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Closed-form correlations of pressure drop and thermal resistance for a plate fin heat sink with uniform air jet impingement

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This paper proposes closed-form correlations of the pressure drop and thermal resistance for a plate fin heat sink with uniform air jet impingement, which will be useful in determining the theoretical limits for its performance under ideal conditions. The correlations are based on the volume averaging technique and validated using the corresponding numerical results. A comparison between the proposed correlations and existing experimental data revealed that the performance results by previous studies were obtained under conditions of non-uniform jet impingement and that flow uniformity at the inlet is one of the major parameters influencing the cooling performance. The contour plot of the thermal resistance ratio showed that uniform flow conditions provide the upper limit for the thermal performance of the plate fin heat sink with jet impingement. Thus, the correlations can be utilized to evaluate how the thermal performance of a heat sink with actual inlet flow conditions deviates from an ideal one. In addition, using the correlations, thermal optimization of the plate fin heat sink with uniform jet impingement was performed. For selected cases, this paper demonstrates that the cooling performance of the heat sink with optimum geometry is approximately 50% better than that of the conventional heat sink.

1. Introduction

These days many industries, such as IT, automotive, aerospace, and electronics, require a highly reliable thermal management technology that achieves effective cooling for advanced electronic devices [1–6]. A plate fin heat sink has been widely utilized among various cooling systems because of advantages of its simple structure as well as easy fabrication. The cooling performance of this type of heat sink mainly depends on geometrical configuration, the flow direction, and the flow rate. Moreover, it has a higher cooling capacity for jet impingement than for parallel flow because of more effective cooling and lower pressure drop for jet impingement [7].

Previous researchers have dealt with heat sinks with jet impingement [7–15]. Jang and Kim [7] developed models based on the porous medium approach for a microchannel heat sink with jet impingement. They presented pressure drop and thermal resistance correlations based on experimental results and their model. Biber [8] used a fully-developed flow model to numerically study effects of variable channel widths on a single isothermal channel with jet impingement and provided dimensionless correlations of the pressure loss coefficient and the average Nusselt number of the channels. Furthermore, Duan and Muzychka [9,10] provided correlations of pressure drop and thermal resistance of an impingement air-cooled heat sink with plate fins using simple models based on laminar flow in rectangular channels. Kim et al. [11] experimentally and analytically investigated the thermal performances of both plate fin and pin fin heat sinks with jet impingement and suggested a model for predicting pressure drop and thermal resistance based on the volume averaging approach. They also compared the thermal performances of the optimized heat sink with plate fins and the optimized heat sink with pin fins subject to jet impingement at the given level of pumping power. Li et al. [12] experimentally and numerically studied the plate fin heat sinks under impingement cooling with a circular nozzle by varying the Reynolds number, distance between the heat sink and nozzle, and fin geometry. They showed that increasing the Reynolds number improved the thermal performance of the heat sink but the increasing rate of the improvement declined slowly with an increasing Reynolds number. They further suggested the optimum nozzle distances for minimal thermal resistance of the heat sink with respect to the Reynolds number. Mesalhy and El-Sayed [13] investigated the effect of the jet width, number of fins, and fin height on the thermal performance of the heat sink.

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Nomenclature

c_f	specific heat [J kg^{-1} K^{-1}]
h_{sf}	interstitial heat transfer coefficient [W m^{-2} K^{-1}]
k	conductivity [W m^{-1} K^{-1}]
K_y	permeability [m^2]
P	pressure [Pa]
Pr	Prandtl number
P_{pump}	pumping power [W]
q	heat transfer rate [W]
Re_{w_c}	Reynolds number based on channel width $(= V_0 w_c/\nu_f)$
R_{th}	thermal resistance [K W^{-1}]
T	temperature [°C]
V_0	inlet velocity of the fluid phase [m s^{-1}]

Greek symbols

β	aspect ratio of a channel $(= H/w_c)$
ε	porosity $(= w_c/(w_c + w_w))$
η_{fin}	fin efficiency
ν	dynamic viscosity [m^2 s^{-1}]
ρ	density [kg m^{-3}]
σ_F	flow non-uniformity parameter

Subscripts/superscripts

opt	optimum
w	wall
∞	ambient

Special symbols

$\langle\rangle$	volume-averaged value

They presented an optimized heat sink geometry using an Entropy Generation Minimization method. Recently, Chingulpitak and Wongwises [14] reviewed research works related to the effect of flow directions and behaviors on the thermal performance of conventional heat sinks and concluded that more studies were necessary to deeply discuss the effects of flow directions and behaviors on the performance of the heat sinks.

Existing models and correlations [7–11] for a plate fin heat sink with air jet impingement were developed with the assumption that the fluid uniformly impinges on the inlet of the heat sink. However, they did not experimentally confirm flow uniformity at the inlet. Recently, our group [15] reported that flow conditions at the inlet affect the flow behavior in the heat sink from stagnation pressure measurement. Furthermore, numerical study done by Kim and Kuznetsov [16] clearly showed that a fluid in jet impingement tended to bypass the heat sink due to the high flow resistance of the heat sink. As shown in Fig. 1(a), the fluid does not uniformly impinge on the inlet of the heat sink and much of it also follows a path along which there is the least flow resistance. It is therefore difficult to achieve uniform jet impingement at the inlet. As a result of the non-uniformity, the fluid is not sufficiently supplied to the core region of the heat sink and effective cooling of the heat sink is thus limited, which in turn causes a highly non-uniform wall temperature distribution of the heat sink [17]. In a nutshell, this led us to believe that the performance results of previous studies [7–11,17] were obtained under conditions of non-uniform jet impingement. On the other hand, Do et al. [18] analyzed fluid flow and heat transfer characteristics of the heat sink under the assumption of uniform jet impingement. They showed that the fluid sufficiently penetrated into the heat sink and effectively cooled it in that condition. Therefore, we can expect the thermal performance of a heat sink with uniform jet impingement to be better than that with non-uniform jet impingement.

Although they presented solutions for the velocity and temperature distributions of the heat sink with uniform jet impingement, pressure drop and thermal resistance cannot be explicitly determined through that model's results because additional numerical calculations need to be involved. Moreover, to our knowledge, there have been no closed-form correlations on pressure drop and thermal resistance of a heat sink with uniform jet impingement.

The main objective of this paper is to present closed-form correlations of pressure drop and thermal resistance for a plate fin heat sink with uniform air jet impingement, which will be useful in determining the theoretical limits for the performance of the heat sink under ideal conditions. To verify the proposed correlations, numerical simulations are performed for various heat sink geometries and inlet velocities. In addition, the effect of flow uniformity on the thermal performance of a plate fin heat sink with jet impingement is to be examined. For this, the proposed correlations are compared with existing experimental data, which were apparently performed under conditions of non-uniform jet impingement. Finally, thermal optimization is performed for varying fin heights and pumping powers when the flow impinged uniformly onto the plate fin heat sink.

2. Closed-form correlations

The plate fin heat sink considered in this study is shown in Fig. 2. The cooling air uniformly impinges on the heat sink along the y-axis and then flows parallel to the x-axis, which is schematically represented in Fig. 1(b). The fluid absorbs heat from a heat-dissipating component attached to the base of the heat sink.

2.1. Governing equations

The governing equations were obtained using the volume averaging technique [19]. By averaging the Navier-Stokes equations and energy equations along the z-axis, the governing equations and boundary conditions of the fluid and solid phase can be obtained, respectively, as follows:

The x-momentum equation:

$$(1+\alpha_M)\left[\langle u\rangle_f \frac{\partial\langle u\rangle_f}{\partial x} + \langle v\rangle_f \frac{\partial\langle u\rangle_f}{\partial y}\right] = -\frac{1}{\rho_f}\frac{\partial\langle P\rangle_f}{\partial x} + \nu_f\left(\frac{\partial^2\langle u\rangle_f}{\partial x^2} + \frac{\partial^2\langle u\rangle_f}{\partial y^2}\right) - \frac{\varepsilon\nu_f}{K_x}\langle u\rangle_f \tag{1}$$

The y-momentum equation:

$$(1+\alpha_M)\left[\langle u\rangle_f \frac{\partial\langle v\rangle_f}{\partial x} + \langle v\rangle_f \frac{\partial\langle v\rangle_f}{\partial y}\right] = -\frac{1}{\rho_f}\frac{\partial\langle P\rangle_f}{\partial y} + \nu_f\left(\frac{\partial^2\langle v\rangle_f}{\partial x^2} + \frac{\partial^2\langle v\rangle_f}{\partial y^2}\right) - \frac{\varepsilon\nu_f}{K_y}\langle v\rangle_f \tag{2}$$

Energy equation of the fluid phase:

$$\varepsilon\rho_f c_f\langle v\rangle_f \frac{\partial\langle T\rangle_f}{\partial y} = \varepsilon k_f \frac{\partial^2\langle T\rangle_f}{\partial y^2} + h_{sf}a_{sf}(\langle T\rangle_s - \langle T\rangle_f) \tag{3}$$