International Journal of Heat and Mass Transfer 111 (2017) 1050–1062

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Numerical and experimental study of mixed convection heat transfer and fluid flow characteristics of plate-fin heat sinks

Han-Taw Chen [a,*], Hung-Chia Tseng [a], Shih-Wei Jhu [a], Jiang-Ren Chang [b]

[a] Department of Mechanical Engineering, National Cheng Kung University, Tainan 701, Taiwan
[b] Department of Systems Engineering and Naval Architecture, National Taiwan Ocean University, Keelung 202, Taiwan

ARTICLE INFO

Article history:
Received 20 February 2017
Received in revised form 12 April 2017
Accepted 13 April 2017

Keywords:
Plate-fin heat sinks
Mixed convection
Near-wall treatment
CFD
Inverse method

ABSTRACT

This study applies three-dimensional computational fluid dynamics (CFD) commercial software along with the inverse method and experimental data to determine the mixed convection heat transfer and fluid flow characteristics of a plate-fin heat sink in a wind tunnel. The inverse method of the finite difference method along with the experimental temperature data is applied to determine the unknown heat transfer coefficient on the fin. Commercial software combined with various flow models is used to obtain air temperature and velocity profiles, heat transfer coefficient on fins, fin surface temperature and pressure drop. More accurate heat transfer and fluid flow characteristics can be obtained by the appropriate flow model and the number of grid points, if the resulting heat transfer coefficient and the fin temperature at each measurement location are close to the inverse results of the heat transfer coefficient and the experimental temperature data, respectively. The interesting finding is that the results obtained by the RNG k-ε turbulence model are more accurate than those by the laminar flow model. FLUENT 4 has better accuracy than FLUENT 15 along with standard wall functions and enhanced wall treatment. In addition, the total number of grid points needs to be increased with increasing air velocity and fin spacing. The dimensionless wall distance can vary with air velocity. The pressure drop has a large variation in the specific range of the fin spacing, and the secondary vortices can be found at both corners of the wind tunnel. It is worth mentioning that the strength of the secondary flow decreases with decreasing fin spacing. The effect of the flow model, near-wall treatment, FLUENT version and grid points on the results obtained cannot be ignored. To our knowledge, few researchers have used similar methods to investigate this problem in the open literature. The two proposed correlations are closer to the obtained inverse and numerical results than the existing results.

1. Introduction

Due to the rapid development of electronic technology, electronic devices are ubiquitous in our lives. When the size and weight of electronic components are reduced, the heat flux per unit area increases dramatically. Under these circumstances, the working temperature of electronic components may exceed the desired temperature level. Thus, promoting the heat transfer rate during the working process at the desired operating temperature plays an important role for insuring a reliable operation of electronic components. This implies that proper heat sink design is attractive for these applications since it provides a more economical solution to the heat issue. Conventional electronic cooling using a heat sink with cooling fans is superior in terms of unit price, weight and reliability. In order to design a practical heat sink, some criteria, such as a large heat transfer rate, a low pressure drop and a simple structure, should be considered. Heat sink fins increase the heat flow per surface area. However, it is necessary to note the interaction between the local heat transfer and flow distributions within the fins while designing the heat sink. Thus, complex three-dimensional (3D) flow and thermal fields can be observed for various fin geometries. Heat transfer dissipation from an array of parallel rectangular fins on a horizontal surface has been studied for a long time. Although many researchers have applied 3D computational fluid dynamics (CFD) commercial software to obtain the heat transfer and fluid flow characteristics of plate-fin heat sinks (PFHSs) for various air velocities, the accuracy of the results needs to be verified, especially by comparison of reliable results and the experimental temperature measurements. Thus, 3D CFD software along with appropriate flow models and experimental data can help in designing high-performance heat sinks for these devices.

* Corresponding author.
E-mail address: htchen@mail.ncku.edu.tw (H.-T. Chen).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2017.04.065
0017-9310/

Nomenclature

A_f	lateral surface area of the fin, m^2
A_j	area of the jth sub-fin region, m^2
$[A]$	global conduction matrix
C	fin tip-to-shroud clearance
c_p	specific heat of air, J/(kg K)
D_h	hydraulic diameter of the fin array, m
$[K]$	global conduction matrix
$[F]$	force matrix
Gr	Grashof number, $.Gr = g\beta(T_0 - T_\infty)D_h^3/\nu^2$
g_j	gravitational acceleration component in the x_j direction, m/s^2
H	fin height, m
h	heat transfer coefficient on the fins, W/(m^2 K)
$\bar{h}$	average heat transfer coefficient on the fins, W/(m^2 K)
$\bar{h}_b$	heat transfer coefficient based on the fin base temperature, W/(m^2 K)
$\bar{h}_j$	average heat transfer coefficient in the jth sub-fin region, W/(m^2 K)
k	turbulent kinetic energy
k_a	thermal conductivity of air, W/(m K)
k_f	thermal conductivity of fin, W/(m K)
L	fin length, m
ℓ_x	distance between two adjacent nodes in the x direction, $L/(N_x - 1)$
ℓ_y	distance between two adjacent nodes in the y direction, $H/(N_y - 1)$
N	number of sub-fin regions
Nu	Nusselt number, $\bar{h}t/k_a$
Nu_d	Nusselt number, $\bar{h}_b D_h/k_a$
N_t	total number of grid points
N_{tf}	number of grid points on the lateral surface of the fin
N_x	number of nodes in the x direction
N_y	number of nodes in the y direction
p	pressure, Pa
$p*$	wetted perimeter of the fin array, m
Pr_t	turbulent Prandtl number
Q	total heat transfer rate dissipated from the fin, W
q_j	heat transfer rate dissipated from the jth sub-fin region, W
Ra	Rayleigh number, $Ra = g\beta(T_0 - T_\infty)H^3/(\nu\alpha)$
Re	Reynolds number, $V_a t/\nu$
Re_d	Reynolds number, $V_a D_h/\nu$
Ri	Richardson number, $Ri = RaPr/Re_d^2$
S	fin spacing, mm
S_{ij}	mean strain rate tensor, $(\partial u_i/\partial x_j + \partial u_j/\partial x_i)/2$
T	fin temperature, K
T_a	air temperature, K
T_j	fin temperature at the jth measurement location, K
T_0	fin base temperature, K
T_∞	ambient air temperature, K
$[T]$	global temperature matrix
t	fin thickness, m
u_i	air velocity component in the x_i direction, m/s
V_a	frontal air velocity, m/s
x, y, z	Cartesian coordinates, m
x_i	index symbol of the Cartesian coordinate system, m
y^+	dimensionless wall distance

Greek symbols

α_ε, α_k	turbulent Prandtl numbers for diffusion of k and ε
β	volumetric thermal expansion coefficient
β_t	parameter in RNG k-ε turbulence model
δ_{ij}, δ_{j2}	Kronecker delta function
ε	viscous dissipation rate of turbulence kinetic energy
η	mean flow field
η_f	fin efficiency
ν	laminar kinematic viscosity, kg/(s m)
ν_{eff}	effective kinematic viscosity, kg/(s m)
ν_t	turbulent kinematic viscosity, kg/(s m)
ρ	air density, kg/m^3

PFHSs have been widely used in cooling electronic devices because they are easy to manufacture, and have simple structure and low cost. Various forms of the PFHSs have been manufactured and supplied to markets in large quantity [1]. Numerical studies and some experiments for the thermal performance of plate-fin and plate-pin-fin heat sinks have been investigated by Yu et al. [1], Kim et al. [2], Li and Chao [3] and Yang and Peng [4]. The thermal resistance of the heat sink is obtained by the ratio of the temperature difference and the heat dissipation power applied to the fin base. This temperature difference is the difference between the maximum temperature at the fin base and the ambient air temperature. El-Sayed et al. [5] varied the fin height, fin width, fin spacing, number of fins and the distance from the fin tip to the shroud to study the performance PFHSs. They concluded that the pressure drop decreased with increasing fin spacing. The mean Nusselt number increases with increasing fin spacing. Under the assumptions of the one-dimensional heat conduction model and constant mean heat transfer coefficient, they proposed the relationship between average Nusselt number and Reynolds number. Dogan and Sivrioglu [6] used the CFD software FLUENT along with the experimental method to investigate the effect of the clearance gap on laminar mixed convection heat transfer from the fin array in the horizontal channel. The numerical results are obtained under the grid independence assumption. Culham and Muzychka [7] applied entropy generation minimization for optimization of PFHSs. Iyengar and Bar-Cohen [8] presented a coefficient of performance analysis for PFHSs in forced convection using least-energy optimization with the entropy minimization methodology. Najafi et al. [9] studied the energy and cost optimization of plate and fin heat exchangers using genetic algorithm. Xie et al. [10] used the entropy generation minimization theory coupled with constructal law to determine the optimal pin-fin of the heat exchanger.

Velayati and Yaghoubi [11] applied the finite volume method along with the SIMPLE pressure-velocity coupling algorithm to solve 3D turbulent flow and heat transfer characteristics of parallel heated rectangular plates mounted over an insulating base plate. Their results show very complex 3D flow characteristics within parallel bluff plates. This complex flow pattern is accompanied by fluid separation and reattachment. An approximate grid point of $165 \times 88 \times 32$ is used to determine the fluid flow and heat transfer characteristics of parallel heated rectangular thick plates. It is seen from Refs. [4,11] that the solution is substantially independent of the grid for each case. Sparrow et al. [12] used experimental investigation to study heat transfer and pressure drop for airflow in arrays of heat generating rectangular modules deployed along one wall of a flat rectangular duct. Under the assumptions of uniform air velocity and one-dimensional heat flow, Elshafei [13] applied theoretical and experimental studies to investigate the effects of the duct velocity, fin density, tip-to-shroud clearance on flow bypass, thermal performance and pressure drop across a longitudinal aluminum fin array. It is seen from Ref. [14] that a comparison of the numerical predictions and experimental data is made for the hydrodynamic and thermal fields in a two-channel plate heat exchanger using laminar and two-equation turbulence models.