International Journal of Thermal Sciences 72 (2013) 18–33

Contents lists available at SciVerse ScienceDirect

International Journal of Thermal Sciences

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

Benchmark solutions for natural convection flows in vertical channels submitted to different open boundary conditions ☆

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ARTICLE INFO

Article history:
Received 19 November 2012
Received in revised form
2 May 2013
Accepted 2 May 2013
Available online 19 June 2013

Keywords:
Natural convection
Vertical channel
Recirculation flow
Open boundary conditions
Benchmark solutions

ABSTRACT

Comparison exercises have been carried out by different research teams to study the sensitivity of the natural convection occurring in a vertical asymmetrically heated channel to four sets of open boundary conditions. The dimensionless parameters have been chosen so that a return flow exists at the outlet. On the whole, results provided by the partners are in good agreement; benchmark solutions are then defined for each of the boundary conditions. Whilst the local and average Nusselt numbers based on the entrance temperature do not depend much on conditions applied in the aperture sections, the net fluid flow rates crossing the channel and the characteristics of the recirculation cells are highly influenced. But we proved that these modifications of flow patterns do not alter significantly the fluid flow rates leaving the channel through the exit section.

1. Introduction

Heat transfer and fluid flows driven by natural convection in open channels have been extensively studied over the last past decades, for vertical or inclined configurations. This great interest raised by this subject stems from its wide range of practical applications such as solar chimney, solar energy collectors, Trombe walls, or the cooling of electronic components and many others [1–4].

Since the precursory experimental works performed by Elenbaas in 1942 [5], who determined the different flow regimes versus

☆ Supported by the French Research Group GDR AMETh: Analyse – Maîtrise des Ecoulements et Echanges Thermiques

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[1] To the memory of Gilles Desrayaud, deceased in January 2009.
[2] Coordinators of the numerical exercise.

1290-0729/$ – see front matter
http://dx.doi.org/10.1016/j.ijthermalsci.2013.05.003

Nomenclature

A	aspect ratio of the channel, $=H/l$
BC	boundary conditions
d_w	width of the downward flow (Eq. (13))
d_ψ	width of the recirculation (Eq. (14))
$\vec{e}_x$, $\vec{e}_z$	coordinate axes
Fr	Froude number, $=\alpha_0^2/(gl^3)$
g	gravitational acceleration, m/s^2
GB	boundary conditions (Eqs. (5c) and (6c)), see Table 1
GB-0	boundary conditions (Eqs. (5c) and (6d)), see Table 1
H	channel height, m
l	channel width, m
LB	boundary conditions (Eqs. (5b) and (6c)), see Table 1
LB-0	boundary conditions (Eqs. (5b) and (6d)), see Table 1
$\widetilde{Nu}_1$; Nu_1	inverse reduced temperature at the left wall; Nusselt number on the heated surface based on the reference temperature (Eq. (10))
$\widetilde{Nu}_2$; Nu_2	inverse reduced temperature between the left wall and the bulk; Nusselt number on the heated surface based on the bulk temperature (Eq. (11))
p	difference between static and hydrostatic pressures, $=\Pi+z/Fr$
Pr	Prandtl number, $=\nu_0/\alpha_0$
$q_{\rm in}(z=0)$	mass flow rate entering through $z=0$ (Eq. (7))
$q_{\rm in}(z=A)$	mass flow rate entering through $z=A$ (Eq. (8))
Ra	Rayleigh number, $=g\beta_0\Phi l^4/(\lambda_0\nu_0\alpha_0)$
t	time
T	temperature, K
T1, $\cdots$, T8	participating teams, see Table 2
(u,w)	velocity components, $u=\vec{v}\cdot\vec{e}_x$, $w=\vec{v}\cdot\vec{e}_z$
$\vec{v}$	velocity vector
(x,z)	spatial coordinates

Greek symbols

α	thermal diffusivity, m^2/s
β	thermal expansion coefficient, 1/K
θ	reduced temperature, $=(T-T_0)\times\lambda_0/(l\Phi)$
θ_b	bulk temperature (Eq. (9))
λ	thermal conductivity, W/(m K)
μ	dynamic viscosity, kg/(m s)
ν	kinematic viscosity, m^2/s
Π	static pressure
ρ	density, kg/m^3
Φ	heat flux, W/m^2
ψ	stream function (Eq. (4))

Subscripts

0	reference temperature
m	median value (Eq. (16)), benchmark solutions (Fig. 9)
σ	standard deviation (Eq. (15))

Others

—	spatial average on the heated wall (Eq. (12))
$\langle\rangle$	ensemble average (Eq. (15))

a modified Rayleigh number or Elenbaas number (Rayleigh number calculated on the channel-chimney width divided by the aspect ratio), natural and mixed convections have been widely studied in open channels, both numerically and experimentally. Because the aim of this paper is to focus our attention on the influence of the boundary conditions applied in the apertures of the channel, and then to define benchmark solutions, the detailed description of the numerous contributions dealing with natural or mixed convection in vertical channels is transferred to few relevant and recent papers chosen to provide complete reviews of this topic (see Refs. [1,6–13] and references therein). Among the first numerical simulations, we can mention the contribution of Bodoia and Osterle [14] who investigated fluid flows and heat transfer occurring through isothermal vertical plates. Their results were in good agreement with the Elenbaas experimental data [5]. Since then, and despite the plenty of numerical studies, the choice of the boundary conditions for open cavities is still a delicate issue.

For thermal natural convection, the distribution of the total heat fluxes conveyed by the fluid flow through the open boundaries depends on heat transferred at walls but also on physical conditions prevailing in the surroundings, on both sides of the apertures. It results a close coupling between the dynamic and thermodynamic variables inside and outside the channel. Thus, the thermal and kinematic inlet/outlet boundary conditions cannot be a priori prescribed without accounting for the surrounding conditions [6,15,16]. Finally, the thermally driven channel behaves as a thermal engine by converting the thermal gradient into a fluid flow. This corresponds to the so-called thermosyphon effect. Although thermal natural convection was only considered here-above, this applies for general mass transfer, whatever the origin of the density variation may be.

To overcome the issue of choosing the boundary conditions in inlet/outlet sections, some authors proposed to extend the computational domain, upstream and/or downstream the channel. Thus, the boundary conditions are pushed away from the true inlet/outlet sections of the channel. One of the ideas leading to the displacement of the open boundaries far from the channel limits is to reduce, as much as possible, the effect of an unawareness of the "real" boundary conditions. St. Venant's principle, often invoked in solid mechanics, may also be used in fluid mechanics: if the artificial boundaries are placed sufficiently far away from the channel apertures, the velocity and temperature distributions at the entrance/exit of the channel are no longer affected by the applied boundary conditions [17]. The issue now raised is how far the artificial open extensions must be placed in order that the physical quantities in the inlet/outlet sections become insensitive to the conditions set at boundaries. Although this approach seems attractive, the increase in size of the computational domain proves to be expensive, both in memory and in computational time. For these reasons, the domain extensions are often either relatively reduced or large but coarsely discretized. The shapes of the walls at the entrance region, with sharp angles or smooth rounded surfaces, affect also significantly the fluid flows and heat transfer. Using entrance walls with right angles, Naylor et al. [18] predicted a fluid separation at the channel inlet which is approximatively correlated with the dimensional flow rate. Their inlet boundary conditions were based on Jeffrey-Hamel flow which consists in a similarity solution of isothermal flow caused by the presence of a source or sink at the point of intersection of two walls. Kettleborough [17], with a parabolic approximation, and Nakamura et al. [19], with a full elliptic model, used boundary conditions which physically correspond to fully developed flow entering a channel with a large sudden section reduction. They also found a separation of the boundary layer, but at some distance from the leading edge at the entry to the channel. This issue was reconsidered in a recent paper by Boetcher and Sparrow [20] who studied buoyancy-induced flow in a horizontal open-ended cavity. They examined the impact of the size of the extended domain, the boundary conditions on its