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GA-BASED OPTIMIZATION OF COMPACT PLATE HEAT EXCHANGERS WITH MANIFOLD-MICROCHANNEL GROOVES

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ABSTRACT

Manifold-microchannel combinations used on heat transfer surfaces have shown the potential for superior heat transfer performance to pressure drop ratio when compared to chevron type corrugations for plate heat exchangers (PHE) [1-4]. However, compared with heat transfer enhancements such as intermating troughs and Chevron corrugations, manifold-microchannels (MM) have several times more variables that influence the heat transfer and pressure drop characteristics, including microchannel width, depth, passes, manifold depth, width, and manifold fin thickness. Previous work has reported on the effects of some of the variables, and provides some models for their effects on thermal and hydraulic performance. The current paper presents a genetic algorithm (GA)-based procedure to analyze the implicit effects of some of the manifold-microchannel variables, and compare the performance of manifold-microchannel plate heat exchangers to those using standard Chevron corrugations. The objective of the present work is to evaluate the performance of manifold-microchannel heat transfer enhancements and demonstrate the potential for using GA-based procedure to optimize the heat exchanger.

This paper also presents the modifications of the standard GA algorithm when applied to the optimization of MM. The resulting GA procedure is particularly well suited to PHEs for several reasons, including the fact that it does not require continuous variables or functional dependence on the design variables. In addition, the computational effort required for the GA technique in our implementation scales linearly, with a scaling coefficient that is significantly less than one, making it economical to analyze PHEs with several variables with degrees of freedom (DOF) with respect to the fitness function. The results of optimizing a manifold-microchannel plate heat exchanger are presented, and the exchanger's performance is compared to more conventional PHE of the same volume utilizing chevron corrugations. Finally, results from the empirical procedure presented in this paper for a manifold-microchannel are compared with experimental measurements in Andhare [5].

INTRODUCTION

Heat exchangers play a prominent role in aircraft thermal management systems, while the increasing requirement for effective cooling of aircraft subsystems is driving the efforts in improved performance of heat exchangers. This paper focuses on one those drivers; that is, heat transfer enhancement in the form of microchannels. An objective of the current paper is to examine the performance of MM and compare it with that of the traditional plate heat exchangers. A second objective of this paper is to present an evaluation of a GA-based optimization procedure as a means to obtain the optimum MM exchangers for a given figure of merit. Both single objective and multi-objective functions are investigated.

The first century of heat exchanger engineering design has produced innovative devices. Such devices include fins of different topologies and geometries, such as plain fins, strip fins, wavy fins, louvre fins, pin fins; corrugations on the heat surfaces, such as intermating corrugations and Chevron troughs, microchannel grooves; and fins on the outside of tubes, such as helical fins and plain fins with different geometrical profiles [6-11]. However, the past few years has seen a significant increase in the complexity of geometric features that are being utilized for enhancing the transfer of heat. Some of this is being driven by the increasing maturity of rapid manufacturing methods, such as 3D printing [12,13], which allow a heat transfer enhancement concept to be envisioned and quickly prototyped in a short amount of time. This development is challenging traditional heat exchanger modeling techniques to keep pace by providing equally quick and low-cost methods of estimating the performance of these newer and sometimes more esoteric heat transfer enhancement devices.

In many cases, the objective of the heat transfer enhancement is to maximize the amount of heat transferred per unit volume and weight of the exchanger, while at the same time minimizing the pumping power required to move the fluids through the exchanger. Particularly, for aircraft subsystems, the requirement for smaller and lighter heat exchangers has driven innovative designs. In this paper, we examine the use of manifold-microchannel technology in compact heat exchangers.

Microchannels alone have been used in heat exchangers and heat sinks for several years [16]. The extremely small passages result in significantly increased heat transfer surfaces with potential for greater overall heat transfer rate. However, because of the micro-passages, the Reynolds number of the flow through the grooves is also very low, and fully developed laminar flow ensues, limiting the thermal performance of the exchangers, while increasing the pressure drop. Manifold-microchannel combinations potentially improve on plain microchannel configurations by forcing the flow though the microchannels in short flow passages controlled by the manifold offsets, while reducing the pressure drop through the passages by providing wider flow passages through the manifolds. The manifold offsets, with proper choice of the offset lengths, ensure that the flow is in the developing regime throughout most of the passages in the exchanger.

Roughly speaking, there are three ways to obtain the performance of a heat exchanger. This includes experimental measurements, the use of computational fluid dynamics and heat transfer (CFD) to simulate the exchanger; and exact, closed-form methods that attempt to solve the governing equations for the performance of the heat exchanger. Experimental investigations [1,5, 14, 15] are invaluable for heat exchanger studies, but they are expensive as the exchanger has to be manufactured first. There are also scaling issues. Analytical procedures are also challenging because of the complexity of heat exchangers. There have been many useful analytical results for compact heat exchangers [24], but they are generally of limited capabilities, as drastic simplifying assumptions have to be made in order to make the performance equations solvable by hand. Manifold-microchannel arrangements are geometrically too complicated for analytical solutions. Consequently, this approach is not feasible for the problem of our current interest. The CFD-based approach is expensive; and the complexity of many heat exchangers, with a large number of passages – coupled with the need to resolve boundary layers - makes it impractical to perform a complete CFD analysis on realistic heat exchangers in a timely manner without drastic, accuracy-destroying simplifications in the CFD procedure. CFD analysis of small sections of a heat exchanger can sometimes serve as a way to fundamental understanding. However, this has to be augmented with empirical methods in order to be applicable to the performance analysis, sizing, and optimization of complete exchangers [16-18]. We will be utilizing this approach with previously-published CFD results for a section of the manifold-microchannel heat exchanger and extending the model to analyze a complete heat exchanger via empirical means. For an introduction to CFD application in the HVAC&R industry, consult Ladeinde and Nearon [26].

Several researchers have reported on the performance of manifold-microchannel technology [1-5, 14-18]. This includes experimental investigations [1,5, 14, 15], and numerical modeling work [16-18]. The numerical work by Arie et. al.[18] involves CFD-based modeling and optimizing of a PHE with manifold-microchannel heat transfer enhancements over a range of variables. The present work avoids the computational expense of the CFD procedure, thereby enabling the development of a computational tool that is suitable for fast, real-time analysis and optimization of complete heat exchanger systems utilizing MM. As alluded to above, to obtain the data required to close the empirical thermal and hydraulic performance models of the exchangers, we draw upon the numerical work in the previously mentioned paper [18].

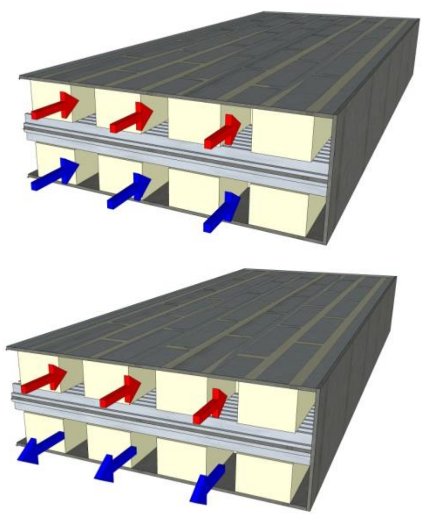
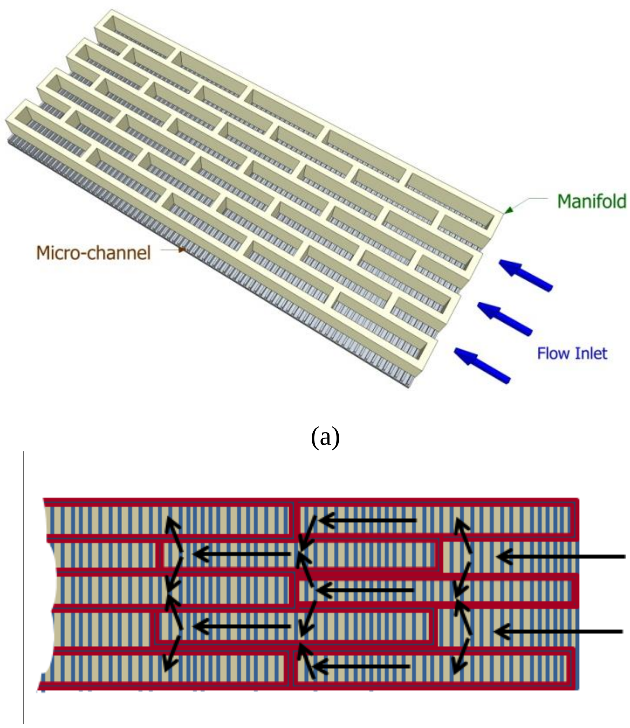


Fig 1 Sections of a manifold-microchannel heat transfer enhancement between heat exchanger plates

Regarding optimization, which is a significant contribution of this paper, CFD approaches this task via the adjoint method. However, the method is simply infeasible for the problem at hand for several reasons, including the computational cost, which is orders of magnitude higher than that of the proposed GA approach, and the inability of adjoint methods to handle discontinuous functions, such as that in Eqn. (4) below.

MODELING

Figure 1 represents the model of the manifold-microchannel combination that is the focus of the current work. The microchannels are arranged perpendicular to the flow direction, while the manifolds are arranged parallel to the flow direction within the plates. The manifolds have relatively large free flow passages compared to the microchannels, potentially minimizing the pressure drop through the exchanger.



(a)

(b)

Fig 2 Manifold-microchannel arrangement: (a) 3-D cut-away view, (b) top view.

The manifolds also run in short sections and are then offset such that the flow is obstructed and forced to go down through the microchannels and then emerge in offset passages of the manifold as it continues through the plates, as shown in Figure 2(a). Figure 2(b) shows a top view of the manifold-microchannel arrangement of Figure 2(a).

Table 1 List of manifold-microchannel variables

Variable	Description
h_{mnd}	Manifold height
h_{ch}	Channel height
W_{mnd}	Manifold width
W_{ch}	Channel width
t_{mnd}	Manifold thickness
t_{ch}	Channel thickness
L_{mnd}	Length of manifold between offsets

The geometric variables that characterize the manifold-microchannel heat transfer enhancement are listed in Table 1, some of the variables are shown in Figure 3.

The relationship between a manifold-microchannel variable and an equivalent Chevron variable of the same volume, with *h* the corrugation depth, is

$$2h = h_{mnd} + h_{ch} . \tag{1}$$

The hydraulic diameter of the flow through the manifold-microchannel is defined as in Arie et al. [18]:

$$D_h = \frac{2h\left(L_{mnd} + W_{mnd)}\right)}{2h_{mnd} + L_{mnd} + W_{mnd)}} , \tag{2}$$

where $2h$ is the plate separation.

The free flow area per plate is computed as

$$S = h_{mnd} W_{mnd} \frac{W}{2\left(W_{mnd} + t_{mnd}\right)} , \tag{3}$$

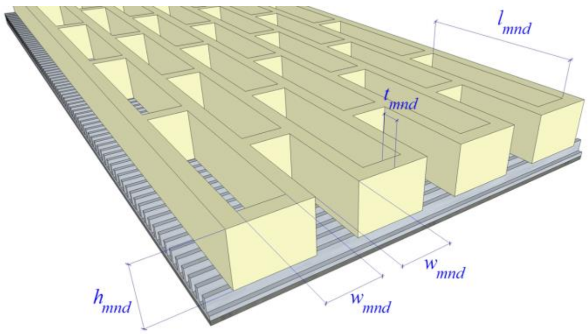
where W is the passage width, $h_{mnd}W_{mnd}$ is the flow area of a single manifold groove within the plate, and the balance of the expression in (3) is the number of such manifold grooves per plate passage. Note that alternate manifold grooves are blocked from the flow as shown in Figure 2; hence the factor 2 in the above equation. Note also that for the same passage height or overall heat exchanger volume, the manifold-microchannel configurations could have significantly smaller free flow area, depending on the microchannel height, and thus higher Reynolds numbers compared to analogous Chevron configurations.

The Colburn and friction factors characterizing the manifold-microchannel flow have been obtained during the course of the present research by discretizing the CFD results in [18]. The resulting data fit is as follows:

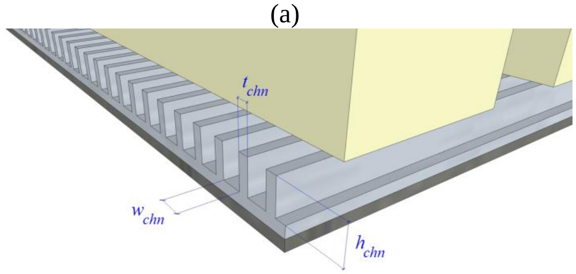
$$\begin{cases} \mathrm{Re} > \mathrm{Re}_{crit} & \begin{aligned} j &= 2.106 - 5.5326x10^{-3}\,\mathrm{Re} + 5.64x10^{-6}\,\mathrm{Re}^2 - \\ &\quad 1.933x10^{-9}\,\mathrm{Re}^3 \\ f &= 0.0905 + 1.64167x10^{-3}\,\mathrm{Re} - 2.0x10^{-6}\,\mathrm{Re}^2 + \\ &\quad 8.33x10^{-10}\,\mathrm{Re}^3 \end{aligned} \\ \mathrm{Re} \le \mathrm{Re}_{crit} & \begin{aligned} j &= 1.1476 - 1.15576x10^{-2}\,\mathrm{Re} + 5.78583x10^{-5}\,\mathrm{Re}^2 - \\ &\quad 1.3744x10^{-7}\,\mathrm{Re}^3 + 1.57167x10^{-10}\,\mathrm{Re}^4 - \\ &\quad 6.9833x10^{-14}\,\mathrm{Re}^5 \\ f &= 2.605 - 3.0977x10^{-2}\,\mathrm{Re} + 1.5964x10^{-4}\,\mathrm{Re}^2 - \\ &\quad 3.9979x10^{-7}\,\mathrm{Re}^3 + 4.854167x10^{-10}\,\mathrm{Re}^4 - \\ &\quad 2.29167x10^{-13}\,\mathrm{Re}^5 \end{aligned} \end{cases} \tag{4}$$

where Re_{crit} = 670.

The model equations for the Chevron corrugations that are used for the comparisons in the current work are based on those in the INSTED Thermal Analysis Software, as presented in Ladeinde and Alabi [19, 25].



(a)



(b)

Figure 3 Manifold-microchannel geometric variables (a) Manifold variables, (b) Microchannel variables

GA-BASED OPTIMIZATION PROCEDURE

The heat exchanger optimization problem may be posed as follows:

Let a heat exchanger system modeled by the equations

$$H(\vec{X}) = 0 \quad , \tag{5}$$

have N_{var} design variables

$$\vec{X} = [X_1, X_2, X_3, ..., X_{N\,\text{var}}] \equiv X_i, \tag{6}$$

where X_i may include a mix of binary, integer, and real numbers.

The variables are assumed to be bounded as follows:

$$X_i = \{x : x \in [X_{i,\min}, X_{i,\max}]\}. \tag{7}$$

The model is further assumed to be subject to some constraints

$$G(\vec{X}) \leq 0, \tag{8}$$

where the constraint equation also includes the bounds on the variables, or

$$\begin{bmatrix} X_{i,\min} - X_i \leq 0 \\ X_i - X_{i,\max} \leq 0 \end{bmatrix}. \tag{9}$$

The space of possible solutions is an *Nvar* dimensional space, with the optimum being a point in the space

$$\vec{X}_{opt} = [X_{1,a}, X_{2,b}, X_{3,c}, ..., X_{N\,\text{var},kNVar}], \tag{10}$$

and the optimum value for each variable lying within the bounds

$$X_{i,k} = [X_{i,\min}, X_{i,\max}]. \tag{11}$$

The genetic algorithm utilizes a set of operators to derive an optimum value starting from an initial set consisting of several randomly generated configurations of the model $H(\vec{X})$.

Our GA operators were developed on the following principles (the standard GA procedures can be found in [20-23]):

a) Offsprings of good parents are likely to be better individuals. Thus, we expect that results derived from the best configurations are even better results.

b) Mutations ensure that various traits in nature are explored such that good characteristics that do not exist in the current population can be discovered. In our procedure, this translates into the likelihood of finding the optima regardless of the starting configuration. In other words, the procedure is not very susceptible to local optima.

c) Continuous applications of the GA principles successively over several generations would drive the entire population towards a better one as measured by the fitness function. In our procedure, we apply the GA operators to the entire population, replacing the original population or a portion of it with the offsprings in each generation. Based on a convergence criterion whereby the overall combined fitness function of the entire population is no longer changing significantly, we expect the best individuals from the population to be optimal.

The property of GAs where the operators are abstracted from the models themselves, and the resilience to getting stuck in local optima is attractive for heat exchanger problems where the models contain several integer variables, logical operators, and models with discontinuous, piecewise-continuous, or step functions such as those in Eqn (4).

The starting configurations make up what is termed the population, and each individual configuration has a property that measures their viability or fitness based on the objective function. The fitness values are computed from Eqns (5) through (9). Each individual is encoded with their property, whereby Eqn (6) becomes:

$$\vec{X} \equiv [H_{mnd}, H_{chn}, W_{mnd}, W_{chn}, L_{mnd}]. \quad (12)$$

Weight, coefficient of performance, COP, and combined weight and COP, are selected as the fitness functions for the current work.

The GA operators are Selection, Crossover, and Random Mutation. The process of selection includes a sorting of all individuals in the population set, and a random operator weighted to select individuals closer to the top of the sorted set. The individuals are sorted based on the value of the fitness function. The selection operation results in two selected individuals from the population set.

The crossover operator performs a weighted average of the variable values from each selected pair (“parents”).

$$\vec{X}_{child} = \gamma \vec{X}_{parent1} + (1-\gamma)\vec{X}_{parent2} \quad (13)$$

where γ is set to 0.5 in this work.

The mutation operator introduces randomness into the crossover such that a deviation from the combination of the values of the two-parent individuals is applied to a randomly selected variable in (12). The deviation is also computed via a random function and weighted such that the resulting combined variable lies within the bounds in (7).

The variables of the GA process include the population size, number of individuals to retain in each generation, and the frequency of mutation operations. For the current work, the population size was set to 4000, half of the population is replaced during each generation, and the frequency of mutation operations is set to 1 in 100 crossover operations.

The bounds in Table 2 were set for the variables, similar to the values in Arie et al. [18], who used a different kind of optimization procedure.

Table 2 Manifold-Microchannel Variable Bounds

Variable	Minumum (μm)	Maximum (μm)
h_{mnd}	1000	9600
h_{ch}	100	2600
W_{mnd}	300	2600
W_{ch}	20	103
t_{mnd} (fixed)	400	400
t_{ch} (fixed)	50	50
L_{mnd}	3200	36000

The manifold-microchannel heat exchanger is optimized and compared in performance to Chevron corrugations in a similar PHE of the same volume. The other variables of the heat exchanger are as follows:

Plate height and width are, respectively, 1m and 0.25m; with 101 plates providing for 50 hot flow passages and 50 cold flow passages. The working fluid in both streams is water. The hot water enters the exchanger at 388.16 K and 2kg/s, and is cooled by a water stream at an ambient temperature of 288.16 K. We carried out comparisons with Chevron PHE at cold water flow rates of 0.2, 0.4, 0.6, 0.8, and 1kg/s; and Chevron corrugation angles of 30^o, 45^o, and 60^o. The plates, corrugations, fins, and manifold were all assumed to be made of copper. The overall exchanger arrangement was assumed to be single pass for both streams, although the procedure has been developed to analyze any number of consistent passes in the streams.

RESULTS

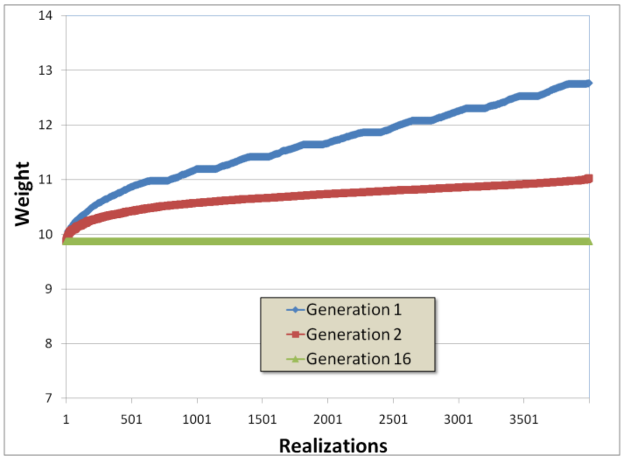
The results of optimizing the PHE with manifold-microchannel between the plates are presented in Table 3. Three fitness functions were considered: weight, COP, and multiobjecive fitness function combining weight and COP. Each optimization calculation took about 5 minutes on an Intel Pentium PC and utilized a population size of 4,000 realizations. A maximum of 16 generations has been used.

Figure 4 illustrates the convergence of the population as the GA generations proceed, by comparing the fitness function from the first generation, following the initialization step, to the last generation. The successful convergence of the population is qualitatively evident in the figure where the weight of the entire population can be observed to have converged around a value of 9.8665.

Table 3 Manifold-microchannel optimal configuration

	Optimal (μm)		
Variable	**Weight**	**COP**	**Weight/COP**
h_{mnd}	7886.9	8264.4	8269.1
h_{ch}	1784.7	2525.3	2067.2
W_{mnd}	2231.8	2572.9	2550.8
W_{ch}	85.5	102.91	102.91
t_{mnd} (fixed)	400	400	400
t_{ch} (fixed)	50	50	50
L_{mnd}	24228.5	24936.7	30145.8
Weight	9.8664	10.9765	9.8691
COP	1.9722	2.6275	2.6272

Figure 4b shows that while the last generation has a population that has converged (to 9.8665), the population is not monolithic as it appears in Figure 4a when plotted on the same scale as the less homogenous population the procedure commenced with in the first generation.



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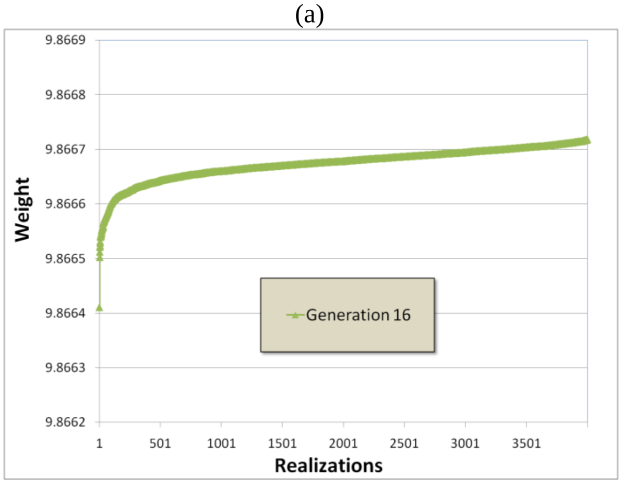

(b)

Figure 4 Convergence of the GA procedure illustrated with weight as the fitness function

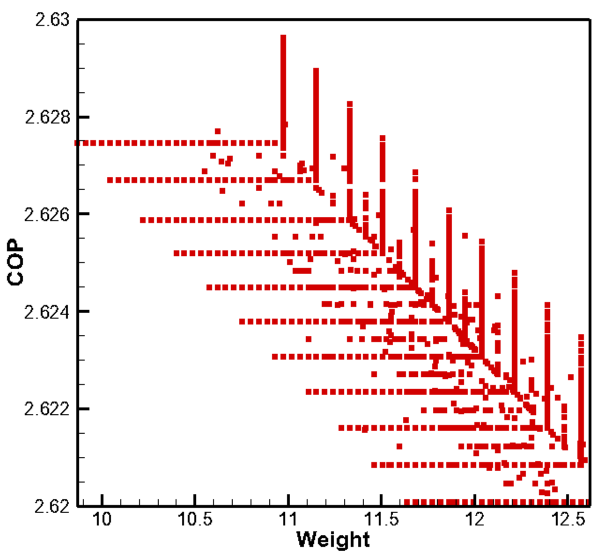

Figure 5. Some of the realizations in the vicinity of the optimal value

Figure 5 shows some of the heat exchanger realizations in the population plotted based on their weight-COP values. An advantage of the GA is that it provides an engineer with several viable options to choose from based on other factors that may be of interest besides the figures of merit. Figure 5 shows the options in the present case.

The optimal configuration was compared in performance to Chevron corrugations having the same passage height. The results are shown in Figures 6 and 7, where the Colburn and friction factors are compared.

The results show similar trend as in [18] in which the increase in pressure drop in the manifold-microchannel is smaller than in the Chevron configuration having the biggest angle, while the heat transfer from the manifold-microchannel matched or exceeded that of the same Chevron configuration. Note that flow rate is used for comparison rather than the Reynolds number because the Reynolds number inside the exchanger for the same flow rate differs between the manifold-microchannel and the Chevron configurations. This is due to the differences in the hydraulic diameter and free flow area for MM and the standard Chevron configurations.

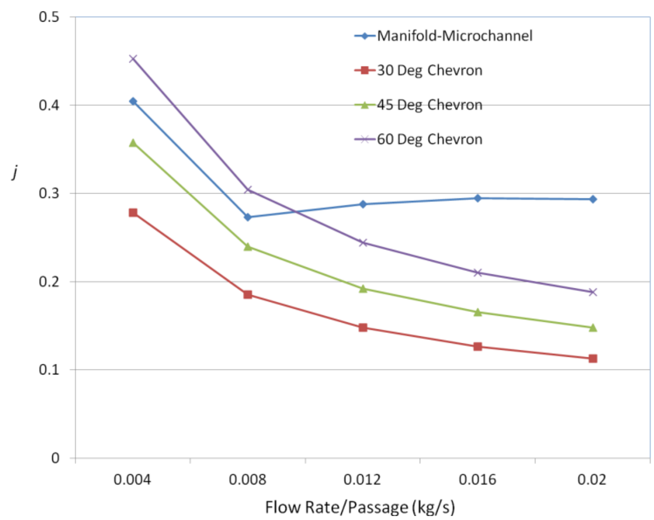

Figure 6 Comparison of Colburn factor between MM and Chevron corrugations

The manifold-microchannel calculations are also compared with experimental measurements using the dimensions in Andhare [5]. A comparison of the heat transfer per unit volume and temperature drop is presented in Figure 8, and the coefficient of pressure per unit temperature drop in Figure 9.

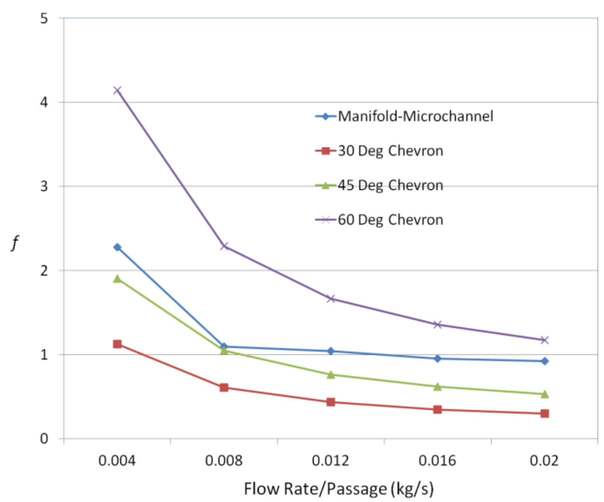

Figure 7 Comparison of friction factor between MM and Chevron Corrugations

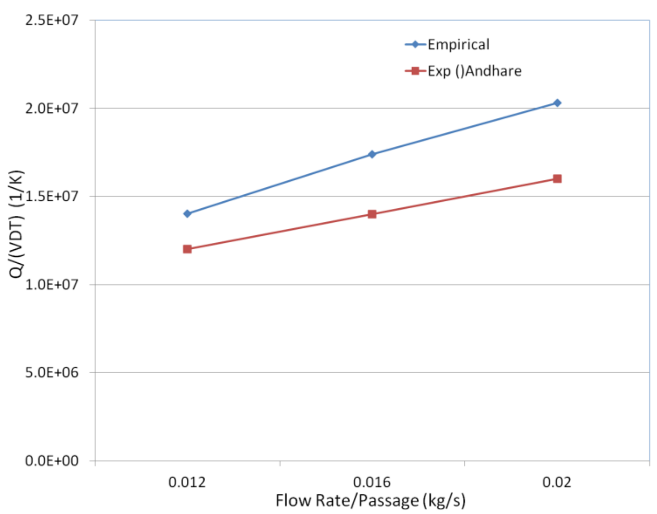

Figure 8 Comparison of heat transfer per unit volume and temperature drop for the manifold-microchannel calculations between current procedure and experiments

The results are between 15-20% of the experimental values for the heat transfer measurements, and between 2-30% for the coefficient of pressure measurements. In both cases, the overall trend of the experimental measurement is captured. We believe that much of the variations is due to the fact that the empirical model was obtained from a single point manifold-microchannel configuration. Even though the model was derived over a Reynolds number range between 100 and 1,000, we expect that the performance of MM will also be influenced by other variables, including the manifold height and width, and the microchannel height and width – variables that are not included in the current model. Note, however, that those variables still do implicitly influence the current calculations through the equivalent diameter and free flow area computations, Eqns (2) and (3). However, it is expected that both the Colburn and friction factors will be dependent to some extent on those other geometric variables. Future work utilizing more experimental measurements or numerical simulations, or a combination of these, could result in models that include the effects of those additional geometric variables and should be expected to perform better than the discretized j and f data in Eqn. (4).

However, the current work does demonstrate the viability of empirical model based performance analysis and optimization of manifold-microchannel heat transfer enhancements between plates, provided good models of the performance characteristics of the manifold-microchannels can be obtained. This procedure has potential for application even for other types of heat transfer enhancements within the heat transfer plates, including straight microchannels. We plan to extend the current work to other types of heat transfer enhancements as well as improving on the current empirical models by utilizing experimental and more numerical simulation results to improve the Colburn and friction factor models in MM.

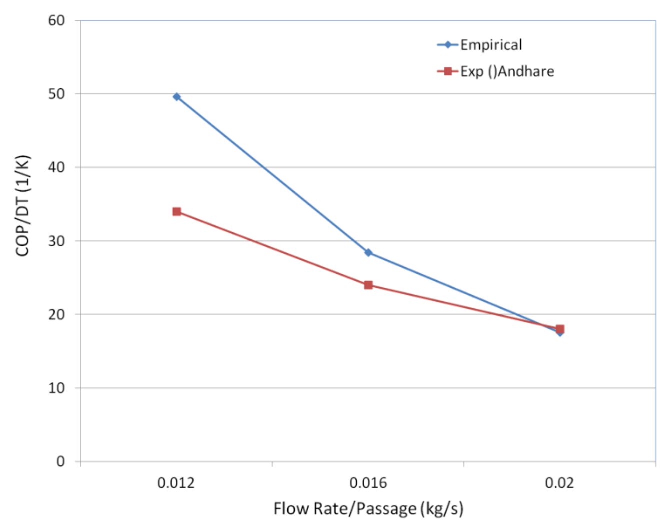

Figure 9 Comparison of coefficient of pressure per unit temperature drop for the Manifold-microchannel calculations between current procedure and experiments

CONCLUSIONS

This paper presents the results of analyzing manifold-microchannel PHEs using empirical models. The j and f in the empirical models were fitted from CFD simulation results presented in Arie et. al.[18]. The empirical formulation provides estimates for the thermal and hydraulic characteristics of the manifold-microchannel through the Colburn and friction factors as functions of the local Reynolds number inside the exchanger. The procedure incorporates the effects of manifold and microchannel geometries into the hydraulic diameter and free flow area such that the Reynolds number within the exchanger is a function of those variables. Utilizing a GA-based procedure, a framework for optimizing manifold-microchannel heat exchangers has been developed. Comparisons between the optimized manifold-microchannel configuration and configurations utilizing standard Chevron corrugations were also presented. Similar, to the work of Arie et al. [18] who used a different approach, the results in this paper show that the manifold-microchannel unit performs better than Chevron corrugations in PHE of the same volume, without a proportional increase in pressure drop.

Fairly good agreement between calculations using the current procedure and experimental measurements of Andhare [5] is also reported in this paper. The maximum deviation in the pressure drop is 30%. We believe much of the differences are attributable to the fact that the j and f data were derived from a single manifold-microchannel configuration as a function of Reynolds number. The thermal and hydraulic performance of the manifold-microchannel unit is likely to be additionally improved by including the unit's other variables in the curve fitting. This statement underscores the challenges in relying on CFD as the basis for reduced-order modeling.

The current work does demonstrate the viability of performance analysis and optimization of manifold-microchannel heat transfer enhancements between plates using empirical models. The procedure reported here could also be similarly applied to other types of heat transfer enhancements within PHEs, including straight microchannels and other kinds of emerging heat transfer enhancements that have been introduced in the past few years – such as 3D-printed heat exchangers. Future work is also planned to improve the current empirical model by using more experimental and numerical results to determine the Colburn and friction factors for the MM configurations. The success of the procedure in this paper obviously depends on the fidelity of the original CFD or experimental data used to generate the j/f data.

NOMENCLATURE

D_h	hydraulic diameter
f	friction factor
h	corrugation height
h_{ch}	channel height
h_{mnd}	manifold height
j	Colburn factor
H	plate height
L_{mnd}	length of manifold between offsets
S	free flow area
W_{ch}	channel width
W_{mnd}	manifold width
W	plate width
N_p	number of passages
t_{ch}	channel thickness
t_{mnd}	manifold thickness

Greek symbols

γ	cross over combination factor

Subscripts

child	offspring configurations of a GA scheme
ch	microchannel
crit	critical
fin	fin
mnd	manifold
opt	optimal
parent	parent configurations of a GA scheme

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