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Effect of secondary flows due to buoyancy and contraction on heat transfer in a two-section plate-fin heat sink

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ABSTRACT

The effect of buoyancy forces on laminar heat transfer inside a variable width plate-fin heat sink is numerically analyzed: the configuration under investigation comprises an array of rectangular fins, the number of which is doubled at the streamwise middle length of the plate, leading to a stepwise reduction in the respective channel width and hydraulic diameter. The mixed convection problem is thoroughly examined for Archimedes numbers in the range $Ar = 1.32–5.82$ and Reynolds numbers, based on the channel hydraulic diameter before the stepwise reduction, in the range $Re = 559–667$, under the thermal boundary condition of axially constant heat flux. It is illustrated that the secondary flow pattern emanating from the flow contraction and manifested through the presence of a pair of counter-rotating horseshoe vortices and a pair of counter-rotating (fin) sidewall vortices interacts with longitudinal rolls created by buoyancy forces. In fact, the lower horseshoe vortices that are co-rotating with the buoyancy-induced rolls are significantly enhanced in magnitude and cause intense fluid mixing in the vicinity of the channel bottom wall, with a substantial distortion of the temperature field. The numerical results indicate that the joint action of the buoyancy-induced rolls and the combined secondary flow pattern has a beneficial impact on the heat sink thermal performance, a fact quantified through the circumferentially-averaged local Nusselt number distributions. The effect of the top lid thermal conductivity on the heat transfer inside the heat sink is also discussed. Finally, a comparative investigation is conducted between the present variable-channel-width configuration and two configurations of fixed-width heat sink designs. The comparative results reveal that the introduction of stepwise channels leads to superior heat transfer performance, i.e. lower values of the total thermal resistance with mitigated pressure drop penalty and increased temperature uniformity on the cooled surface.

1. Introduction

Heat transfer enhancement and effective heat flux dissipation are of vital importance to a significant number of engineering applications ranging from cooling of electronics and various industrial processes to concentrated solar power applications. A practice commonly employed in various heat exchanging devices, in order to increase the overall heat transfer rate, is the use of extended surfaces, such as fins or pins, with a proper geometrical layout so that the ratio of the area available for heat transfer to the overall volume of the device is maximized. An additional factor which, in many cases, leads to enhanced heat transfer is the onset of secondary flow patterns, which can arise, among others, due to the morphology of the surface geometry, the effect of buoyancy forces or by some external excitation (e.g. surface vibration) [1,2]. If the topology of these patterns is such as to allow free-stream fluid entrainment toward the heated surfaces, then the overall heat transfer rate is significantly enhanced due to the local thinning of the boundary layer and the intense thermal mixing [1]. Research in the field of heat transfer enhancement is still ongoing especially regarding heat exchanging configurations employing extended surfaces, which are the most commonly encountered in industrial applications. Various fin designs have been proposed and investigated, e.g. wavy, louvered, corrugated or offset-strip fins and a comprehensive overview of the flow and heat transfer behavior of characteristic plate-fin configurations can be found in [3,4]. Enhanced heat-transfer designs have also been proposed for layouts in the microscale [5]. In addition, techniques used for heat transfer enhancement through the prevalent flow conditions are reported in the review articles by Jacobi and Shah [2] and Bergles [6].

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Nomenclature

a	wall thickness to channel width ratio $a = \frac{W_w}{W_{ch}}$
A	area, m^2
Ar_i	Archimedes number of heat-sink section i $(i = 1,2)$ $Ar_i = \frac{Gr_i}{Re_i^2}$
AR	aspect ratio $AR = \frac{H_{ch}}{W_{ch}}$
C_p	pressure coefficient $C_p = \frac{p-p_{ref}}{1/2\rho w_i^2}$
c_p	specific heat, J/kg K
CR	concentration ratio $CR = \frac{A_{reflector}}{A_{receiver}}$
$D_{h,i}$	hydraulic diameter of heat-sink section i $(i = 1,2)$ $D_{h,i} = \frac{2W_{ch,i}H_{ch}}{(W_{ch,i}+H_{ch})}, m$
Gr_i	Grashof number of heat-sink section i $(i = 1,2)$ $Gr_i = \frac{g\beta q'' D_{h,i}^4}{\nu^2 k_f}$
H	overall heat-sink height $H = H_{ch} + t_s$, m
H_{ch}	channel height, m
h	heat transfer coefficient $h = \frac{-k(\partial T/\partial n)_w}{(T_w - T_{f,m})}$, W/m^2 K
k	thermal conductivity, W/m K
L	length, m
L_Z^*	non-dimensional reattachment length $L_z^* = \frac{L_z}{W_w}$
Nu_i	Nusselt number of heat-sink section i $(i = 1,2)$ $Nu_i = \frac{h \cdot D_{h,i}}{k_f}$
p	pressure, Pa
Pr	Prandtl number $\Pr = \frac{c_p \mu}{k}$
p_{ref}	pressure at the channel mid-length (L/2) for undisturbed parallel flow, Pa
Q	heat rate, W
q''	heat flux, W/m^2
Ra	Rayleigh number $Ra = GrPr$
Re_i	Reynolds number in heat-sink section i $(i = 1,2)$ $Re_i = \frac{w_{m,i} D_{h,i}}{\nu}$
R_{th}	thermal resistance, K/W
s	unwound coordinate, m
S^*	non-dimensional unwound coordinate $S^* = \frac{s-z_0}{D_{h,1}}$
T	temperature, K
t_s	solid substrate thickness, m
$\dot{V}$	volumetric flow rate, m^3/s
W	width, m
w	flow axial velocity, m/s
x, y, z	streamwise, vertical and spanwise coordinate respectively, m
X^*	non-dimensional spanwise coordinate $X^* = \frac{x}{W_{ch,2}} - \frac{1}{2}$
Y^*	non-dimensional height-wise coordinate $Y^* = \frac{y-t_s}{H_{ch}}$
Z^*	non-dimensional streamwise coordinate $Z^* = \frac{z-z_0}{D_{h,1}}$
z_0	location of the flow contraction $z_0 = 0.25$ m

Greek symbols

β	volumetric thermal expansion coefficient, K^{-1}
μ	dynamic viscosity, Pa s
ν	kinematic viscosity, m^2/s
ρ	density, kg/m^3
ω	vorticity $\vec{\omega} = \vec{\nabla} \times \vec{v}$, s^{-1}

Subscript

cal	caloric
conv	convective
cond	conductive
ch	channel
cs	cross section
f	fluid
FD	fully developed
hs	heat sink
i	inlet
init	initial
int	interface
max	maximum
ove	overall
ref	reference
s	solid, section
tot	total
w	wall

Two basic configurations of flow-contraction, usually found in heat dissipation applications, are the forward facing step and flow entrance into an array of parallel fins, with the latter having been primarily investigated under forced convection conditions and using air as the cooling medium [7–9]. However, few studies are available in the open literature that discuss the effect of buoyancy on heat transfer in such configurations. The published numerical and experimental results concerning laminar mixed convection over a forward-facing step are summarized in [10]. The effects of fin height and spacing on mixed-convection heat transfer around an array of longitudinal plates shrouded by a rectangular duct were in depth investigated in the experiments of Maughan and Incropera [11]. The bottom and upper duct walls were isothermally heated and cooled respectively. It was observed that for Reynolds numbers within the laminar regime and small fin height, the reduction of fin spacing reduced the overall heat transfer, despite the additional heat-transfer areas, due to the weakening of the buoyancy induced secondary vortices. However, the opposite trend was observed for fins of considerable height as the enhancement due to the extended surfaces had a decisive impact on the overall heat transfer rate. Chong et al. [12] conducted an experimental investigation of the laminar mixed convection inside a horizontal rectangular duct having a vertical longitudinal plate at the middle of its cross-section. The imposed thermal boundary condition was that of axially uniform heat flux with circumferentially constant wall temperature. The authors concluded that heat transfer was enhanced with an increase in the Reynolds number, due to the increased impact of forced convection, while duct inclination was found to have either a beneficial or a hindering influence on heat transfer, depending on the orientation of the buoyancy force in comparison to the flow direction. More recently, Dogan and Sivrioglu [13] examined both numerically and experimentally the overall heat transfer from an array of longitudinal fins mounted inside a rectangular horizontal channel under mixed convection conditions. A constant heat flux was applied at the channel bottom wall, while the heat was dissipated through laminar air flow. The authors analyzed the effect of fin spacing and clearance gap between fin tips and the channel top surface on the average heat transfer coefficient inside the channel. They came to the conclusion that an optimal fin spacing exists that maximizes heat transfer, while a widening of the clearance gap had a negative impact on heat transfer, a trend more pronounced for larger Reynolds numbers.

Whereas the available literature referring to mixed convection in flow-contraction geometries of rectangular cross section is quite limited, many researchers have addressed the same issue in parallel flow inside straight rectangular channels. Cheng et al. [14] theoretically investigated laminar mixed convection of a large Prandtl fluid in horizontal channels under constant heat flux. They demonstrated that the buoyancy effect enhances heat transfer and re-