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Natural convection in symmetrically heated vertical channels

Witold M. Lewandowski*, Michał Ryms, Hubert Denda

Faculty of Chemistry, Department of Chemical Apparatus and Theory of Machines, Gdansk University of Technology, ul. G. Narutowicza 11/12, 80-233, Gdansk, Poland

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ABSTRACT

The study describes natural convection through a vertical channel, open on four sides and bound by two isothermal walls. Both balancing and infrared experimental investigations were performed in air ($\boldsymbol{Pr}$ = 0.71) to estimate the impact of channel width $\boldsymbol{s}$ and wall-to-ambient temperature difference ($\boldsymbol{t}_{\mathrm{w}}$ - $\boldsymbol{t}_{\infty}$) on natural convective heat transfer, the formation of air flow patterns and the rate of flow pattern formation. The study 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, thus creating peripherally open channels of different widths $\boldsymbol{s}$ = 0.045, 0.08, 0.180 and ∞ m. The surface temperature $\boldsymbol{t}_{\mathrm{w}}$, identical for both heating plates, was changed every 10 K and set at $\boldsymbol{t}_{\mathrm{w}}$ = 40, 50, 60, 70 and 80 °C, while the ambient temperature was maintained within the 18–25 °C range. In the balance method, heat fluxes were determined based on measurements of voltage and electric current supplying the heaters placed inside the walls. In the gradient method, the heat fluxes were calculated from the temperature distribution in air, within a plane perpendicular to the heating plate surfaces. Temperature fields were visualized using a plastic detecting mesh and a thermal imaging camera. The distribution of temperature $\boldsymbol{t}_{\mathrm{x,y}}$, and its gradient at the walls $\mathrm{d}\boldsymbol{t}/\mathrm{d}\boldsymbol{x}|_{\boldsymbol{x}=0,\boldsymbol{y}}$ were obtained at different heights $\boldsymbol{y}$ along the channels. The gradient values obtained and the results, presented as dependencies of the Nusselt number on the Rayleigh number, indicate that the channel width has a significant impact on heat transfer. Compared to the vertical plate $\boldsymbol{s}$ = ∞, the following levels of convective heat transfer enhancement were observed: 29.5% ($\boldsymbol{s}$ = 0.045 m), 38.8% ($\boldsymbol{s}$ = 0.085 m) and 61.6% ($\boldsymbol{s}$ = 0.180 m).

1. Introduction

The research resulting in this study was inspired by questions which arose during the construction of convector plate air heaters, illustrated in Fig. 1.

The most important of these questions is whether increasing the width of the plate spacing enhances (in for example radiators, cooling of integrated circuits, reflectors) or inhibits (for transparent insulations, fleece clothing, etc.) convective heat transfer and what the optimal spacing of these plates for such applications should be. Prior to exploring this problem, it was necessary to locate it within the map of multiple convective heat exchange cases. These include convective heat transfer in an open space, taking place from flat (vertical, diagonal, horizontal) [1-3,[1-3], cylindrical (horizontal, diagonal, vertical) [4-6], spherical and complex surfaces [7-9]. An equally important issue is natural convection in a closed space, occurring inside cylindrical, spherical or cuboid ducts [10]. In the latter case, the ambient thermodynamic conditions neither intensify nor inhibit the phenomena occurring inside the channels. However, if these channels are open on one side (top, bottom, front or rear), on two sides (rear and front or top and bottom) or on all four sides, there is an indirect case of natural convection, formally occurring in a closed space but simultaneously inhibiting or intensifying the impact of the environment.

Despite its many practical applications, e.g. in construction (convector heaters, air heaters, coolers) or electronics (heatsinks, components and electronic components), this case has not been tested very often, as evidenced by the next section of this study, which analyzes papers on convective heat exchange in channels.

2. Literature review

Depending on the configuration of the boards and their temperature in convective heat transfer in channels, the following cases can be specified [11,12]:

- convection in a vertical channel due to two- or three-dimensional cocalized heat source [13] or different configurations of open channel (at the top/bottom or on all sides) [14-16],
- convection in a vertical gap for the following conditions:
  - symmetric: isoflux [17,18], or isothermal heating plates [17,19-

* Corresponding author.
E-mail addresses: wlew@pg.edu.pl, wlew2@o2.pl (W.M. Lewandowski).

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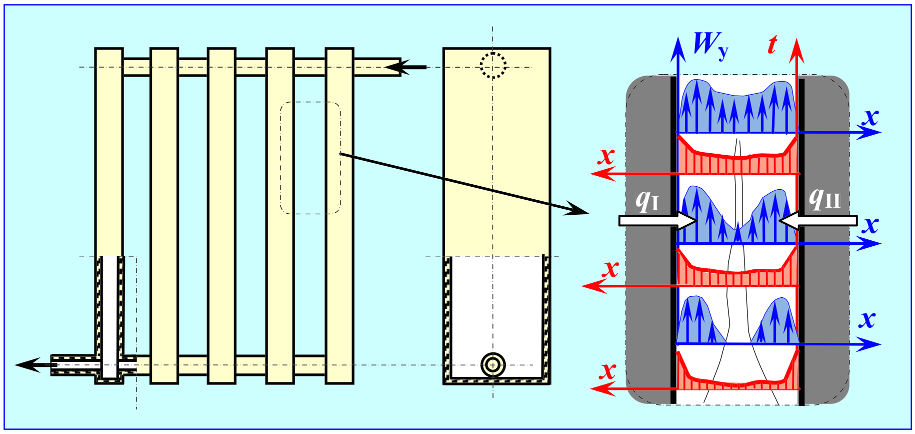

Fig. 1. Diagram of a multi-plate convector with natural convective temperature and velocity fields, generated in air between two adjacent plates.

22],
- asymmetric (hot-cold, hot-adiabatic, warm-hot) isoflux [13,17,23-25], or isothermal heating plates [17,24,26-35],
- vertical plane with an open-ended channel and isothermal, symmetrically heated walls [24,36,37],
- vertical plane channel with different wall temperatures (hot-cold, hot-adiabatic, warm-hot) [31,38,39,43],

Any of these configurations can be investigated theoretically (analytically: [17,24,40], numerically: [13,15,30,33,34,36,37,39,41,44-46,47,48-50]), experimentally: [14,28,30,32,34,37,46,51-54], or tested by visual methods [55,56,57,58].

In context of the above-mentioned division of cases of convective heat transfer from parallel, vertical plates heated symmetrically or asymmetrically, with a constant surface temperature or a constant heat flow (isoflux plates), the case investigated in this paper regarding a plate heat exchanger (Fig. 1) should be classified as a vertical configuration of isothermal, parallel heating surfaces, the distance between which is $\boldsymbol{s}$.

The history of free convection testing for this configuration is summarized in Table 1. Research started in 1942 by Elenbass [24], and continued by Sparrow, Bahrami [14], Ormiston [41], Martin, Raithby et al. [42]. and Churchill, Usagi [21], eventually led Bar-Cohen et al. [23]. to a theoretical determination of the optimal plate spacing $\boldsymbol{s_{opt}}$, at which the amount of heat transferred from them ($\boldsymbol{Nu_{opt}}$) reaches the maximum value.

Table 1 contains Nusselt-Ryleigh relations for the convective heat transfer in vertical channels created by two isothermal plates with a hight of $\boldsymbol{H}$, width of $\boldsymbol{B}$, and the distance between them $\boldsymbol{s}$.

The optimal width $\boldsymbol{s}_{\rm opt}$ of the channel between the plates provided by Bar-Cohen et al. [17] in Table 1:

$$\boldsymbol{s}_{\rm opt} = \frac{2.714}{\boldsymbol{P}^{1/4}} \tag{1}$$

is related to parameter $\boldsymbol{P}$, which can be transformed into:

$$\boldsymbol{P} = \frac{\boldsymbol{g}\cdot\boldsymbol{\beta}\cdot\boldsymbol{\Delta t}}{\boldsymbol{a}\cdot\boldsymbol{\nu}\cdot\boldsymbol{H}}\cdot\frac{\boldsymbol{s}^4}{\boldsymbol{s}^4} = \frac{\boldsymbol{Ra}'}{\boldsymbol{s}^4} \tag{2}$$

Substituting (2) into (1) returns the relation:

$$\boldsymbol{s}_{\rm opt} = \frac{2.714}{\boldsymbol{P}^{\frac{1}{4}}} = \frac{2.714\cdot\boldsymbol{s}_{\rm opt}}{\boldsymbol{Ra}'^{\frac{1}{4}}},\ \text{hence: } \boldsymbol{Ra}'_{\rm opt} = 2.7144 = 55.255 \tag{3}$$

Table 1
A summary of the most important research on natural convection from vertical, isothermal, symmetrically heated plates forming open, parallel channels. Results of these studies allowed the formulation of the relations describing this phenomenon.

Range of research	Obtained solution	References
Elenbass the first experimental and theoretical study of natural convective heat transfer in a vertical channel and in air (1942).	$\boldsymbol{Nu}^* = \frac{\boldsymbol{Ra}^*}{24}\frac{\boldsymbol{s}}{\boldsymbol{H}}\cdot\left(1 - e^{-\frac{35}{\boldsymbol{Ra}^*}\frac{\boldsymbol{H}}{\boldsymbol{s}}}\right)^{3/4}$ with two asymptotes: $\boldsymbol{Nu}^* \to \frac{\boldsymbol{Ra}^*}{24}\frac{s}{H}$ for $\boldsymbol{Ra}^* \to 0$ and $\boldsymbol{Nu}^* \to 0.6\left(\boldsymbol{Ra}^*\frac{s}{H}\right)^{1/4}$ for $\boldsymbol{Ra}^* \to \infty$ where: $\boldsymbol{Nu}^* = \frac{\boldsymbol{h}\cdot\boldsymbol{s}}{\boldsymbol{\lambda}}$ and $\boldsymbol{Ra}^* = \boldsymbol{g}.\boldsymbol{\beta}.\boldsymbol{\Delta t}.\boldsymbol{s}^3/(\boldsymbol{\nu}.\boldsymbol{a})$	[24]
Modification of the Elenbass relation by introducing two cases: "fully developed flow " $\boldsymbol{s}\to 0$, and "limiting or boundary layer flow regime" $\boldsymbol{s}\to\infty$	$\boldsymbol{Nu}_0 = \left(\boldsymbol{Nu}_{fd}^m + \boldsymbol{Nu}_{bl}^m\right)^{1/m}$ where: $\boldsymbol{m} = -1.9$ and $\boldsymbol{Nu}_{\rm fd} = \boldsymbol{Ra}^*/24$ for $\boldsymbol{s} \to 0$ and $\boldsymbol{Nu}_{\rm bl} = 0.62.(\boldsymbol{Ra}^*)^{1/4}$ for $\boldsymbol{s}\to\infty$ where: $\boldsymbol{Nu}_0 = \boldsymbol{U}.\boldsymbol{I}.\boldsymbol{s}/(\boldsymbol{B}.\boldsymbol{H}.\boldsymbol{\lambda}.\boldsymbol{\Delta t})$	[38]
Further experimental, numerical and analytical research of the above-mentioned problem was focused on fully developed regime	$\widetilde{\boldsymbol{Nu}}_{\rm fd} = \frac{\widetilde{\boldsymbol{Ra}}}{6}\cdot\left(1 + \sqrt{1 + \frac{12}{\widetilde{\boldsymbol{Ra}}}}\right)$ with two asymptotes: $\widetilde{\boldsymbol{Nu}}_{\rm fd} = \sqrt{\frac{\widetilde{\boldsymbol{Ra}}}{3}}$ for $\widetilde{\boldsymbol{Ra}} \to 0$ and $\widetilde{\boldsymbol{Nu}}_{\rm fd} = \frac{\widetilde{\boldsymbol{Ra}}}{3}$ for $\widetilde{\boldsymbol{Ra}} \to \infty$where: $\widetilde{\boldsymbol{Nu}} = \frac{\boldsymbol{U}\cdot\boldsymbol{I}\cdot\boldsymbol{H}}{\boldsymbol{B}\cdot\boldsymbol{\Delta t}\cdot.\boldsymbol{\lambda}\cdot\boldsymbol{s}}$ and $\widetilde{\boldsymbol{Ra}} = \frac{\boldsymbol{g}\cdot\boldsymbol{\beta}\cdot\boldsymbol{\Delta t}}{\boldsymbol{\nu}\cdot\boldsymbol{a}\cdot\boldsymbol{H}}\cdot\left(\frac{\boldsymbol{s}}{2}\right)^2$	[14], [41], [42]
Vertical open channels with symmetrically heated, rectangular short plates, air and $10^4 < \boldsymbol{Ra} < 10^9$	$\boldsymbol{Nu}_0 = \left[\left(\frac{\boldsymbol{Ra}'}{24}\right)^{-m} + (0.59\sqrt[4]{\boldsymbol{Ra}'})^{-m}\right]^{-1/m}$ where: $\boldsymbol{Ra}' = \frac{\boldsymbol{g}\cdot\boldsymbol{\beta}\cdot\boldsymbol{\Delta t}\cdot\boldsymbol{s}^4}{\boldsymbol{\nu}\cdot\boldsymbol{a}\cdot\boldsymbol{H}}$.	[21]
Theoretically optimizing the distance between heating plates in a multi-plate heater $\boldsymbol{s}_{\rm opt}$ for an optimal value of the Nusselt number $\boldsymbol{Nu}_{0,\rm opt}$.	$\boldsymbol{Nu}_0 = \left[\frac{576}{\boldsymbol{Ra}'^2} + \frac{2.873}{\sqrt{\boldsymbol{Ra}'}}\right]^{-1/2}$ $\boldsymbol{s}_{\rm opt} = 2.714\,\boldsymbol{P}^{-1/4}$ where: $\boldsymbol{P} = \frac{\boldsymbol{g}\cdot\boldsymbol{\beta}\cdot\boldsymbol{\Delta t}}{\boldsymbol{\nu}\cdot\boldsymbol{a}\cdot\boldsymbol{H}}$ and $\boldsymbol{Nu}_{0,\rm opt} = 1.31$	[17]