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Infrared techniques for natural convection investigations in channels between two vertical, parallel, isothermal and symmetrically heated plates

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ABSTRACT

The effect of the gap width between two symmetrically heated vertical, parallel, isothermal plates on intensity of natural convective heat transfer in a gas ($\boldsymbol{Pr}$ = 0.71) was experimentally studied using the balance and gradient methods. In the former method heat fluxes were determined based on measurements of the voltage and electric current supplying the heaters placed inside the walls. In the latter, heat fluxes were calculated from the temperature distribution in the air in the plane perpendicular to the surface of the heating plates. Temperature fields were visualised using a thermal imaging camera. The analysis was conducted on two parallel vertical plates of height $\boldsymbol{H}$ = 0.5 m and width $\boldsymbol{B}$ = 0.25 m with the heated surfaces facing each other. Vertical planes with peripherally open channels and three different distances $\boldsymbol{s}$ = 0.045, 0.08 and 0.18 m were created this way. The surface temperature of the heating plates $\boldsymbol{t}_w$ was changed every 5 K and set at $\boldsymbol{t}_w$ = 30, 35, 40, 45, 50, 55, 60, 65, 70, 75 and 80 °C, while the ambient temperature range was from 18 to 25 °C.

1. Introduction

Heat transfer by natural convection is not only a theoretically difficult research problem; it is also a tricky one to tackle experimentally. Measurements of temperature, velocity and heat fluxes, are very hard to make owing to their very small and continually changing values in moving convective streams. However, this method of heat transfer is attractive because it is reliable, simple and cost-effective, especially in construction, electronics and power engineering. Natural convection, on the other hand, despite the low intensity of heat exchange due to the large amount of heat transferred from the surfaces of buildings, industrial facilities, power transmission lines, etc., causes gigantic energy losses. An important, but rarely studied research topic relating to naturally occurring convection is heat transfer within a channel formed by two vertical walls. We come across this in heating technology (radiators), buildings and electronics, energetics, household devices, and many other situations.

Natural convection in a vertical plane channel is not an unequivocally defined problem [1,2], as the following configurations of heat transfer within the channel can occur:

- vertical cavity (by Rayleigh-Bénard convection in a closed space) [3-7,49],
- vertical flat gap partially heated [48] or opened (from the bottom, top or from all sides) [8,9],
- vertical natural convection arrays for the following conditions:
  asymmetric (hot-cold, hot-adiabatic, warm-hot) isoflux [18,21,40,41,46,48] or isothermal heating plates [10–16,21,40,41,47,50,52,59]
  symmetric: isoflux [40,41,51] or isothermal heating plates [12,21,40,41,45],
- vertical plane channel with different wall temperature (hot-cold, hot-adiabatic, warm-hot) [16,17,26]
- vertical plane with an open-ended channel and isothermal, symmetrically heated walls [19–21],

Any of these configurations can be investigated theoretically (analytically: [21,38,40], numerically: [7,11,14,19,20,24,26–28,32, 35–37,48,50,52,57,58]), experimentally: [7,8,12,14,16,18,20,29–31, 35,39,52,59] or tested by visual methods [42,43,49,55].

Within the context of the above division, the current work concerns the experimental and visual study of natural convection heat transfer within a vertical plane channel formed by two vertical, parallel, isothermal, symmetrically heated walls.

The first results of an experimental and theoretical study of natural convective heat transfer in a vertical channel were published

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Nomenclature

$\boldsymbol{a}$	thermal diffusivity, m^2/s
$\boldsymbol{b}$	half-width of channel, $\equiv \boldsymbol{s}/2$, m
$\boldsymbol{A}$	area, $\equiv 2\boldsymbol{HB}$, m^2; coefficient
$\boldsymbol{B}$	plate width, m
$\boldsymbol{C}$	coefficient
$\boldsymbol{g}$	acceleration due to gravity, m/s^2
$\boldsymbol{H}$	channel or vertical plate length, m
$\boldsymbol{I}$	amperage of electric current, A
$\boldsymbol{m}$	exponent in Eqs. (2) and (5)
$\boldsymbol{Nu}_b$	channel balance Nusselt number, $\equiv \boldsymbol{UI}/(\lambda(\boldsymbol{t}_w - \boldsymbol{t}_\infty)\boldsymbol{B})$ dimensionless
$\widetilde{\boldsymbol{Nu}}$	modified Nusselt number, $\equiv \boldsymbol{UIs}/(\lambda(\boldsymbol{t}_w - \boldsymbol{t}_\infty)\boldsymbol{H}^2)$ dimensionless
$\boldsymbol{Nu}_y$	local Nusselt number, $\equiv \boldsymbol{\alpha}\, \boldsymbol{y}/\boldsymbol{\lambda}$ dimensionless
$\boldsymbol{Nu}$	total Nusselt number, Eq. (25), dimensionless
$\boldsymbol{Nu}*$	total channel Nusselt number, $\equiv \boldsymbol{\alpha}\, \boldsymbol{s}/\boldsymbol{\lambda}$, dimensionless
$\boldsymbol{P}$	plate/air parameter in Eq. (22), $\equiv \boldsymbol{Ra}/\boldsymbol{H}^4$, m^{-4}
$\boldsymbol{q}$“	heat flux, $\equiv \boldsymbol{q}/\boldsymbol{A}$, W/m^2
$\boldsymbol{Ra}_b$	Rayleigh number, $\equiv \boldsymbol{g\beta\Delta tH}^3/(\boldsymbol{va})$, dimensionless
Ra^*	channel Rayleigh number, $\equiv \boldsymbol{g\beta\Delta ts}^3/(\boldsymbol{va})\cdot\boldsymbol{s}/\boldsymbol{H}$, dimensionless
$\widetilde{Ra}$	modified Rayleigh number, $\equiv \boldsymbol{g\beta\Delta tb}^3/(\boldsymbol{va})\cdot\boldsymbol{H}/\boldsymbol{b}$, dimensionless
$\boldsymbol{s}$	plate spacing, m
$\boldsymbol{t}$	temperature, °C
$\boldsymbol{U}$	voltage of electric current, V
$\boldsymbol{x,y,z}$	coordinates, m

Greek letters

$\boldsymbol{\alpha}$	heat transfer coefficient, Eq. (13) $W/(m^2\,K)$
$\boldsymbol{\beta}$	expansivity coefficient, 1/K
$\boldsymbol{\lambda}$	thermal conductivity, W/(m K)
$\boldsymbol{v}$	kinematic viscosity, m^2/s
$\boldsymbol{\rho}$	density, kg/m^3
∞	entrance or ambient value

Subscripts

a	auxiliary
av	average
b	balance method
L	left
m	main
R	right
w	wall
y	local

in 1942 by Elenbass [21], who carried out investigations on square vertical plates. By analysing a simplified set of equations, and by adjusting constants to fit experimental data, he proposed the following equation for the Nusselt number as a function of the modified Rayleigh number $\boldsymbol{Ra}^*$, called the Elenbass Rayleigh number:

$$Nu_0 = \frac{Ra^*}{24}\cdot\left(1 - e^{-35/Ra^*}\right)^{3/4} \tag{1}$$

This equation, confirmed by experimental studies, is valid for the range $10^{-1} < \boldsymbol{Ra}^* < 10^5$ and for fairly short vertical plates in air. In addition, for $10^4 < \boldsymbol{Ra} < 10^9$ it has two asymptotes: one for small values of the $\boldsymbol{s}/\boldsymbol{H}$ ratio (fully developed flow regime) and the other for large values of $\boldsymbol{s}/\boldsymbol{H}$ (boundary layer flow regime).

Further studies of natural convection in vertical open channels were published by Raithby and Hollands [22,23], and Aung et al. [17], who derived a different relation that also captures both limiting cases $\boldsymbol{s} \to 0$ and $\boldsymbol{s} \to \infty$. After modifying the relation (1), they obtained a new version of the Elenbass equation:

$$Nu_0 = \left(Nu_{fd}^m + Nu_{bl}^m\right)^{1/m}; \quad m = -1.9, \tag{2}$$

where Nu_{bl} is the Nusselt number for the boundary layer regime near the entrance and $\boldsymbol{Nu}_{fd}$ is the Nusselt number for fully developed flow throughout the flow passage along the greater part of the channel.

$$Nu_{bl} = 0.62\cdot(Ra^*)^{1/4}, \quad \text{for } b \to \infty \tag{3}$$

$$Nu_{fd} = Ra^*/24, \quad \text{for } b \to 0, \tag{4}$$

A comparison of Eqs. (1) and (2) with Sparrow and Bahrami's experimental data [8] and Ormiston's numerical solutions [24] is given in [25].

Further research by Churchill and Usagi [45], performed for natural convective heat transfer within a vertical channel with isothermal, symmetrically heated rectangular plates, led to an equation similar to but simpler than (2):

$$Nu_0 = \left[\left(\frac{Ra^*}{24}\right)^{-m} + \left(0.59\sqrt[4]{Ra^*}\right)^{-n}\right]^{-1/m}, \tag{5}$$

which is valid for fairly short vertical plates in air and when $10^4 < \boldsymbol{Ra} < 10^9$.

From the correlating procedure described by Churchill and Usagi [45], the exponent m in Eq. (5) is equal to approximately 2, so the relationship for the vertical channel with two isothermal, symmetrically heated surfaces takes the form:

$$Nu_0 = \left[\left(\frac{576}{Ra^*}\right)^2 + \frac{2.873}{\sqrt{Ra^*}}\right]^{-1/2}, \tag{6}$$

In turn, Martin et al. [25] focused on short, wide channels $\boldsymbol{H}/\boldsymbol{b} \le 10$, in which the proportion of fully developed natural convection is larger and interaction with the region below the bottom inlet into the channel cannot be neglected. As a result they proposed a modified version of the relationship between the Nusselt and Rayleigh numbers for the fully developed regime. It can be written thus:

$$\widetilde{Nu}_{fd} = \frac{\widetilde{Ra}}{6}\cdot\left(1 + \sqrt{1 + \frac{12}{\widetilde{Ra}}}\right), \tag{7}$$

with two asymptotes:

$$\widetilde{Nu}_{fd} = \sqrt{\frac{\widetilde{Ra}}{3}} \quad \text{for } \widetilde{Ra} \to 0, \tag{8}$$

$$\widetilde{Nu}_{fd} = \frac{\widetilde{Ra}}{3} \quad \text{for } \widetilde{Ra} \to \infty, \tag{9}$$

Eq. (8) represents a new asymptote that accounts for the effect of upstream conditions. Because of the transition to the boundary layer regime, the range of Ra over which the Elenbass asymptote is valid is limited. Eq. (9) is an already known asymptote of the Elenbass Eqs. (1) and (4).