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Fluid flow and heat transfer of natural convection adjacent to upward-facing, rectangular plates of arbitrary aspect ratios

K. Kitamura [a,*], A. Mitsuishi [a], T. Suzuki [a], F. Kimura [b]

[a] Department of Mechanical Engineering, Toyohashi University of Technology, 1-1 Hibarigaoka, Tempaku-cho, Toyohashi, Aichi 441-8580, Japan
[b] Department of Mechanical Engineering, Faculty of Engineering, University of Hyogo, 2167 Shosha, Himeji, Hyogo 671-2201, Japan

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ABSTRACT

Natural convective flows adjacent to the upward-facing, heated rectangular plates were investigated experimentally. Main concerns were directed to the effects of aspect ratios on the turbulent transition of flows and also on the heat transfer from the plates. For the sake of this, rectangular plates having different width $W = 35–500$ mm and aspect ratios $AR = 1, 2, 3, 5, 8$, were fabricated and tested. The experiments began with the visualizations of the flow fields over the plates, then, the measurements of average Nusselt numbers followed. The flow visualizations depicted that the turbulent transition of flows takes place over the plates of $AR = 1, 2, 3, 5, 8$, when the width-based Rayleigh numbers beyond $Ra_W = 1.5 \times 10^6, 6.5 \times 10^5, 5.0 \times 10^5, 3.5 \times 10^5, 3.0 \times 10^5$, respectively. While, when the equivalent diameter of the plate d_e was adopted, the critical Rayleigh numbers become almost identical as $Ra_{de} = (1.5–1.7) \times 10^6$ regardless of aspect ratios. The average Nusselt numbers based on the plate width Nu_W were plotted with the Rayleigh numbers Ra_W. The plots showed gradual decrease with the aspect rations, in particular, in the low Rayleigh number region. The above Nusselt and Rayleigh numbers were next converted to those based on the equivalent diameter. Then, the Nusselt numbers Nu_{de} from the plates of different aspect ratios showed identical variations with the Rayleigh numbers Ra_{de}. Based on these results, empirical correlations for the laminar and turbulent heat transfer were proposed.

1. Introduction

Natural convective flows induced over upward-facing, heated horizontal plates have been the subject of numerous analytical and experimental investigations. This is not only because the horizontal plates are one of the most fundamental geometry for the study of natural convection, but also because the flows are encountered in a wide variety of engineering applications. These applications together with the simple geometry have motivated a considerable body of research on the natural convection over heated plates. Table 1 summarizes prior heat transfer experiments carried out with upward-facing, isothermal plates of various planforms, where the types of experiment, measured quantities, test fluid, size and shape of test plates and proposed heat or mass transfer correlations are listed together with the experimental ranges of Rayleigh numbers.

Among these experiments, the heat transfer experiment by Fishenden and Saunders [1] may be the first work that has correlated the average Nusselt numbers with the form

$$Nu = CRa^n \tag{1}$$

They proposed the following sets of constant C and exponent n of the correlation as, $C = 0.54$, $n = 1/4$ for the laminar regime, $10^5 < Ra_W < 2 \times 10^7$, and $C = 0.14$, $n = 1/3$ for turbulent regime, $2 \times 10^7 < Ra_W < 3 \times 10^{10}$. Where, the side length of square plate W was adopted as a characteristic length in the Nusselt and Rayleigh numbers. While, Goldstein et al. [2] have carried out the mass transfer experiments using naphthalene sublimation technique and have obtained the average Sherwood numbers from the circular, square and rectangular plates of 7:1 aspect ratio. They found that the Sherwood numbers from the above plates become identical if the length scale $L^* = A/P$ is adopted as a characteristic length in Sherwood and Rayleigh numbers, where A and P stands for the surface area and perimeter of the plates, respectively. Lloyd and Moran [3] have measured the average Sherwood numbers from the circular, square, rectangular and triangular plates using electrochemical technique. They also reported that the data for all planforms are reduced to a single correlation if the above length scale L^* is adopted as a characteristic length, and 1/4- and 1/3-power correlations hold for the laminar and turbulent flows.

* Corresponding author. Tel.: +81 532 44 6666; fax: +81 532 44 6661.
E-mail address: kitamura@me.tut.ac.jp (K. Kitamura).

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Table 1
Prior experiments on natural convection adjacent to upward-facing isothermal plates.

Worker(s) (year)	Ref.	Experiments and measured quantities	Test fluid	Shape and size of test plate	Proposed correlations for average *Nu* or *Sh* numbers (ranges of Ra numbers)
Fishenden and Saunders (1950)	[1]	Heat transfer exp. Average *Nu* No.	Air	Square: max 600×600 mm^2	Laminar: $Nu_W = 0.54Ra_W{}^{1/4}$ ($10^5 < Ra_W < 2 \times 10^7$) turbulent: $Nu_W = 0.14Ra_W{}^{1/3}$ ($2 \times 10^7 < Ra_W < 3 \times 10^{10}$)
Goldstein et al. (1973)	[2]	Naphthalene sublimation exp. Average *Sh* No.	Naphthalene in air ($Sc = 2.5$)	Circular: $d = 12.7$–203 mm Square: $W = 12.7$–202 mm Rectangular: $W = 20.3$–58.4 mm, $AR = 7$	$Sh_{L*} = 0.96Ra_{L*}^{1/6}$ ($1 < Ra_{L*} < 200$) $Sh_{L*} = 0.59Ra_{L*}^{1/4}$ ($200 < Ra_{L*} < 6 \times 10^3$) where, $L^* = A/P$
Lloyd and Moran (1974)	[3]	Electrochemical mass transfer exp. Average *Sh* No.	Cupric sulfate in sulfuric acid ($Sc = 2200$)	Circular: $d = 3.16$–101.6 mm Square: $W = 6.35$, 12.7 and 101.6 mm Rectangular: $W \times L = 12.7 \times (25.4$–$127)$ mm^2, 25.4×157 mm^2, 51.8×127 mm^2	Laminar: $Sh_{L*} = 0.54Ra_{L*}^{1/4}$ ($2.2 \times 10^4 < Ra_{L*} < 8 \times 10^6$) turbulent: $Sh_{L*} = 0.15Ra_{L*}^{1/3}$ ($8 \times 10^6 < Ra_{L*} < 5 \times 10^9$) where, $L^* = A/P$
Al-Arabi and El-Riedy (1976)	[4]	Heat transfer exp. Average *Nu* No.	Air	Circular: $d = 100$–500 mm Square: $W = 50$–450 mm Rectangular: $W \times L = 150 \times (150$–$600)$ mm^2	(Square) $Nu_W = 0.70Ra_W{}^{1/4}$ ($2 \times 10^5 < Ra_W < 4 \times 10^7$) $Nu_W = 0.155Ra_W{}^{1/3}$ ($4 \times 10^7 < Ra_W < 8 \times 10^8$) (Rectangular) $Nu_W \approx 40$ at $Ra_W \approx 2 \times 10^7$
Yousef et al. (1982)	[5]	Heat transfer exp. Local and Average *Nu* No. Temperature distr. in boundary layer	Air	Square: $W = 100$, 200 and 400 mm	Laminar: $Nu_W = 0.622Ra_W{}^{1/4}$ ($3 \times 10^6 < Ra_W < 4 \times 10^7$) turbulent: $Nu_W = 0.162Ra_W{}^{1/3}$ ($4 \times 10^7 < Ra_W < 2 \times 10^8$)
Goldstein and Lau (1983)	[6]	Naphthalene sublimation exp. Average *Sh* No.	Naphthalene in air ($Sc = 2.5$)	Square: $W = 25.8$ mm to 203 mm	$Sh_W = 1.30Ra_W{}^{1/5}$ ($640 < Ra_W < 3.07 \times 10^5$)
Lewandowski et al. (2000)	[7]	Heat transfer exp. Average *Nu* No. Flow Visualization	Water	Rectangular: $W \times L = 70$ mm (fixed) $\times$ (70–326.2) mm	$Nu_W = C(A)Ra_W{}^{1/5}$ $C(A) = 1.138$–1.080 depend on aspect ratios ($6 \times 10^5 < Ra_W < 6 \times 10^7$)
Martorell et al. (2003)	[8]	Heat transfer exp. Average *Nu* No.	Air	Rectangular: $W \times L = (10$–$90)$ mm $\times$ $(140$–$280)$ mm	$Nu_W = 1.23Ra_W{}^{0.173}$ independent of aspect ratios $AR = 2.33$–28 ($290 < Ra_W < 3.3 \times 10^5$)
Radziemska and Lewandowski (2005)	[9]	Heat transfer exp. Average *Nu* No.	Air and water	Rectangular: $W \times L = (5$–$100)$ mm $\times$ 100 mm (fixed)	$Nu_W = C(A)$. $Ra_W{}^{1/5}$ $C(A)$ depends on aspect ratio ($5 \times 10^4 < Ra_W < 3 \times 10^6$ for air) ($5 \times 10^4 < Ra_W < 2 \times 10^8$ for water)