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Unsteady heated vertical channel flow in a cavity

Yiqin Li[a,b], Hervé Pabiou[a,*], Christophe Ménézo[b]

[a] Univ Lyon, CNRS, INSA-Lyon, Université Claude Bernard Lyon 1, CETHIL UMR5008, F-69621, Villeurbanne, France
[b] Université Savoie Mont-Blanc, LOCIE UMR CNRS 5271, Campus Scientifique Savoie Technolac – Bâtiment Helios, Avenue du Lac Leman, F-73376, Le Bourget-du-Lac, France

A R T I C L E I N F O

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A B S T R A C T

This experimental work investigates buoyant flow in a differentially heated vertical channel located inside a water cavity. The flow is found to be highly unsteady, and the key aspect of this study is to consider this unsteady behavior as a succession of states that turn out to be driven by the flow outside the channel. A conditional mean operator with respect to the average wall temperature is used to disentangle the different states through which the flow passes. Most of these states are characterized by a transition from laminar heat transfer in the bottom part of the channel to turbulent heat transfer, with a transition point that moves toward the exit as the average wall temperature increases. For the highest values of the average wall temperature, no transition is observed, and the heat exchange is found to be similar to that along a single vertical plate. For an intermediate range of wall temperature, a transition zone with turbulent heat transfer is observed in the upper part of the channel, and the heat transfer is found to follow the same laws as found for a symmetrically heated channel. For the lowest values of the wall temperature, the beginning of the turbulent zone is observed near the entry. The analysis is extended to several channel widths. The origin of the unsteady behavior is attributed to the flow in the whole cavity, and the conditional mean operator allows characterization of the flow inside the channel independently from the flow in the surroundings.

1. Introduction

Natural convection has been widely studied for passive cooling of electronics components and optimization of cooling fins, and its properties in these situations are now fairly well understood. However, natural convection in larger structures such as double-skin façades on buildings poses a much trickier problem, since the flow exhibits transitional or turbulent regimes that are less well understood. A typical configuration for convection around vertical geometries is that of natural convection in an open vertical channel with wall heating in infinite surroundings. In that case, one may define a modified Rayleigh number to characterize the flow: $Ra_b^* = g\beta q_w b^5/\lambda\nu\kappa H$, where g is the gravitational acceleration, q_w the wall heat flux, b and H the width and height of the channel, and β, ν, λ, and κ the volumetric thermal expansion coefficient, kinematic viscosity, thermal conductivity, and thermal diffusivity of the fluid, respectively. The Rayleigh number compares the buoyancy term that tends to lift the fluid with the viscous and thermal diffusive terms that tend to slow it down. Natural convection flow in vertical channels has been intensively investigated for many years for applications at low Rayleigh number ($Ra_b^* < 10^5$) [1]. However, there has been little investigation of high-Rayleigh-number cases, which are appropriate for applications such as photovoltaic double-skin façades ($Ra_b^* \sim 10^{10}$).

For high-Rayleigh-number natural convection flow, transitions from laminar to turbulent flow have been reported. Miyamoto et al. [2] studied a 5 m-high air channel heated on one wall with an isoflux condition, the other wall being adiabatic. The wall temperature and velocity were measured for several aspect ratios ($H/b = 100$, 50, and 25). The maximum wall temperature was located between 1 and 2 m above the channel inlet for modified Rayleigh numbers between 2×10^4 and 2×10^7. The authors attributed this maximum to a transition from laminar to turbulent flow. A few years later, Webb and Hill [3] studied natural convection flow in a vertical air channel under an isoflux heating condition on one wall with adiabatic extensions and an adiabatic condition on the other walls. Modified Rayleigh numbers ranging from 503 to 1.75×10^7 were reached. The authors found no indication of transition to turbulent heat transfer in any of their experiments. A correlation between the local Nusselt number and the local Grashof number for modified Rayleigh numbers below 10^7 was found. More recently, a transition has been observed by Daverat et al. [4] on the same experimental bench as used in the present study but with a symmetrical heating configuration. A 611 mm-high and 45 mm-wide

* Corresponding author.
E-mail address: herve.pabiou@insa-lyon.fr (H. Pabiou).

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Nomenclature

b	Channel width (m)
$B(g)$	Fixed error of a quantity g (units are those of g)
g	Acceleration due to gravity (m s^{-2})
H	Channel height (m)
l	Channel depth (m)
Nu_x	Local Nusselt number, Eq. (8b)
P_e	Mean pressure outside the channel (Pa)
$\widehat{P_e}$	Mean driving pressure outside the channel (see B) (Pa)
q_w	Mean wall heat flux (W m^{-2})
$q_{c,\mathrm{sym}}$	Turbulent vertical heat flux outside the conductive thermal boundary layer in the top part of the channel in a symmetrically heated channel (see Eq [42] in Ref. [6]). (W m^{-2})
Ra_b^*	Modified Rayleigh number, $Ra_b^* = g\beta q_w b^5/\lambda\nu\kappa H$
Ra_x	Local Rayleigh number, Eq. (8a)
S	External stratification parameter, Eq. (11)
t	Time (s)
T	Temperature (K)
U, V	Mean velocity component in the x, y directions, respectively (m s^{-1})
$\overrightarrow{U_e}$	Mean velocity vector outside the channel (m s^{-1})
$\overrightarrow{u'_e}$	Velocity fluctuation vector outside the channel (m s^{-1})
U_I	Typical velocity of laminar flow along a vertical flat plate (m s^{-1})
u', v'	Instantaneous velocity fluctuations in the x, y directions, respectively (m s^{-1})
x	Distance from inlet in the ascending direction (m)
$x_T(t)$	Instantaneous height of the transition
y	Distance from the left wall (m)
$Y_{U=0}$	Distance from the heated wall of the position of zero vertical velocity at $x/H = 0.75$ (m)
Y_{cell}	y position of the plane separating contrarotating cells located immediately above the channel (m)
z	Distance from the front lateral wall (m)
αq_w	Horizontal heat flux through the conductive sublayer in the top part of the channel (W m^{-2})
β	Isobaric thermal expansion coefficient of water (K^{-1})
δ_V	conductive thermal boundary-layer thickness in the bottom part (m)
$\theta_w(x)$	Reduced wall temperature, Eq. (4) (K)
$\Gamma = H/b$	Aspect ratio
$\Delta T_w(x)$	Mean wall temperature difference at the entry of the channel (K)
ΔT_I	Typical temperature difference with respect to the inlet temperature of laminar flow along a vertical flat plate (K)
$\mathscr{E}(g)$	Uncertainty of a quantity g (units are those of g)
κ	Thermal diffusivity of water (m^2 s^{-1})
λ	Thermal conductivity of water (W m^{-1} K^{-1})
ν	Kinematic viscosity of water (m^2 s^{-1})
ρ	Density of water at the reference temperature (kg m^{-3})

Operators

σ_g	Root mean square of a quantity g (units are those of g)
$\langle\cdot\rangle_\xi$	Averaging operator over a variable ξ
$\langle\cdot\vert\theta_w\rangle$	Conditional mean operator, Eq. (5)
∇	Partial derivative operator
$\cdot$	Dot product of two vectors
$\otimes$	Tensor product of two vectors

Subscripts

inlet	Referring to channel inlet
e	Referring to the surroundings of the channel
max	Referring to the maximum
H	Referring to the channel outlet
T	Referring to the height of the transition
w	Referring to the heated wall
$x < x_T$	Referring to the bottom part of the channel
$x > x_T$	Referring to the top part of the channel

water channel was heated with a uniform heat flux on both walls to reach $Ra_b^* = 1.7\times10^6 - 4.3\times10^7$. A change in flow behavior was observed at $Ra_b^* = 10^7$. Detailed observations of the behavior on the same experimental bench have been reported in a recent work by the same team [5]. The channel was heated on its two walls with a uniform heat flux $q_w = 1150$ W m^{-2}; its width was $b = 59$ mm, giving a modified Rayleigh number $Ra_b^* = 6.7\times10^7$. Temperature and velocity measurements in several sections of the channel showed that this change in behavior corresponded to a transition from laminar heat transfer in the bottom part of the channel to turbulent heat transfer in the upper part. More precisely, below $x/H = 0.71$, most of the heat was transported in the conductive sublayer, whereas above $x/H = 0.71$, a significant part of the heat was transferred by fluctuations to the center of the channel, leading to a sudden increase in the temperature of the bulk flow. This transition is analyzed in detail by Li et al. [6] through a scaling analysis. Assuming a two-dimensional mean flow, the half-channel is split into seven zones where the momentum and energy equations are reduced to their leading terms. The heat transfer in the bottom part is shown to be similar to that along a single heated vertical plate with a wall temperature that obeys the following power law [7]:

$$\langle\Delta T_w\rangle_t(x) = \langle\Delta T_{w,T}\rangle_t\left(\frac{x}{\langle x_T\rangle_t}\right)^{1/5} \quad (x \le \langle x_T\rangle_t) \tag{1}$$

where x is the distance to the entry, $x_T(t)$ is the height of the transition, $\Delta T_w(x,t) = T_w(x,t) - T_{\mathrm{entry}}(t)$, $T_w(x,t)$ is the local wall temperature, $T_{\mathrm{entry}}(t)$ is the temperature at the entry and $\langle\ \rangle_t$ is the time-average operator, $\Delta T_{w,T} = \Delta T_w(x_T, t)$. In the upper part, αq_w denotes the horizontal heat flux through the edge of the conductive sublayer toward the center of the channel. The decrease in wall temperature above the transition is found to be well approximated by the following equation:

$$\langle\Delta T_w\rangle_t(x) = \langle\Delta T_{w,T}\rangle_t\left(\alpha + (1-\alpha)\frac{x}{\langle x_T\rangle_t}\right)^{1/5} \quad (x \ge \langle x_T\rangle_t) \tag{2}$$

with $\alpha = 2.6$. This decrease in wall temperature is a consequence of a transition from laminar to turbulent heat transfer. More precisely, Eq. (2) models a transition zone, and the fully turbulent regime with an increasing wall temperature is not observed in this study. These works show that, for a symmetrically heated channel, the transition is triggered by the meeting of the two shear layers that develop in the left and right half-channels between the velocity peaks close to the walls and the center of the channel. By comparison, Liu and Vliet [8] observed the beginning of the transition zone in a vertical flat-plate configuration in a water flow at $Ra_x = 10^{13}$, where Ra_x is defined by Eq. (8a). In the study by Daverat et al. [5], the Rayleigh number at the exit of the channel, Ra_H, is slightly less than 10^{13}, indicating that in a symmetrically heated vertical channel, the transition point is moved down with respect to the single-plate configuration.

Furthermore, flow reversals re-entering the channel from the outlet have been observed in several studies with asymmetrical wall heating. Sparrow et al. [9] performed an experimental study of flow reversals in a 14.5 cm-high water channel with an isothermal condition on one wall. They observed a V-shaped flow reversal close to the unheated wall near