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Experimental study of mixed convection heat transfer in a vertical channel with a one-sided semicylindrical constriction with prescribed heat flux

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ABSTRACT

An experimental study of mixed convection heat transfer is carried out in a vertical channel with a one-sided semicylindrical constriction with prescribed heat flux while the other bounding walls are insulated and adiabatic. The semicylinder is placed horizontally at the mid-plane with a blockage ratio (*BR*, ratio between the semicylinder diameter and the thickness of the rectangular section) of 0.3 and a semicylinder aspect ratio (*AR*, ratio between the length and diameter of the semicylinder) of 6. The effect of opposing buoyancy on the flow and thermal behavior is analyzed for fixed Prandtl number of $Pr = 7$, Reynolds number based on semicylinder diameter of $20 \le Re \le 350$ and buoyancy strength or modified Richardson number, $Ri^* = Gr^*/Re^2$, from 20 to 350. For relatively large values of Ri^*, flow visualization images and thermal analysis confirm the presence of a complex three-dimensional (3D) two-vortex structure with two recirculation bubbles present at the forebody and rear of the semicylinder. Surface temperature distributions and averaged Nusselt number at different Reynolds and modified Richardson numbers have been obtained. The results show that variation of the local temperature distributions with angular position and spanwise location become evident, and their relation to the presence of a complex 3D vortex structure that develops close to the semicylindrical constriction has been studied and discussed in detail. Moreover, empirical correlations for the overall Nusselt number are obtained using both *Re* and Gr^* and *Re* and Ri^* as the controlling parameters.

1. Introduction

Flow in constricted channels with irregular surfaces is often present in many fluidic devices and has wide engineering applications in the cooling of electronic equipment, design of compact heat exchangers, nuclear reactors, solar collection systems, building energy systems, fin-tube baseboard heaters, flat-plate condensers in refrigerators, energy storage systems and electric machinery. Among these, cooling of electronic equipment draws much attention because the production of smaller components with higher power densities has generated an increased interest for dependable and efficient cooling technologies. In most electronic equipment applications, cross-sectional protuberances or heated irregular surfaces in blocked passages are a common occurrence. Because complex geometries generate detachment and reattachment of flows and develop recirculation regions that enhance mixing and significantly improve the thermal performance of heat transfer devices, a good amount of research has been carried out to understand the role of geometrical inhomogeneities and the effect of these obstructions on the modification of the flow and heat transfer characteristics (Viswatmula and Amin, 1995; Roeller et al., 1991; Young and Vafai, 1999; 1998a, 1998b; Habchi and Acharya, 1986; Pirouz et al., 2011; Hamouche and Bessaih, 2009; Boutina and Bessaih, 2011; Chang and Shiau, 2005; Rao and Narasimham, 2007; Du et al., 1998). Examples of these complex geometries can be found in grooved (Adache and Uehara, 2001; Herman and Kang, 2002; Ghaddar et al., 1986; Greiner, 1991; Pereira and Sousa, 1993; Farhanieh et al., 1993) and corrugated channels (Wang and Chen, 2002; Alawadhi and Bourisliy, 2010; Forooghi and Hooman, 2013; Guzmán et al., 2009), channel expansions and chimneys (Thiruvengadam et al., 2009; Wahba, 2011; Auletta et al., 2001;

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Nomenclature

A_{cyl}	surface area of the semicylinder, m^2
AR	semi-cylinder aspect ratio, W/D
BR	blockage ratio, $D/2H$
D	semicylinder diameter (characteristic length), m
D_H	hydraulic diameter, m
f	frequency, Hz
g	gravity acceleration, m s^{-2}
Gr^*	modified Grashof number, $Gr^* = g\beta\dot{q}D^4/k\nu^2$
h	heat transfer coefficient, W m^{-2} K^{-1}
H	channel width, m
I	measured electrical current, A
k	fluid thermal conductivity, W m^{-1} K^{-1}
$\widetilde{Nu}$	time-and-space averaged Nusselt number
Pr	Prandtl number, $Pr = \nu/\alpha$
Q	net convective heat transferred to the fluid, W
$\dot{q}$	net convective heat flux transferred to the fluid, W m^{-2}
$\dot{q}_{cond}$	calculated conduction losses per unit surface to the ambient, W m^{-2}
$\dot{q}_{el}$	measured input power per unit surface supplied to the semicylinder, W m^{-2}
$\dot{q}_{rad}$	calculated radiation losses per unit surface to the ambient, W m^{-2}
Re	Reynolds number based on semicylinder diameter, $Re = u_0D/\nu$
Ri^*	modified Richardson number, $Ri^* = Gr^*/Re^2$
St	Strouhal number based on semicylinder diameter, $St = fD/u_0$
t	time, s
T_{amb}	room temperature, K
T_0	fluid temperature at the channel inlet or reference temperature, K
T_w	local surface temperature, K
$\overline{T}_w$	mean local wall or surface temperature, K
$\widetilde{T}_w$	mean global wall temperature, K
$\overline{T}_{wz}$	row-averaged surface temperature, K
$\overline{T}_{w\gamma}$	angular-averaged surface temperature, K
u_0	fluid velocity at the channel inlet, m s^{-1}
V	measured voltage, V
W	semicylinder span or channel depth, m
x, y, z	rectangular Cartesian coordinates
X	nondimensional axial coordinate, $X = x/D$
Y	nondimensional transverse coordinate, $Y = y/D$
Z	nondimensional spanwise coordinate, $Z = z/W$

Greek symbols

α	thermal diffusivity, m^2 s^{-1}
β	thermal volumetric expansion coefficient, K^{-1}
$\delta T_{w\gamma}$	amplitude of the temperature variations along the azymuthal direction
δT_{wz}	amplitude of the temperature variations along the axial direction
ΔT_w	local surface temperature difference, $\Delta T_w = (T_w - T_0)$, K
$\Delta\overline{T}_w$	mean local surface temperature difference, $\Delta\overline{T}_w = (\overline{T}_w - T_0)$, K
$\Delta\widetilde{T}_w$	mean global surface temperature difference, $\Delta\widetilde{T}_w = (\widetilde{T}_w - T_0)$, K
$\Delta\widetilde{T}_{wa}$	average temperature difference, $\Delta\overline{T}_{wa} = (\widetilde{T}_w - T_{amb})$, K
$\Delta\overline{T}_{wz}$	row-averaged temperature difference, $\Delta\overline{T}_{wz} = (\overline{T}_{wz} - T_0)$, K
$\Delta\overline{T}_{w\gamma}$	angular-averaged temperature difference, $\Delta\overline{T}_{w\gamma} = (\overline{T}_{w\gamma} - T_0)$, K
ε	surface emissivity of aluminum
γ	angular coordinate
γ_i	refers to the angular position in degrees (see Fig. 3)
ν	kinematic viscosity, m^2 s^{-1}
ρ	fluid density, kg m^{-3}
σ	Stefan-Boltzmann constant, $5.670373(21)10^{-8}$ W/ m^2 K^4
τ	nondimensional time, tu_0/D

Manca et al., 2003; Andreozzi et al., 2012), passages with eddy promoters (Diani et al., 2013; Santos and de Lemos, 2006; Guerroudj and Kahalerras, 2010; Premachandran and Balaji, 2006; Tanda, 2004), and channels with curved or wavy walls (Wang and Vanka, 1995; Jang and Yan, 2004; Oztop, 2005; Barboy et al., 2012; Ko and Cheng, 2007; Parvin and Hosain, 2012; Singh et al., 2016; Mills et al., 2016). In spite that the flow and heat transfer from curved surfaces is more complex than their counterparts in straight vertical channels, the available literature in ducts with concave or convex surfaces appears to be more limited. Moukalled et al. (2000) studied numerically the effects of the Reynolds and Prandtl numbers on the heat transfer characteristics in channels with concave and convex entry surfaces subjected to favorable and unfavorable pressure gradients. Their results show that the overall heat transfer in a concave-entry channel is always greater than a straight channel of equal height, while this comparison is not always favorable for convex-entry geometries. Lakkis and Moukalled (2008) conducted a numerical investigation to study laminar natural convection heat transfer in channels with isothermal convex surfaces for six values of the Grashof number ($10 \leq Gr \leq 10^4$) and 11 radius of curvature ($1 \leq \kappa \leq \infty$). Their results show that at the lowest radius of curvature, computations reveal the formation of recirculation zones in the exit section for all values of Gr. They reported that as the radius of curvature increases, the Gr value at which recirculation occurs also increases until it disappears at κ values grater than 1.5, and their results showed that for all configurations studied, the averaged Nusselt number increased for increasing values of the Grashof number. Pirompugd and Wongwises (2013) evaluated analytically the partially wet fin efficiency for the longitudinal fin of rectangular, triangular, concave and convex parabolic profiles, and indicated that the fin with larger cross-section has a higher conduction heat transfer rate and more fin efficiency. Mittal et al. (2003) studied numerically the pulsatile flow in a planar horizontal channel with a one-sided semicircular constriction over a range of Reynolds numbers from 750 to 2000. They found that the flow downstream of the constriction is dominated by complex dynamics associated with two shear layers, one of which separates from the lip of the constriction and other from the opposite wall, and their flow visualizations showed that beyond a Reynolds number of 1000, a series of distinct Kelvin-Helmoltz type vortices are formed in the shear layer that separtes off the lip of the constriction.

The above review of literature suggests that little attention has been paid to evaluate the free and forced convection from discretely heated curved surfaces, and from the foregoing discussion, it is clear that no prior results are available for opposing mixed convection heat transfer in channels with discrete concave or convex surfaces with a prescribed heat flux boundary condition. In particular, studies that address the effects of wall confinement on the 3D flow and thermal behavior of the vortical structure and its corresponding temperature fluctuations have been practically overlooked. This is the motivation of the present paper. In this work, a detailed experimental study of mixed convection heat transfer