Robust Optimisation of Serpentine Fluidic Heat Sinks for High-Density Electronics Cooling

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Abstract

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Deterministic and probabilistic optimisation have been conducted to explore the optimum design for the microchannel heat sink equipped with chevron fins. Surrogate-based optimisation strategy has been employed to reduce the demand for the computational resources in this study. The main channel width (W_{ch}), the secondary channel length (l_{sc}) and the oblique angle of the fins (θ) were selected as design variables. Monte Carlo simulation was used to propagate the uncertainty associated with the design variables to the outputs of interest, i.e.

thermal resistance (R_{th}) and pressure drop penalty (ΔP). The results showed that the optimum designs produced using probabilistic optimisation have R_{th} and ΔP higher than that obtained using the deterministic optimisation process by 8% and 4.8%, respectively.~~The goal of fluidic heat sinks for electronics cooling is to provide effective and energy-efficient cooling which ensures that the processors are below critical temperatures with minimal power input (Ahmed et al. in Int J Heat Mass Transf 118:129–153, 2018 [1]).~~

1. Introduction

The goal of fluidic heat sinks for electronics cooling is to provide effective and energy-efficient cooling which ensures that the processors are below critical temperatures with minimal power input [1]. Although air-cooled heat sinks currently dominate the market, a number of recent studies have concluded that increasing densities of integrated circuits (up to 10 kW/cm^2 by 2020 [2]) will require effective liquid-cooling heat sink technologies. These trends have stimulated much recent interest in single-phase flows in fluidic channel devices for cooling high heat flux electronics encountered in, e.g. aircraft and in RF and microwave applications [3]. Single-phase flow in serpentine channel heat sinks is particularly well-suited to providing uniform processor temperatures for high-density electronics cooling applications. Al-Neama and his co-researchers [4] demonstrated recently that combining serpentine channels with fin structures can provide very good temperature uniformity while reducing significantly the pressure drop associated with serpentine systems. Several research groups are working to optimise the performance of microchannel heat sinks; however, most of these optimisation studies are deterministic and do not take into account the uncertainties associated with manufacturing processes and operating conditions [1]. Uncertainty could be classified as aleatory uncertainty and epistemic uncertainty. The former refers to inherent randomness in system behaviour, like randomness in operating conditions and geometric parameters, which is irreducible unless it is taken ~~took~~ into consideration through the design stage, while the latter is a result of limited data and information about the system, such as those due to lack of knowledge about a model structure, which could be reduced by gathering more information about the system [5]. This study is the first to consider the effect of robustness considerations on the design and optimisation of the liquid-cooled serpentine heat sinks with chevron fin structures. It provides a contrast between the results of deterministic and robust optimisation of the liquid-cooled serpentine fluidic heat sinks with chevron fin structures introduced by Al-Neama et al. [4].

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2. Problem Description and Methodology

A 3D geometrical model of the serpentine microchannel heat sink (SMCHS), investigated by Al-Neama et al. [4], is shown in Fig. 1. It consists of 12 main channels with 10 secondary channels dividing the walls between the main channels. These secondary channels have a chevron shape. All the dimensions are depicted in the figure below. Two heaters are attached at the base of this heat sink to mimic the heat generated by the chip processors of the electronic systems. The substrate of the heat sink is manufactured from ~~the~~ copper, and water is used as a coolant. COMSOL V5.3a multiphysics software has been used to solve the fluid flow heat transfer governing Eqs. (1–4), using finite element methods, and obtained the numerical solution for this conjugate heat transfer problem.

$$\nabla.\boldsymbol{u}=0 \tag{1}$$

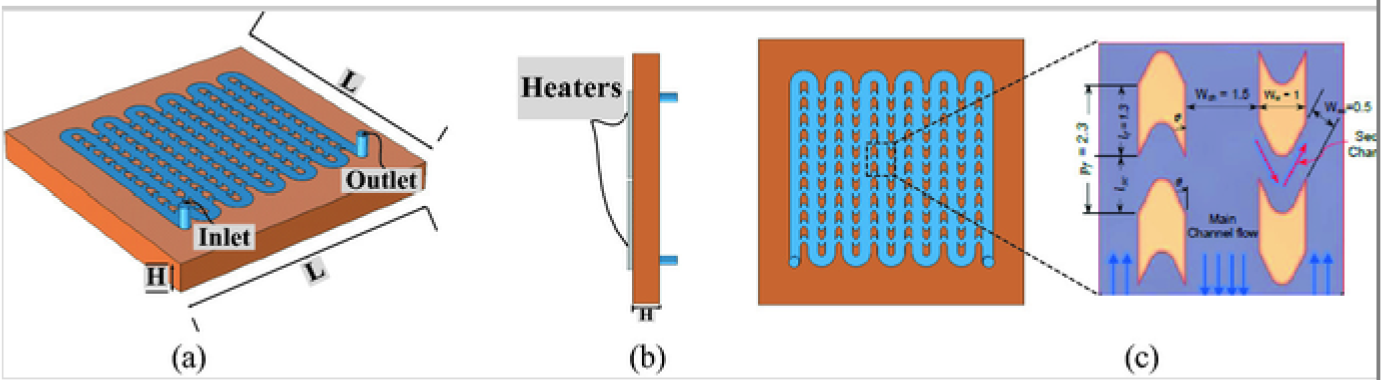
$$\rho\left(\frac{\partial u}{\partial t}+u.\nabla u\right)=-\nabla p+\nabla.\left(\mu\left(\nabla u+(\nabla u)^{\mathrm{T}}\right)-\frac{2}{3}\mu\left(\nabla.u\right)I\right)+F \tag{2}$$

$$\rho C_p\left(\frac{\partial T}{\partial t}+u.\nabla T\right)=\nabla.\left(k\nabla T\right)+Q \tag{3}$$

$$-k_s\frac{\partial T_s}{\partial n}\bigg|_{\text{interface}}=-k_f\frac{\partial T_f}{\partial n}\bigg|_{\text{interface}} \tag{4}$$

Fig. 1

Geometrical model: **a** 3D geometry, **b** side view and **c** top view with enlarged details, Al-Neama [4]



Here, **u**, ρ, p, μ, I, C_p and T are velocity vector [m/s], density [kg/m^3], pressure [Pa], dynamic viscosity [Pa s], turbulent intensity [1], specific heat [J/(kg K)]and absolute temperature of the fluid [K], respectively. t, F and Q are the time [sec], external force applied on the fluid [N] and the heat flux~~source~~ [W/m^3], respectively.

The boundary conditions associated with this problem are as follows: the inlet temperature was taken as 20 °C and the outlet pressure has been set to be ambient pressure. A ~~H~~heat flux is applied at a part of the bottom surface of the heat sink could take values in the range 25–100 W/cm^2. A no-slip boundary condition was applied at all ~~the ~~walls in contact with the fluid. Further, a number of assumptions have been made for simplifying the CFD simulation which are: (1) the flow is steady and single-phase, (2) the fluid is incompressible, (3) viscous dissipation is neglected, (4) radiative heat transfer is neglected, (6) all the outer surfaces exposed to the surroundings are insulated and (6) the channel surfaces are smooth. The results are validated against a previous study, and a good agreement between the present work and those in the literature has been produced.

Deterministic and probabilistic optimisation for the performance of the SMCHS with chevron is carried out. Surrogate-based optimisation is used, i.e. the strategy of optimisation is based on replacing the costly CFD evaluation with a cheaper mathematical model to implement the optimisation process on the microchannel heat sink. This requires creating a sample of the design of experiment points (DoE) in the design space, run the CFD model at these points to generate the build points that are used to construct the surrogate model and finally performing the optimisation on the objective functions, thermal resistance and pressure drop of the SMCHS of the present investigation. Three design variables, namely the width of the main channel (1.0 $mm \le W_{ch} \le 2.0$ ~~in mm~~mm), the length of the secondary channel (0.75 $mm \le l_{sc} \le 1.25$~~, in mm~~mm) and the oblique angle of the fins ($20° \le \theta \le 45°$~~, in degree~~), have been considered.

The open-source DAKOTA toolkit has been utilised to implement this study. It is a powerful toolkit as it has algorithms for optimisation, uncertainty quantification, parameter estimation and sensitivity analysis. Latin hypercube sampling (LHS) is used, and Gaussian processes are employed to build the surrogate meta-models. A multi-objective function deterministic optimisation, using Eqs. (5) and (9–11), is conducted first, to obtain a Pareto front of non-dominated solutions. After that, a probabilistic optimisation, using Eqs. (6–11), is performed to explore the effect of the uncertainties in the input variables of the serpentine microchannel heat sink with chevron fin on its performance criteria. This will be done by propagating the uncertainty in the design variables, which is set to be (±0.025°) for θ angle and (±0.025 ~~mm~~mm) for W_{ch} and l_{sc}, into the quantities of interest, i.e. the thermal resistance and pressure drop, and this can be accomplished using Monte Carlo simulation and normalising the mean and standard deviation of each objective function with respect to its maximum value to produce Eq~~s~~. (6) ~~and (7)~~. Further, a weighting factor (wi) will be introduced to examine the relative influence of the mean and the standard deviation on the performance.

$$\begin{gathered}\textbf{Deterministic Optimisation}:\\ \textbf{Minimise}\left\{R_{\mathrm{th}}\left(W_{\mathrm{ch}},l_{\mathrm{sc}},\theta\right)\ \text{and}\ \Delta P\left(W_{\mathrm{ch}},l_{\mathrm{sc}},\theta\right)\right\}\end{gathered} \tag{5}$$

$$\begin{gathered}\textbf{Normalised}\left(R_{\mathrm{th}}\right):\\ \textbf{OF\_Rth}=wi\times N\mu_{R_{\mathrm{th}}}+(1-wi)\times N\sigma_{R_{\mathrm{th}}}\\ \textbf{Normalised}\left(\Delta P\right):\\ \textbf{OF\_}\boldsymbol{\Delta}\textbf{P}=wi\times N\mu_{\Delta P}+(1-wi)\times N\sigma_{\Delta P}\end{gathered} \tag{6}$$

$$\begin{gathered}\textbf{Probabilistic Single Objective Optimisation for the Normalised}\left(R_{\mathrm{th}}\right):\\ \textbf{minimise}\left\{\textbf{OF\_Rth}\left(W_{\mathrm{ch}},l_{\mathrm{sc}},\theta\right)\right\}\end{gathered} \tag{7}$$

$$\begin{gathered}\textbf{Probabilistic Single Objective Optimisation for the Normalised}\left(\Delta P\right):\\ \textbf{minimise}\left\{\textbf{OF\_}\boldsymbol{\Delta}\textbf{P}\left(W_{\mathrm{ch}},l_{\mathrm{sc}},\theta\right)\right\}\end{gathered} \tag{8}$$

9

$$\textbf{Subjected to :}$$
$$1.0\ mm \le W_{\mathrm{ch}} \le 2.0\ mm$$

$$0.75\ mm \le l_{\mathrm{sc}} \le 1.25\ mm \tag{10}$$

$$20^\circ \le \theta \le 45^\circ \tag{11}$$

3. Results

The results of the multi-objective function deterministic optimisation along with CFD validation points are illustrated in Fig. 2. The validation points are in good agreement with the selected points with a ~~as the~~ maximum difference between the values does not exceed 4%. Figure 3 shows the geometries for two cases cho~~o~~sen for a comparison between an optimal design obtained by multi-objective optimisation of thermal resistance and pressure drop, represented by point A in Fig. 2, and a non-optimal design from set of DoE points, represented by point B in Fig. 2, working with the same boundary conditions. It can be noticed the remarkable difference in the shape and size of the microchannel passages between the two designs which affects the performance of the microchannel heat sink positively where the optimum design has reduced the thermal resistance, heat sink temperature and the pressure drop of the fluid in comparison with the non-optimum design. These differences have ~~are~~ been illustrated in Fig. 4 and Table 1. Therefore, it could be concluded that the hydrothermal characteristics for optimum designs are all better than those of the non-optimum design. This could be attributed to the fact that the optimum design has a larger microchannel passage volume, wider channel width and a good aerodynamic fin shape in comparison with the non-optimum design. All of these factors enhanced heat transfer and reduced the pressure drop.

Fig. 2

Deterministic optimisation of the serpentine MCHS with chevron fins

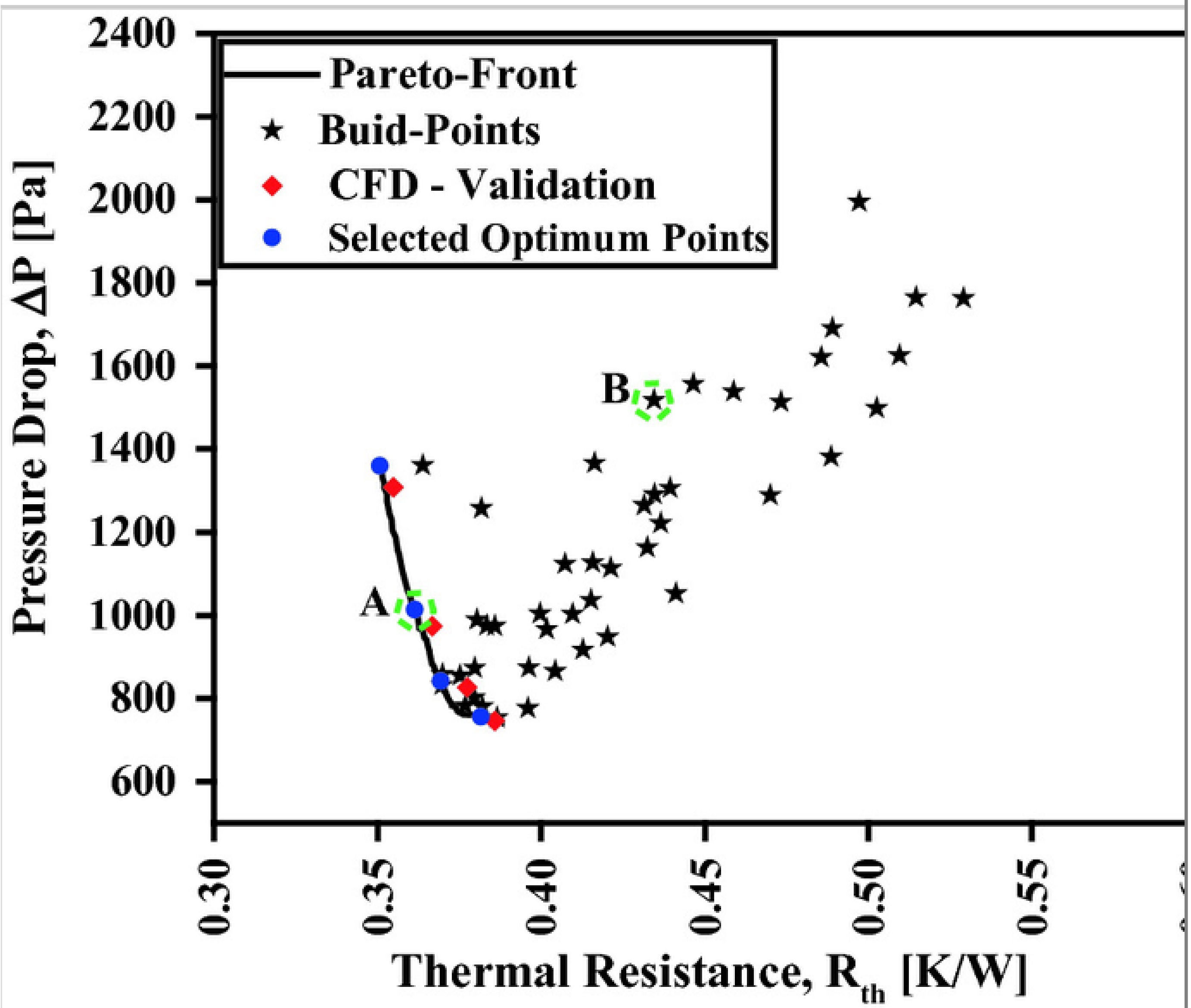


Fig. 3

Geometry for: **a** optimum design, point A in Fig. 2 and **b** non-optimum design, point B in Fig. 2

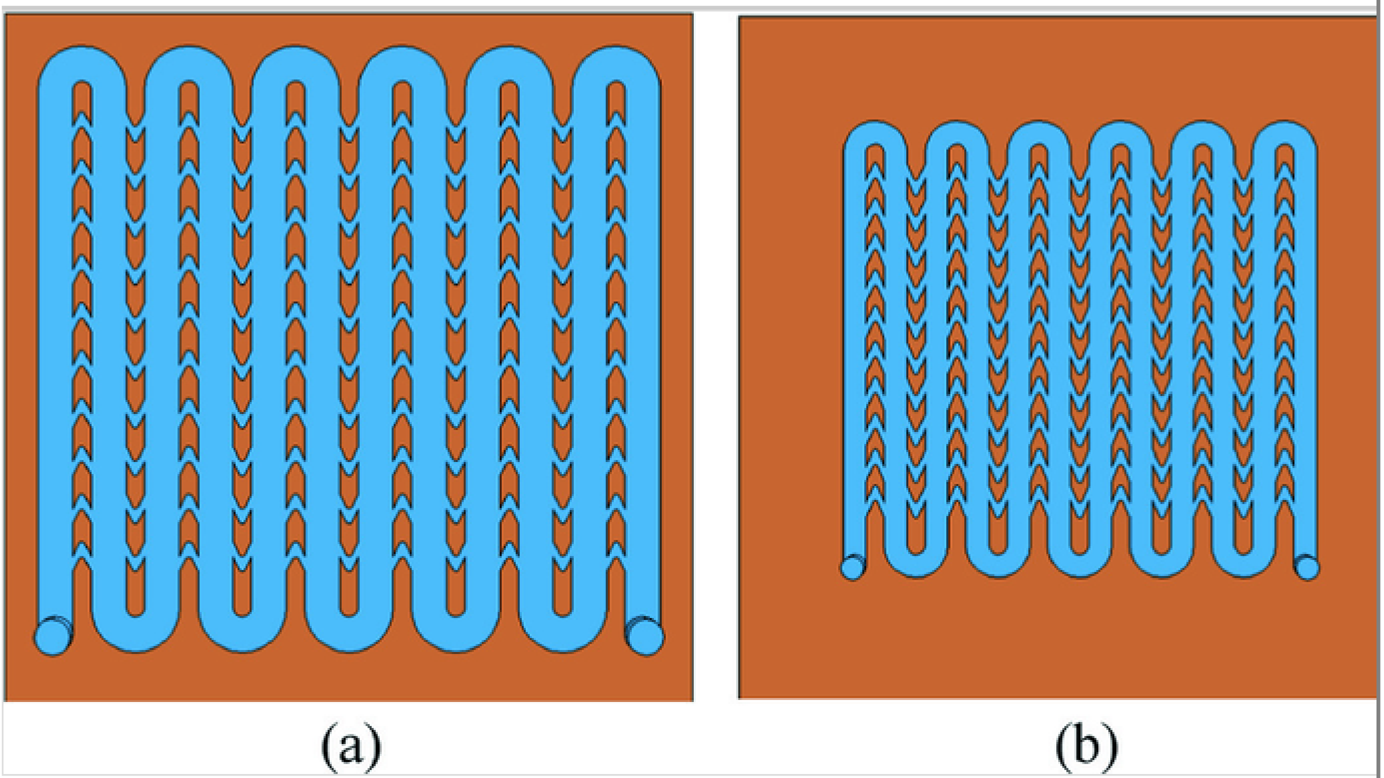


Fig. 4

a and **b** temperature and pressure disterbuitions for optimum design [Point A] and **c** and **d** for non-optimum design [Point B], respectively

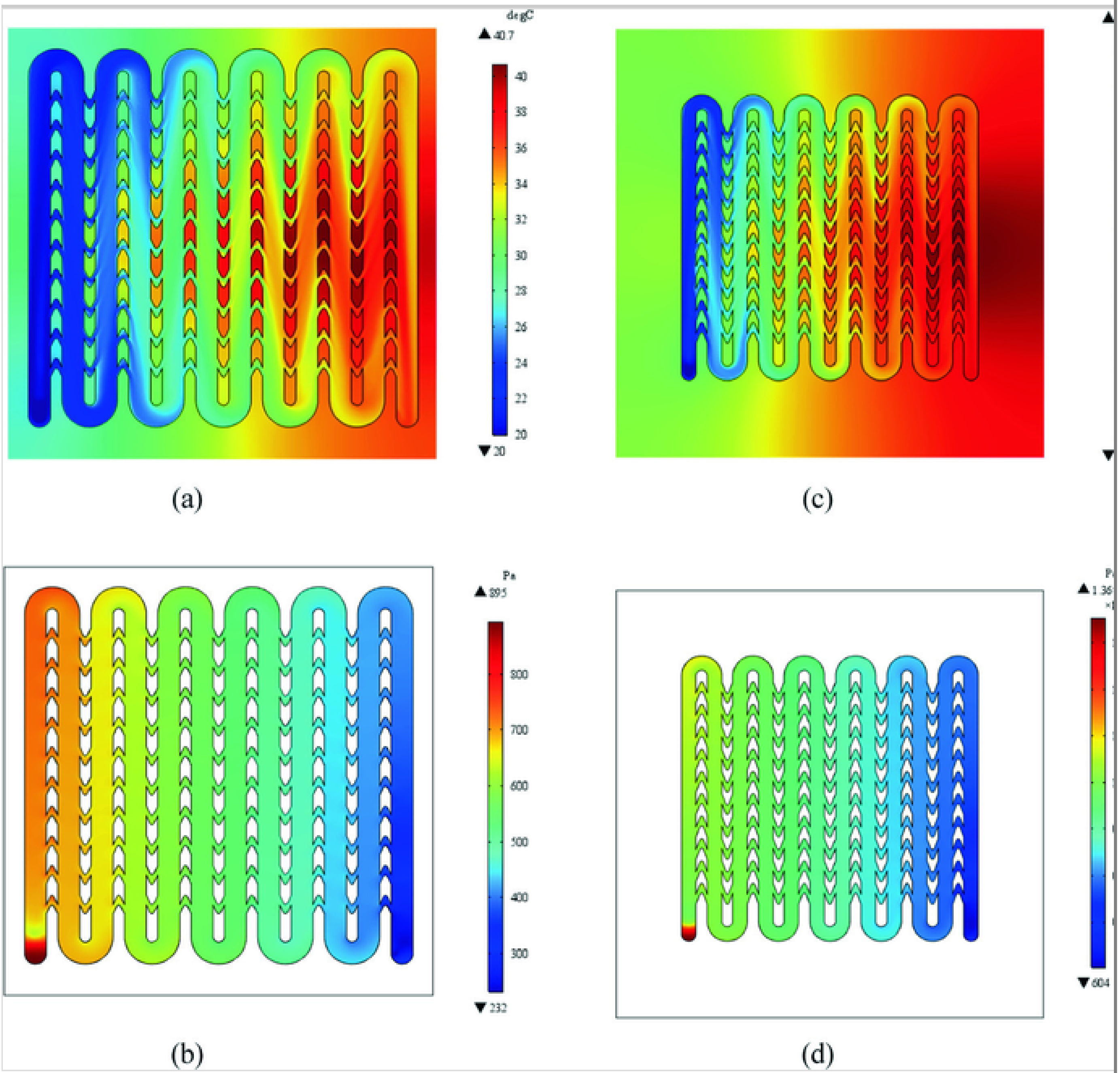


Table 1

Comparison between optimum and non-optimum design

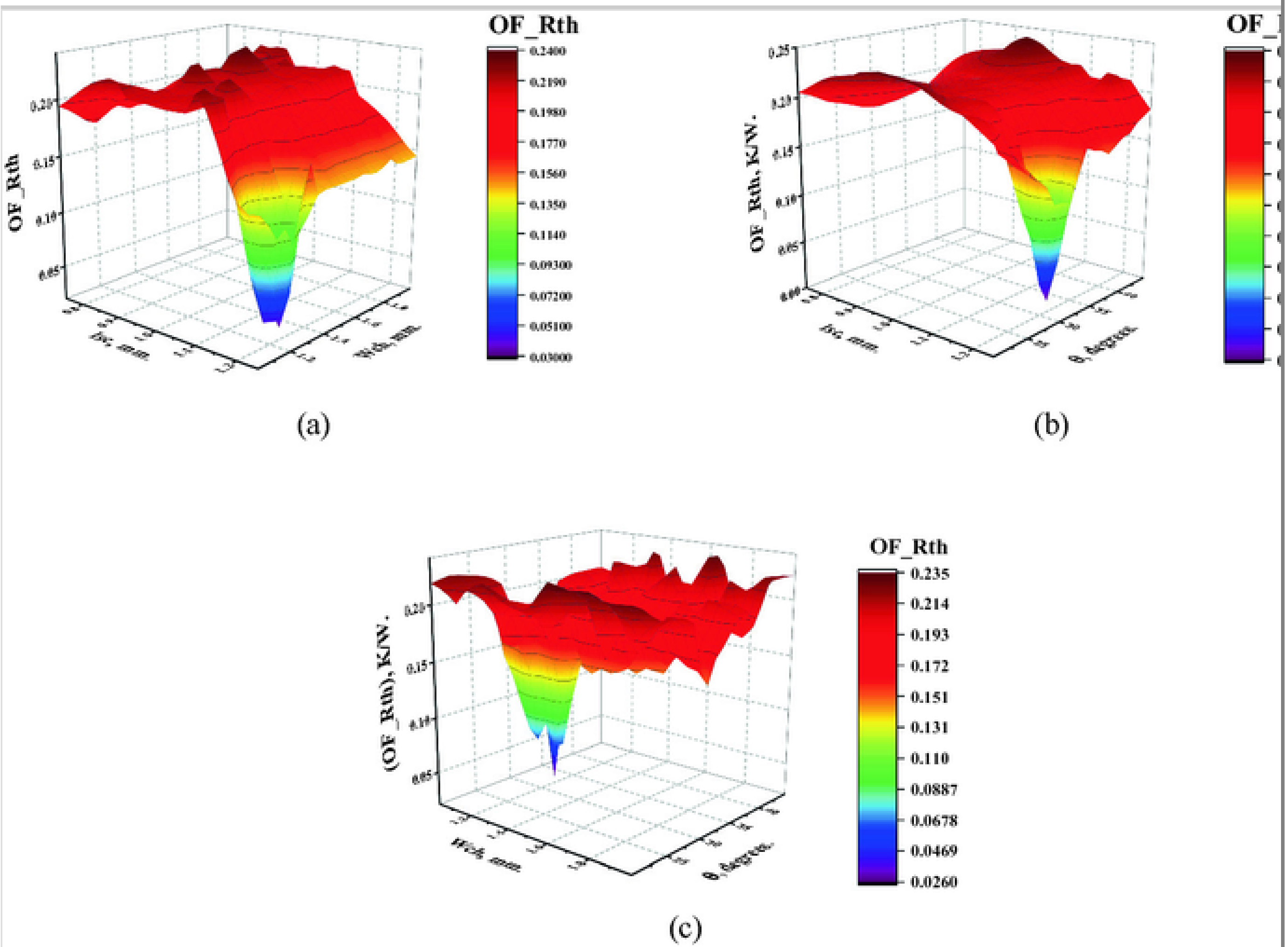
Case	Optimum design variables			Performance criteria		Max. temperature, (°C)
	$\cancel{L}l_{sc}$	W_{ch}	θ	R_{th}(K/W)	ΔP (Pa)	Heat sink Base
Optimum design, point A	0.844	1.955	25.733	0.361	1014.12	40.6

Case	Optimum design variables			Performance criteria		Max. temperature, (°C)
	~~L~~l_{sc}	W_{ch}	θ	R_{th}(K/W)	ΔP (Pa)	Heat sink Base
Non-optimum design, point B	0.865	1.283	21.99	0.434	1516.7	47.1

~~Preliminary results for the probabilistic optimisation of single objective function, thermal resistance (R_{th}), for a weighting factor of $w1 = 0.5$ has been presented below.~~ Figure 5 depicts 3D contours for the thermal resistance as a function of the main channel width (W_{ch}), the secondary channel length (l_{sc}) and the fin oblique angle (θ).Preliminary results for the probabilistic optimisation of single objective optimisation for the normalised thermal resistance (OF_Rth) with a weighting factor of $wi = 0.5$ has been presented here. These results have revealed that the optimum R_{th} (0.375 K/W) occurred with l_{sc}, W_{ch} and θ of respectively 0.852 *mm*, 1.939 *mm* and 39.21°. In contrast, the deterministic optimisation results showed that the optimum R_{th} was 0.347, and the corresponding design variables were l_{sc} = 0.751 *mm*, W_{ch} = 1.987 *mm* and θ = 20.04°. The R_{th} obtained from the probabilistic optimisation was higher than that of the deterministic optimisation by 8%. Similarly, the results for the probablistic single objective optimisation of the normalised pressure drop have shown that the optimum ΔP was 777.2 Pa which is higher than that obtained using the deterministic optimisation (741.63 Pa) by 4.79%. ~~More details to show the differences between these and the deterministic results will be discussed in detail later.~~

Fig. 5.

3D presentation of thermal resistance as function of the main channel width, secondary channel length and fin angle for $w1 = 0.5$



4. Conclusions

Deterministic and probabilistic optimisation for the liquid-cooled serpentine heat sinks with chevron fin shape have~~s~~ been conducted by considering the main channel width ($1.0\ mm \le W_{ch} \le 2.0\ mm$ ~~in mm~~), the secondary channel length ($0.75\ mm \le l_{sc} \le 1.25\ mm$~~, in mm~~) and fins oblique angle ($20° \le \theta \le 45°$~~, in degree~~) as the design variables ~~constraint~~ for the optimisation problem. The results show that these design variables have a vital effect on the performance of the heat sink under investigation. Furthermore, the optimum designs produced using probabilistic optimisation have thermal resistance and pressure drop penalty higher than that obtained using the deterministic optimisation process by 8% and 4.8%, respectively. ~~This study will be expanded to explore the effect of uncertainties of these design variables on the optimum design for this kind of microchannel heat sink to produce a robust design.~~

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