.24	9.24
Fin thickness, δ (mm)	0.12	0.12
Number of row, N	2	2
Total number of tube, N_t	14	14
Number of fins per unit length in the fin pitch direction, N_f (1/m)	108	108
No-flow height, L_3 (m)	0.207	0.207
Air flow length, L_2 (m)	0.06	0.055
Refrigerant flow length, L_1 (m)	0.235	0.228



(a)



(b)

Figure 1. Evaporator geometry: (a) principal dimension of discrete plate fin-and-tube heat exchanger with inline tube arrangement; (b) fin distribution.

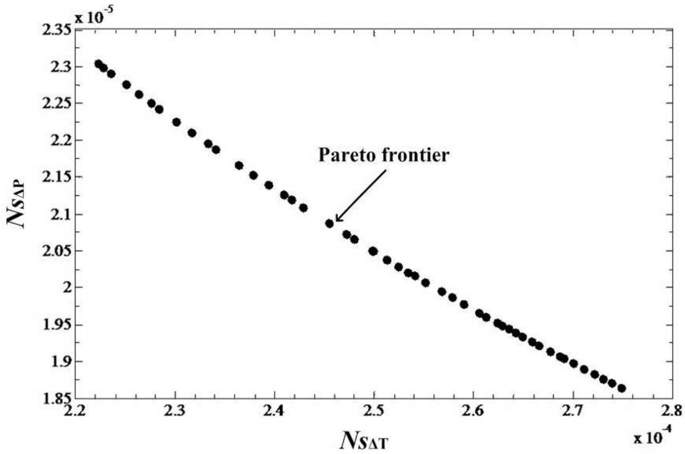


Figure 2. Pareto frontier: the best trade-off values

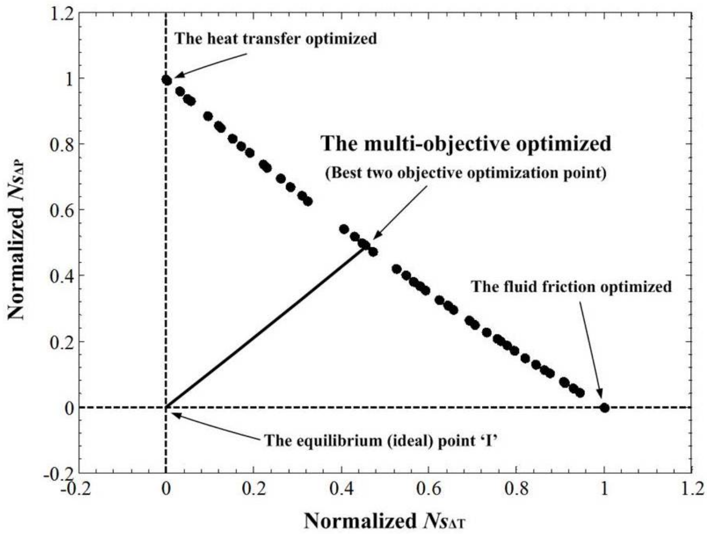


Figure 3. Normalized form of Pareto frontier and schematic of the decision making process.

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