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Correlations for transient natural convection in parallelogrammic enclosures with isothermal hot wall

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HIGHLIGHTS

► Air-filled closed parallelogrammic cavities are treated in this work.
► Transient natural convection is generated by isothermal hot wall.
► Correlations of the type Nusselt–Rayleigh–Fourier are proposed in transient regime.
► Numerical results based on the finite volume method are validated by measurements.
► This study covers a wide range of the Rayleigh number and various angles of inclination.

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ABSTRACT

Correlations of the type Nusselt–Rayleigh–Fourier are proposed to determine the convective exchanges that occur in transient regime in closed cavities of parallelogrammic section. They complement the correlations obtained in a previous work for steady state. The active vertical walls of these cavities are vertical, maintained isothermal and differentially heated, while the closing channel is adiabatic. This work covers a wide range of the Rayleigh number $1.84 \times 10^5 \leq Ra \leq 1.70 \times 10^9$ and different angles of inclination ranging from $-60°$ to $60°$, enabling applications in several fields of engineering such as building, solar energy or power electronics. Correlations are obtained from numerical results based on the finite volume method and are validated by measurements.

1. Introduction

The objective of this work is to propose Nusselt–Rayleigh–Fourier type correlations which allow determination of heat transfer by transient natural convection that occur in 2D air-filled parallelogrammic enclosures. This type of cavities has been treated in previous studies by using numerical and experimental approaches, in both transient and steady-state regimes. Nusselt number adapted to these cavities is defined by Baïri [1]. Contribution of radiation in the global heat transfer has been studied by Asako [2] and confirmed by Baïri et al. [3]. Steady state natural convection in these cavities is treated by Nakamura and Asako[4]. Naylor and Oosthuizen [5] analyze the 2D flows for a large Rayleigh range while influence of Prandtl numbers is considered by Hyun and Choi [6]. Several thermal boundary conditions were studied in particular ranges of the Rayleigh number, depending on the intended application. Baïri [3,7] and Zugari and Vullierme [8] applied it to building while thermoregulation of on-board equipments is treated by Baïri et al. [9,10]. By studying the shear stress in the vicinity of the active hot wall, Baïri [11] confirms the importance of taking into account local phenomena to set up a correct arrangement of sensors used for the thermal regulation of electronic devices. Natural convection for a window with venetian blinds is presented by Costa [12] and Costa et al. [13]. Among works dealing with the question of transient natural convection, let us cite the survey of Bhowmik and Tou [14]. The authors study the convective heat transfer concerning electronic chips located on a vertical wall of a rectangular cavity generating heat flux densities between 1 and 6 kW m^{-2}. Several parameters of influence are examined, including the number of chips and their arrangement, obtaining correlations of the type Nusselt–Rayleigh–Fourier.

The present work complements the earlier particular study [15] in which correlations of Nusselt-Rayleigh type were proposed for

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Nomenclature

a	thermal diffusivity of the air ($m^2\ s^{-1}$)
$c^{iso}_{\alpha,Ra}$	exponent of *Fo* number in Eq. (16) for a combination (α,*Ra*) (−)
c^{iso}_{α}	exponent of *Fo* number in Eq. (19) for an angle α (−)
C_p	specific heat at constant pressure ($J\ kg^{-1}\ K^{-1}$)
Fo	Fourier number (−)
g	acceleration of the gravity ($m\ s^{-2}$)
H	height of the cavity, right distance between the hot and cold walls (m)
$k(\alpha)$	coefficient of the correlations in Eqs. (15), (16) and (19) (−)
n	exponent of *Ra* number in Eqs. (15), (16) and (19) (−)
n_o	outgoing normal to the surface of the upper and lower passive walls
n_o^*	dimensionless normal to the upper and lower passive walls; $n_o^* = n_o/H$
nm	number of mesh elements on the hot wall (−)
$Nu^{iso}_{\alpha,Ra}$	local transient Nusselt number at the hot wall for a combination (α,*Ra*) (−)
$\overline{Nu}^{iso}_{\alpha,Ra}$	average transient Nusselt number at the hot wall for a combination (α,*Ra*) (−)
$\underline{\overline{Nu}}^{iso}_{\alpha}$	average Nusselt number at the hot wall for an angle α at steady state (−)
$\overline{Nu}^{iso}_{\alpha}$	average transient Nusselt number at the hot wall for an angle α (−)
p	pressure (Pa)
p^*	dimensionless pressure (−)
Pr	Prandtl number (−)
Ra	Rayleigh number (−)
S_h	total area of the hot wall (m^2)
$S_{h,j}$	area of the *j*th element of the hot wall (m^2)
t^*	dimensionless time (−)
T	local temperature (K)
T_c,T_h	temperature of the cold and hot walls respectively (K)
T_0	initial uniform temperature of the whole system (K)
T^*	dimensionless temperature (−)
u,v	flow velocity components in x and y directions respectively ($m\ s^{-1}$)
u^*,v^*	dimensionless flow velocity components in x and y directions respectively (−)
W	depth of the cavity (m)
x,x',y	Cartesian coordinates (m)
x^*,y^*	dimensionless Cartesian coordinates (−)
Greek symbols	
α	inclination angle of the cavity (°)
β	volumetric expansion coefficient of the air (K^{-1})
δ	deviation between $c^{iso}_{\alpha,Ra}$ and c^{iso}_{α}, defined in Eq. (18) (%)
λ	thermal conductivity of the air ($W\ m^{-1}\ K^{-1}$)
μ	dynamic viscosity of the air (Pa s)
ρ	density of the air ($kg\ m^{-3}$)

the quantification of steady state convective heat transfer. It concerns the case of active hot and cold walls that are maintained vertical and isothermal at different temperatures and connected by a closing channel which is adiabatic. The results are obtained numerically by means of the finite volume method. In the present study, a wide range of Rayleigh number is treated, varying between 1.84×10^5 and 1.70×10^9. Several inclination angles between −60° and +60° are considered according to the envisaged applications. The mathematical model is validated by measurements that were taken on specific experimental configurations presented in previous studies [1,10,15,16]. Several calculations were performed in transient regime up to reach steady state and the results are close to those obtained with direct steady state calculations. The study of thermal states obtained at particular times of the transient regime have led to interesting conclusions and allow to attain the *Nu*–*Ra*–*Fo* correlations that can be used in thermal engineering applications such as aeronautics, building, solar energy, or power electronics.

2. The treated case

This study concerns the free convection taking place in the air-filled closed cavity of parallelogrammic section presented in Fig. 1(a). The active hot and cold walls of height and depth H and W respectively, are separated by a horizontal distance H and remain always in vertical position. They are maintained isothermal at temperatures T_h and T_c respectively. The channel of the cavity is considered as adiabatic. Top and bottom walls of this channel are inclined at an angle α with respect to the horizontal. For positive angles, the hot wall is below the level of the cold one. In this situation, the convective flow taking place in the cavity is enhanced by the geometry, in comparison with that of the right cavity ($\alpha = 0$). On the contrary, convection is reduced when α is negative. This feature as "conducting" or "insulating" cavity, in the convective sense of the term, can be skillfully profited for engineering applications depending on the intended purpose. Given the thermal conditions treated and that the depth of the cavity W is large compared with H, the flow can be considered as 2D. The simplified model adopted is sketched in Fig. 1 (b). The validity of these assumptions has been verified in Refs. [1,10,15,16]. Certainly the dynamic and thermal aspects of this problem are closely linked, but the present study is only focused on the heat transfer taking place at the hot active wall.

3. Governing equations. Numerical solution

The continuity equation for the considered 2D problem is

$$\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} = 0 \tag{1}$$

where x^* and y^* are the dimensionless Cartesian coordinates and u^* and v^* the dimensionless velocity components defined as

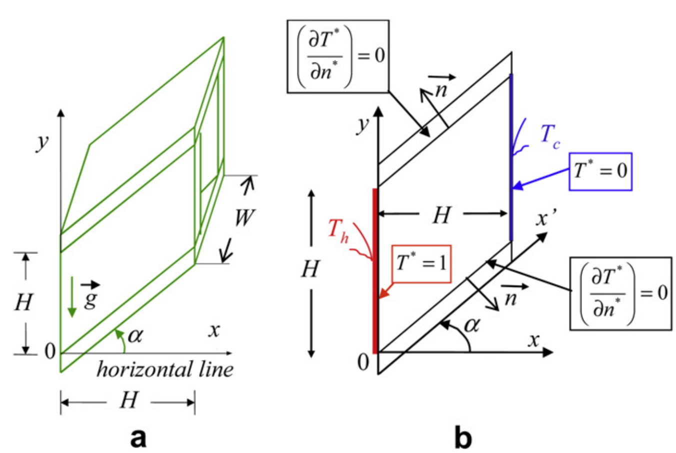

Fig. 1. The treated cavity.