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Heat transfer enhancement of vertical dimpled fin array in natural convection

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ABSTRACT

This numerical study examines the steady-state three-dimensional natural convective flow and heat transfer for a set of vertical fin arrays with/without dimples. The finite volume method is adopted to solve the Navier–Stokes and energy equations using semi-implicit method for pressure-linked equation (SIMPLE) with the converged solutions from the iterative steps to acquire the velocity field, temperature field, and Nusselt number (Nu). The free convective flow and heat transfer for different vertical fin arrays are analyzed at Rayleigh numbers (Ra) of 10^8, 7.75×10^7, 5.5×10^7, and 3.25×10^7 with the fixed Prandtl number of 0.71. For each Ra tested, four vertical fin arrays, namely smooth thirteen-fin array, smooth nine-fin array, dimpled nine-fin array, and dimpled seven-fin array, with the same fin base area and fin-array volume are individually analyzed. The results indicated the respective worst and best heat transfer performances for the smooth thirteen-fin and dimpled nine-fin arrays. As Ra increases, the mean Nu over each fin surface increases, especially for the dimpled fin arrays. Relative to the smooth thirteen-fin array, the maximum increase of mean Nu is 68% for the dimpled nine-fin array.

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1. Introduction

In addition to electronic components, many machineries such as electrical motors/generators and automobiles adopt free convective mechanisms for heat dissipations in attempts to reduce the operating temperatures for ensuring system stability and durability. With electronic cooling applications, the typical temperatures of a heat source from an electronic chipset is below 80 °C to formulate a narrow Ra band for detailed free convection analysis. As the driven potential of gravity for free convection is directional and limited by earth gravity, the cooling performance improvements for a free convective cooling system still commonly rely on the increased heat dissipation area [1] and thermal conductivity of fin. The finned heat sinks by forced convection can dissipate larger amounts of heat primarily owing to the airflows induced by fans, but this type of cooling measure consumes energy and undermines the system durability due to the finite life span of a cooling fan. When occasions are permissible with less concentrated heat fluxes, natural convection heat sinks are therefore widely adopted, in particular for the applications requesting high reliabilities. Heat transfer enhancements subject to the directional and limited driven potential of earth gravitation are accordingly urged to extend the applications of free convective heat sinks.

Zografos and Sunderland [2] conducted an experimental study to explore the effects of pin–fin diameter to center-to-center space ratio on the free convective performances of the in-lined and staggered pin fin arrays. Their results displayed that with the similar operating conditions, the heat transfer performance of pin–fin array outweighed the plate fin array. With a wide range of Ra and array geometries, the empirical heat transfer correlation was developed for predicting the thermal performances of pin fin arrays. The best heat transfer performances emerged from the pin–fin array with the pin–fin diameter to center-to-center spacing of 1/3. Iyengar and Bar-Cohen [3] examined the reported Nusselt number correlations for free convective flows and studied the heat transfer performances of the flat, cylindrical and triangular fins. Among the comparative group reported in Ref. [3], the triangular fin exhibited the better heat transfer performances. Ledezma and Bejan [4] conducted both numerical simulations and experimental measurements to attack free/forced convective flows with the attempts to explore the effects of shapes, orientation and position of fin arrays on the temperature and Nusselt number distributions at various Ra values. The vertical fin array showed the better cooling performance than the horizontal counterpart with the cooling effect enhanced significantly by increasing the slopes of fin arrays. Joo and Kim [5] analytically studied the free convective heat

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Nomenclature

English symbols

A_{fin}	fin area (m^2)
A_{hs}	heat source area
AR	area ratio ($\frac{A_{fin}}{A_{fin,F1}}$)
g	gravitational acceleration ($m\ s^{-2}$)
k	thermal conductivity of fluid ($W{\cdot}m^{-1}{\cdot}K^{-1}$)
k_s	thermal conductivity of solid ($W{\cdot}m^{-1}{\cdot}K^{-1}$)
L	length of calculation domain (m)
$L1$	length of fin (m)
M	height of calculation domain (m)
N	width of calculation domain (m)
Nu	local Nusselt number $\left(=\frac{q''L1}{k(T_w-T_\infty)}\right)$
$\overline{\mathrm{Nu}}$	averaged Nusselt number ($=\int \mathrm{Nu}dA/\int dA$)
P	pressure (N/m^2)
Q	heat generate rate per volume (W/m^3) $\left(=\frac{q''A_{hs}}{V_{fin}}\right)$
q''	heat flux ($W{\cdot}m^{-2}$)
Ra	Raleigh number $\left(=\frac{g\beta q''L1^4}{k\alpha\nu}\right)$
T	temperature (K)
u, v, w	velocity ($m{\cdot}s^{-1}$)
V_{fin}	fin volume
x, y, z	Cartesian coordinates (m)

Greek symbols

α	thermal diffusivity of fluid ($m^2{\cdot}s^{-1}$)
β	thermal expansion coefficient (K^{-1})
ρ	density ($kg{\cdot}m^{-3}$)
μ	kinematic viscosity ($kg{\cdot}s^{-1}{\cdot}m^{-1}$)
ν	dynamic viscosity ($m^2{\cdot}s^{-1}$)

Subscripts

w	wall
∞	ambient

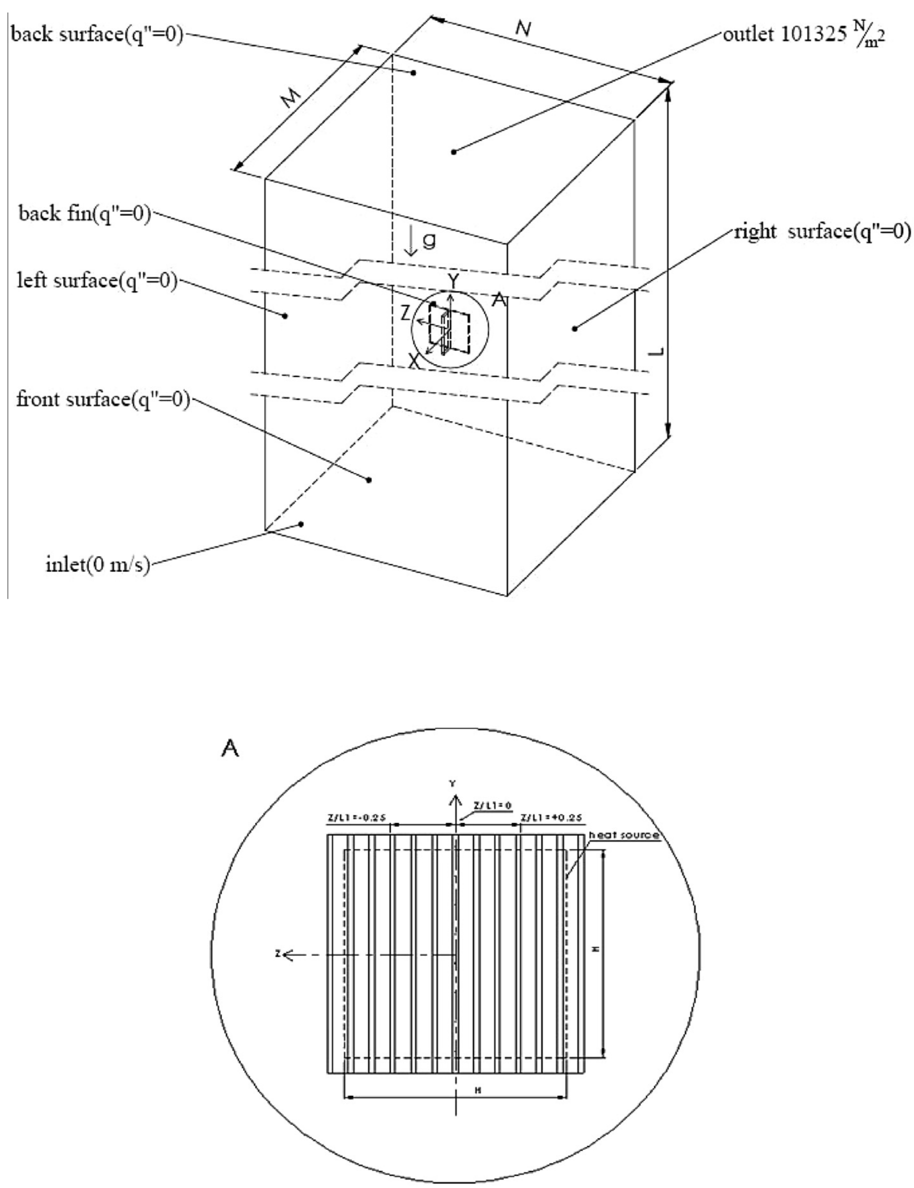

Fig. 1. Three-dimensional flow geometries and the relevant boundary conditions.

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