parallel-plate channel and a circular tube, respectively, plotted against the axial distance in the range $10^{-4} \leqslant Z \leqslant 10^{-1}$ for several different values of the dimensionless time τ. Starting from the inlet region, the local Nusselt numbers decrease continuously with both increasing time and axial location along the conduit until the conduction region is reached. In the conduction region, the Nusselt number remains invariant with the position but decreases with increasing time. Eventually, with increasing time, the local Nusselt numbers for both regions assume the well-known steady-state value.

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Int. J. Heat Mass Transfer. Vol. 30, No. 8, pp. 1756–1758, 1987
Printed in Great Britain
0017–9310/87 $3.00+0.00
Pergamon Journals Ltd.

Natural convection heat transfer in enclosures with an off-center partition

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(*Received* 1 *December* 1986 *and in final form* 19 *January* 1987)

1. INTRODUCTION

NATURAL convection through externally heated and cooled enclosures is of interest in solar collector applications, in the estimation of heat losses from double-pane windows, and in the calculation of heat losses from rooms. Numerous experimental and numerical computational studies have been reported explaining the heat transfer mechanism and presenting correlations for heat transfer rates for such systems. Excellent reviews [1, 2] are available and there is no need to repeat them here.

The problem of primary interest in the literature [1, 2] is that of an enclosure with no partitions. However, in practical cases, a vertical partition is inserted into the enclosure to reduce heat losses by natural convection and thermal radiation. Reported studies of natural convection in a partitioned enclosure are limited. Duxbury [3] reported experiments with air-filled enclosures containing a central partition as shown in Fig. 1. Nakamura *et al.* [4] performed computational and experimental studies including the effect of thermal radiation for the same configuration as that of Duxbury. The present authors [5] proposed a boundary layer solution for this system and confirmed its validity by experiments. Tong and Gerner [6] reported the effect of partition position on the heat transfer rate by numerical computation and concluded that a central partition corresponding to $W'/W = 0.5$ produces the greatest reduction in heat transfer.

This study is an extension of the previous study [5]. We examine the limitations of the boundary layer approximation for various positions of the partition. We show that even if the partition deviates from the center of the enclosure, the heat transfer rate is identical with that for the partition in the central position. This does not appear to have been studied previously.

2. LIMITATIONS OF THE BOUNDARY LAYER APPROXIMATION

We [5] previously indicated that a thermal boundary layer with a constant thickness is developed along the partition at high Rayleigh numbers for the enclosure with a central

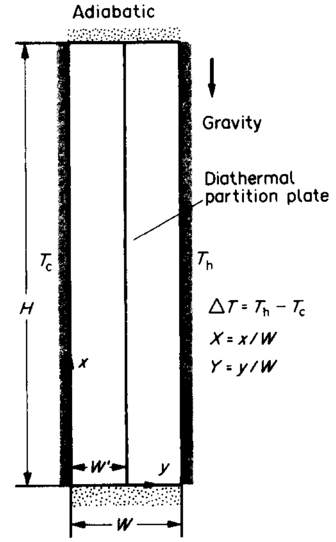

FIG. 1. Schematic diagram of an enclosure divided by a vertical partition.

NOMENCLATURE

g	gravitational acceleration [m s^{-2}]
H	height of the enclosure [m]
h	heat transfer coefficient, $q/(T_h - T_c)$ [W m^{-2} K^{-1}]
L	depth of the enclosure [m]
Nu	average Nusselt number, $h/W\lambda$ [—]
Pr	Prandtl number [—]
q	heat flux per unit area [W m^{-2}]
Ra	Rayleigh number, $g\beta(T_h - T_c)W^3/(\alpha\nu)$ [—]
T	temperature [K]
T_c	temperature at the cold wall [K]
T_h	temperature at the hot wall [K]
W	width of the enclosure [m]
W'	distance between the partition and the cold wall [m]
W_{min}	minimum width of cell satisfying the boundary layer approximation [m].

Greek symbols

α	thermal diffusivity [m^2 s^{-1}]
β	volumetric expansion coefficient [K^{-1}]
δ	thermal boundary layer thickness defined in Fig. 2 [m]
λ	thermal conductivity [W m^{-1} K^{-1}]
ν	kinematic viscosity [m^2 s^{-1}].

partition and presented the boundary layer solution predicting the heat transfer rate through the enclosure.

The temperature distribution proposed by the boundary layer model is shown in Fig. 2. The thickness of the boundary layer is defined as the distance within which the temperature near the partition reaches the core temperature as indicated in Fig. 2. The boundary layer thickness is given by

$$\delta/W = 2.64Ra^{-1/4}(H/W)^{1/4}. \quad (1)$$

In this study it is shown that the same relation is applied for certain values of the boundary layer thickness regardless of the position of the partition. That is, it may be considered from the boundary layer model that the heat transfer rate is independent of the position of the partition if the boundary layer thickness is less than the half-width of each cell constructed by the partition, i.e. $\delta < W'/2$ and $\delta < (W-W')/2$. Thus the minimum width of a cell satisfying the boundary layer approximation is derived by using equation (1)

$$W_{min}/W = 5.28Ra^{-1/4}(H/W)^{1/4}. \quad (2)$$

The validity of equation (2) is confirmed by experimental measurement and numerical computation as subsequently described.

3. EXPERIMENT

Since the experimental equipment and procedure have already been described in detail [7], they will be reviewed here only briefly. Two kinds of enclosure were used. The height and the length were fixed ($H = 300$ mm and $L = 200$ mm), and the width was variable ($W = 30$ and 75 mm). The vertical partition was made of aluminum foil, 15 μm in thickness, and its position was changeable ($W'/W = 0.166$–0.75). The working fluid was water ($Pr = 6$). The experiments were carried out in the range $10^6 < Ra < 10^8$.

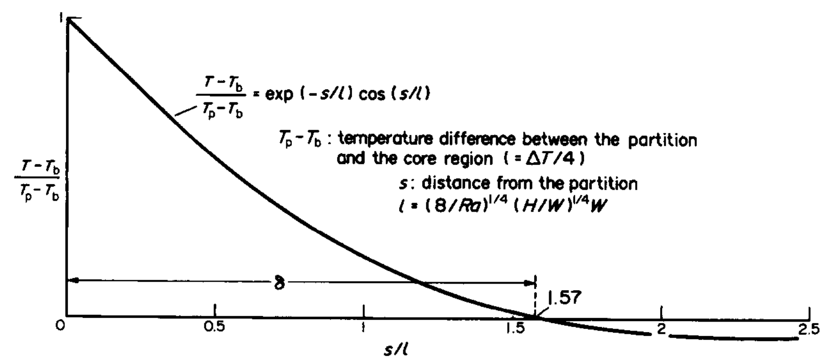

FIG. 2. Temperature profile near the partition proposed by the boundary layer model.

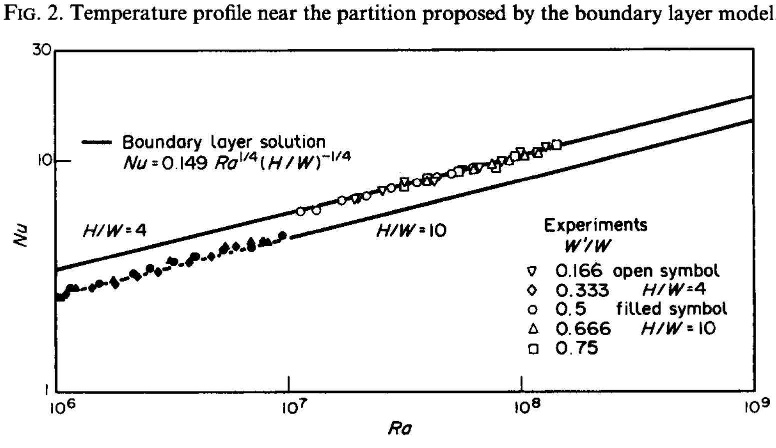

FIG. 3. Experimental Nusselt number vs Rayleigh number for various positions of the partition.

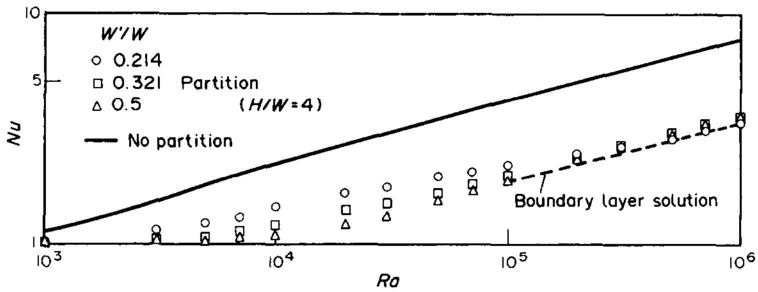

FIG. 4. Experimental Nusselt number vs Rayleigh number for various positions of the partition.

The results of the experiments are shown in Fig. 3 in which the Nusselt number is expressed as a function of the Rayleigh number. The Nusselt numbers for a given Rayleigh number are independent of various values of W'/W for the values of $H/W = 4$ and 10. Also, all of experimental data agree well with the boundary layer solution presented previously for the case of a central partition. Under these experimental conditions, the width of each cell constructed by the partition is larger than the minimum width calculated using equation (2), and thus the premise described in Section 2 is valid.

4. NUMERICAL COMPUTATION

A theoretical investigation was carried out to confirm the validity of equation (2) using a two-dimensional Galerkin finite element method. The solution technique is the same as that previously used [8]. The equations utilized were the Navier–Stokes and the energy transport equations. The computation was performed in the range $10^3 < Ra < 10^6$ for $Pr = 6$ and $H/W = 4$. Solutions were obtained for different positions of the partition from the center of the enclosure to the cold wall ($W'/W < 0.5$). Because the experimental Nusselt numbers are essentially identical whether the position of the partition is located at the cold or hot wall side as shown in Fig. 3.

Figure 4 shows the calculated results of the Nusselt number as a function of the Rayleigh number. In this case, Nusselt numbers for various values of W'/W deviate from the boundary layer solution, particularly for Rayleigh numbers less than 10^5. Equation (2) predicts that the boundary layer approximation is satisfied for W'/W larger than 0.418 at $Ra = 10^5$. This agrees with the results of the numerical computation at $Ra = 10^5$.

Thus the validity of equation (2) is considered to be confirmed both by experimental measurement and numerical computation as indicated above.

Acknowledgements—The authors acknowledge with thanks the assistance of Mr Masayuki Yamamoto with the experiments. This work was supported in part by a Grant-in-Aid for Scientific Research (No. 61750891) from the Ministry of Education, Science and Culture of Japan. We are also very grateful for the revision in the English text by Professor W. Hayduk of the University of Ottawa.

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Int. J. Heat Mass Transfer. Vol. 30, No. 8, pp. 1758–1762, 1987
Printed in Great Britain
0017-9310/87 $3.00+0.00

A generalized correlation for thermal design data of heat-pipe heat exchangers

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(*Received* 16 *May* 1986 *and in final form* 30 *January* 1987)

INTRODUCTION

HEAT EXCHANGERS made of heat pipes have attracted much attention in the application of economic devices for the recovery of waste heat energy [1–3]. Although the characteristics of thermal performance of a single heat pipe have been extensively studied and clearly understood during the past 20 years, the study of the overall performance of heat