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# NATURAL CONVECTION IN AN AIR LAYER ENCLOSED WITHIN RECTANGULAR CAVITIES

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**Abstract**—Natural convection in an air layer contained in a rectangular cavity having side walls of different temperatures has been investigated experimentally for various aspect ratios ranging from 4.9 to 78.7. Measured temperature profiles are analyzed with the flow regimes. Two Nusselt–Grashof correlations are presented for the measured heat-transfer data, fitting the data with an average deviation of less than 7.8%. The Grashof number based on layer thickness ranged from $1.5 \times 10^3$ to $7.0 \times 10^6$.

## NOMENCLATURE

| | |
|---|---|
| $A$, | surface area of cold or hot plate; |
| $a, b, C$, | empirical constants; |
| $D$, | thickness of air layer; |
| $g$, | local gravitational acceleration; |
| $Gr_D$, | Grashof number, $g\beta(\Delta T)D^3/v^2$; |
| $h_c$, | heat-transfer coefficient, $q/A(T_H - T_C)$; |
| $k$, | thermal conductivity; |
| $L$, | height of air layer; |
| $Nu_D$, | Nusselt number, defined by equation (8); |
| $Pr$, | Prandtl number, $v/\alpha$; |
| $q$, | natural convection heat-transfer rate; |
| $Q_L$, | rate of heat loss from insulated surface of heated plate to atmosphere; |
| $Ra_D$, | Rayleigh number, $g\beta(\Delta T)D^3/(v\alpha)$; |
| $T$, | temperature; |
| $T_C$, | temperature of cold plate; |
| $T_H$, | temperature of hot plate; |
| $\Delta T$, | temperature difference, $T_H - T_C$; |
| $T_m$, | mean temperature, $(T_H + T_C)/2$; |
| $\alpha$, | thermal diffusivity; |
| $\beta$, | thermal expansion coefficient; |
| $v$, | kinematic viscosity; |
| $x, y$, | coordinates, defined in Fig. 1. |

## 1. INTRODUCTION

THE AIR layer in a rectangular cavity having isothermal side walls of different temperatures (Fig. 1) is used in many engineering applications as a means of impeding the heat flow due to the low thermal conductivity, low density, and transparency of the air. A number of investigations concerning convection in this kind of confined space for air as the working fluid have been carried out since the turn of this century. In 1946, Jakob [1] analyzed the experimental results of Mull and Reiher [2] and proposed the following correlations:

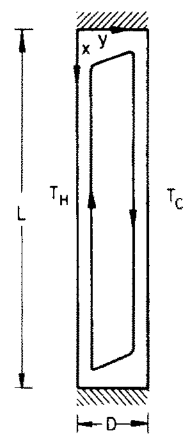


FIG. 1. Rectangular cavity.

$$Nu_D = 0.18\,(Gr_D)^{1/4}(L/D)^{-1/9} \qquad 2\times 10^4 < Gr_D < 2\times 10^5 \quad (1)$$

$$Nu_D = 0.065\,(Gr_D)^{1/3}(L/D)^{-1/9} \qquad 2\times 10^5 < Gr_D < 1\times 10^7. \quad (2)$$

Equations (1) and (2) are supposed to be valid for aspect ratio, $L/D$, in the range from 3.12 to 42.2. Equation (2) is intended to describe turbulent flow conditions. Wilkes and Peterson [3] led to about 50% and Griffiths and Davis [4] arrived at almost 40% higher heat-transfer rates than Mull and Reiher.

In 1954, Batchelor [5] investigated this problem in three cases: (1) small Rayleigh numbers ($Ra_D < 10^3$) with $L/D$ approximately unity, (2) general Rayleigh number with large $L/D$'s, and (3) large Rayleigh numbers with general $L/D$'s. An analytical solution for the first case was found by utilizing a method of expanding the stream function and temperature in power series in terms of the Rayleigh number. He also obtained the solution for the last two cases by making drastic idealizations. For the second case, he recommended that an equation in the form as:

$$Nu_D = 1 + [(2\gamma - 1)/720]Ra_D(D/L) \qquad (3)$$

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could be used, where $\gamma$ is a constant. He predicted transition to turbulent flow at $Ra = 13\,700$ for air in the second case, and at $Ra_D = 10^9(L/D)^{-3}$ for air (perhaps for other fluids) in the last case.

Eckert and Carlson [6] investigated the temperature field with the help of a Zehnder–Mach interferometer. They proposed that the flows may be classified into conduction, transition and boundary layer regimes; the horizontal temperature gradient at the cavity centre provides a useful criterion. Local heat-transfer coefficients were derived from the temperature gradients in the air normal to the plate, the equation given by:

$$Nu_D = 0.119\,(Gr_D)^{0.3}(D/L)^{0.1} \tag{4}$$

is recommended for the boundary-layer regime. The heat-transfer coefficients calculated with equation (4) are 20% higher than previous measurements [2].

In 1966, Wilkes and Churchill [7] published one of the first successful attempts at solving the full two-dimensional, time dependent differential equations describing the flow. They used the implicit alternating-direction finite-difference method and were able to obtain steady-state solutions for small values of $L/D$ of 1, 2 and 3. Values of Nu reported were up to 70% in excess of those given by Jakob's empirical equation. Since that time, a series of papers have been pulbished on the numerical solution of steady natural convection in rectangular enclosures by Elder [8], MacGregor [9], Newell and Schmidt [10], and Thomas and Davis [11]. All those investigations were restricted to the small values of $L/D$. The following expressions were given by Newell and Schmidt [10] for the Nusselt number:

$$L/D = 1 \quad Nu_D = 0.0547\,(Gr_D)^{0.397} \tag{5}$$

$2.5 \leqslant L/D \leqslant 20$

$$Nu_D = 0.155\,(Gr_D)^{0.315}(L/D)^{-0.265}. \tag{6}$$

The purpose of the present investigation is to extend the range of the values of aspect ratio, $L/D$, with large absolute dimensions of the test apparatus, and to examine the role which $L/D$ plays in temperature field and heat-transfer rate within rectangular enclosures. In our approach, the apparatus is operated at the values of aspect ratio, $L/D$, in the range from 4.9 to 78.7 and for various differences in temperature between the heated and cooled plate. In order to give an understanding of natural convection flow within the rectangular cavity, measured temperature profiles are analyzed with flow regimes proposed by [6]. One overall Nusselt–Grashof correlation obtained for the heat-transfer measurements involves $L/D$ implicitly, and fits the data with an average deviation of less than 7.8% with 91.3% of the data within $\pm 20\%$ of values predicted by this correlating equation. In addition, Nusselt–Grashof correlations, involving the aspect ratio, $L/D$, explicitly, are also developed for the range of the test values of Grashof number. The correlation fits the data with an average deviation of less than 7.6%.

## 2. HEAT-TRANSFER APPARATUS AND PROCEDURE

The heat-transfer apparatus was constructed specially for the collecting of heat flux and temperature distribution data with various aspect ratios in an air layer enclosed within a rectangular cavity. It consists of two vertical metal plates, each 1.005-m long and 0.505-m wide, enclosed in a well insulated rectangular box. A photograph of the assembled apparatus with its two sections opened to show the vertical plates is shown in Fig. 2, and Fig. 3 presents schematics of the entire operating system.

One of the two plates was made of a 12.7-mm thick rolled aluminium sheet heated by 20 electric strip heaters. Each heater was constructed by utilizing (365 $\pm 5\,\Omega$) Nichron ribbon wound on a Mica sheet which was 0.52-m long and 40-mm wide. A total of 20 electric strip heaters were divided into three sets. Each set could be individually controlled so that a desired isothermal wall condition could be obtained. Fifteen thermocouples were embedded in the aluminium plate and revealed a maximum temperature variation of 1°C over the surface of the plate at a temperature difference of 80°C between the two vertical plates of the cavity.

A 3.2-mm thick copper sheet was used for the cold plate. It was cooled by water running through a gasket, with nine bafflefins inside to guide the water flow, attached to the back surface of the plate. As before, fifteen thermocouples were used to measure the surface temperature of the plate. With the flow rate of the cooling system at 0.61 $m^3/h$, a maximum temperature variation over the plate was measured of 3°C with a temperature difference between the two vertical walls of the cavity at 80°C.

The whole system of the two metal plates was fixed in a wood frame insulated inside with 90-mm thick glass wool. Spacers between the upper and lower walls enclosing the air space were made of polystyrene foam. These spacers were available with different dimensions $D$. The spacers separating two vertical end plates were also manufactured from polystyrene foam. The height of the air layer under investigation could be changed by additional horizontal polystyrene spacers which were clamped between the two metal plates.

The thermocouple probes used to obtain the temperature profiles were constructed of GG-K30 Chromel and Alumel wires. The junction end of the wires was tied with a tiny nylon line. The wire and line were withdrawn from small slots opened in opposite vertical end walls. They were straightened by hanging a weight on the end of the wire on one side and the line on the other side. The thermocouple probe could advance horizontally across the gap between the air layer in the cavity by moving both ends simultaneously in the same direction. The position of the probe was indicated by the reading of fine scales within 0.159 cm attached at both ends along the slots. On the basis of previous work, [1] and [6], it was concluded that the width used in the present apparatus is sufficient to make end effects of the two vertical end surfaces negligible and closely approximates a two-dimensional

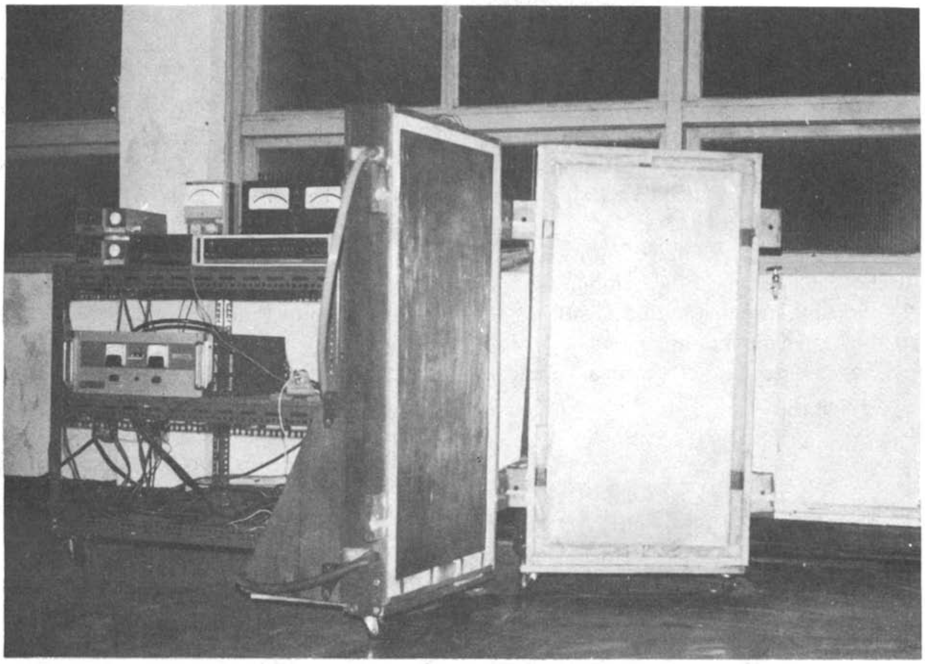

FIG. 2. Experimental apparatus.

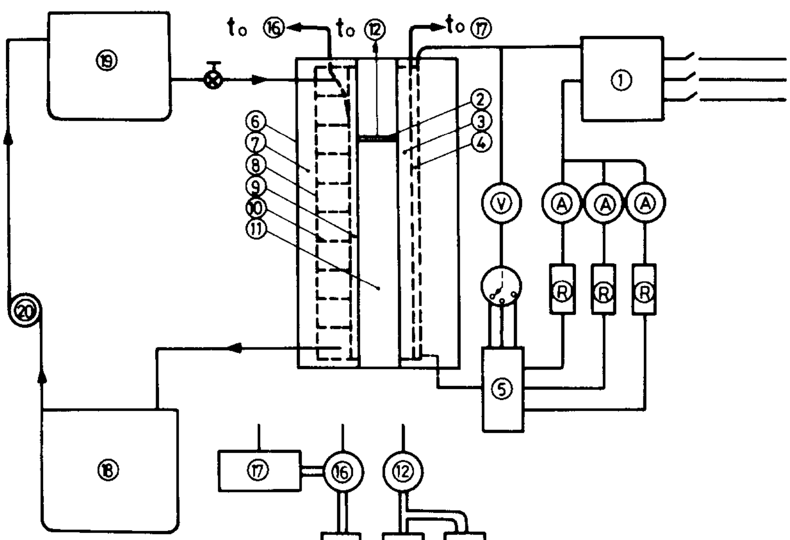

FIG. 3. Schematic diagram of experimental apparatus. (1) D.C. power supply; (2) Thermocouple probe; (3) Aluminum plate; (4) Electric heaters; (5) Electric distributer; (6) Wood box; (7) Glass wool; (8) Water jacket; (9) Copper plate; (10) Baffle-plate; (11) Polystyrene spacer; (12) Rotary switch; (13) Ice point cell; (14, 15) Thermocouple indicators; (16) Rotary switch; (17) Multiple thermocouple scanner; (18, 19) Water reservoirs; (20) Water pump; A—D.C. Ammeter; V—D.C. Voltmeter; R—Adjustable resistor.

situation. In this way, five thermocouple probes were placed in a common vertical plane through the middle of the width of the plates and were spaced at five different heights.

A total of approximately 170 heat-transfer runs were conducted with two different cavity heights, 1 m and 0.4975 m. The layer thickness ranged from 12.7 mm to 101.6 mm. These provided values for $L/D$ ranging from 4.9 to 78.7. At each aspect ratio, nine different values of temperature difference, $\Delta T$, were established by controlling the electrical power input to the heaters. The Grashof number based on layer thickness obtained in the test ranged from $1.5 \times 10^3$ to $7.0 \times 10^6$. At selected temperatures for each aspect ratio, temperature traverses were made utilizing the thermocouple probes. Sufficient time was allowed for establishment of a steady state before each run.

The natural convection heat-transfer rate, $q$, can be found by subtracting all heat dissipated in any form other than by natural convection between the test surfaces from the measured value of the total power input to the heaters. Heat losses were found as follows: (a) the heat loss from the insulated surface of heated aluminum plate to atmosphere through the insulation; (b) the heat transfer between the two vertical plates by radiation; and (c) the heat transfer by conduction through the four spacers enclosing the air space between the two vertical plates.

The item (a), $Q_L$, could be estimated for each $T_H$ by taking half the value of the power input used to maintain $T_H$ with insulating condition identically on both sides of the heated aluminium plate. The curve of Fig. 4 could be used to predict $Q_L$ at a variety of heated-plate temperature.

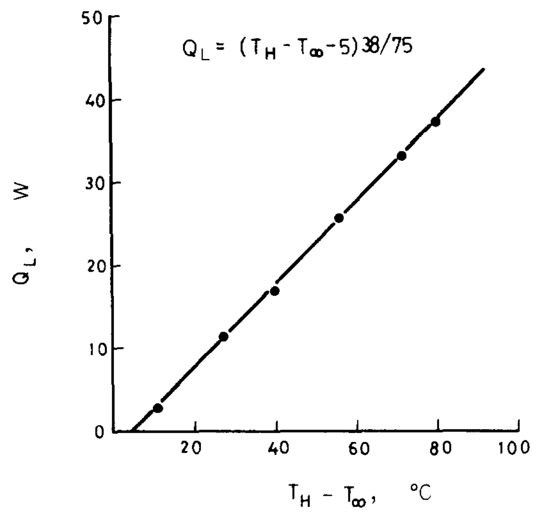


FIG. 4. Calibration curve giving heat loss, $Q_2$.

The heat transferred by radiation can be simply calculated using the laws of radiation. The emissivities of 0.07 and 0.09 were used for copper and aluminium plates respectively in our calculation which also take into account their surface conditions. The approximate magnitude of radiative heat transfer was found ranging from 7 to 17% of the total power input.

The heat loss due to conduction through the four spacers, item (c), was found to be negligible for the gap thickness less than 76.2 mm and only 5%, based on the total power input, was noticed for the largest thickness $D$ of 101.6 mm.

### 3. TEMPERATURE DISTRIBUTION

Profiles of temperature, $T$, vs distance, $y$, from the hot plate were obtained at selected values of $\Delta T$ and vertical positions, $x$, for each value of $L/D$. It should be noted that temperature fluctuations of several degrees occurred for all values of $L/D$ at high values of $\Delta T$. Such fluctuations may be interpreted as the result of unsteady flow, and meaningful profiles were not obtained under these conditions. Only those profiles for which there were no fluctuations are presented herein.

The form of the dimensionless temperature profile for a given vertical position, measured from the upper wall, appears to be essentially independent of the temperature difference, $\Delta T$. Thus dimensionless profiles are presented for only one $\Delta T \approx 20°C$ at each $L/D$ (Figs. 5–10). Since all temperature profiles presented here are dimensionless, the adjective "dimensionless" will not henceforth be used.

Figures 5 and 6 display linear temperature distributions across the major-portion of the air layer except both ends, indication that the heat from the hot plate to the cold one is transported by conduction through the major portion of the layer, and by convection at the lower end of the hot plate and upper end of the cold plate. On the basis of previous work, [6] and [11], the flow field corresponding with this type of temperature distribution is defined as in the "regime of conduction" as long as the horizontal temperature gradient $dT/dy$ at the cavity centre remains at a value of about $-1$.

Figures 7–9 display a precipitous drop of temperature in the region immediately adjacent to both hot and cold plates with a horizontal temperature line in the central core. It reveals that the heat is mainly transported from the hot plate to the cold plate by tangential convection. Furthermore, temperature in-

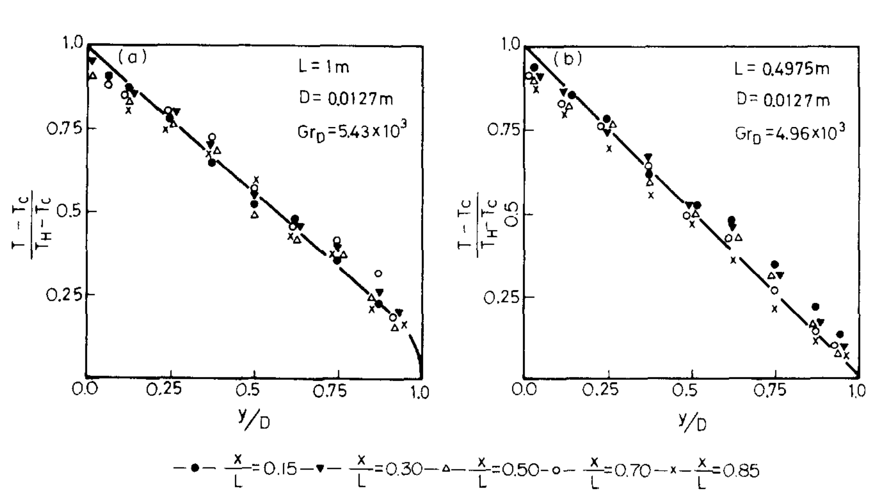


FIG. 5. Temperature profiles for $D = 0.0127$ m; (a) $L/D = 78.7$; (b) $L/D = 39.2$.

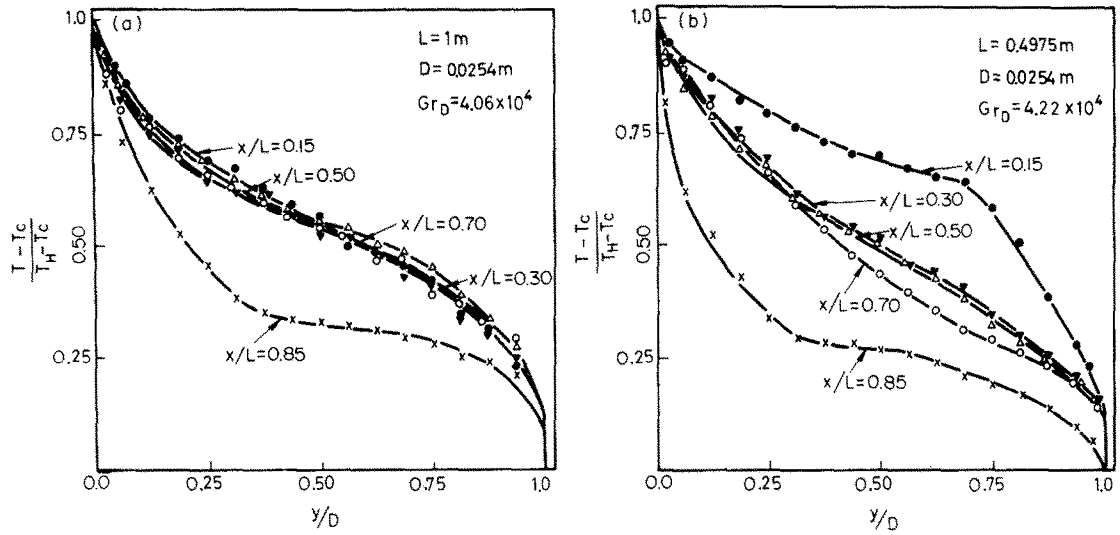


FIG. 6. Temperature profiles for $D = 0.0254$ m (a) $L/D = 39.4$, (b) $L/D = 19.6$.

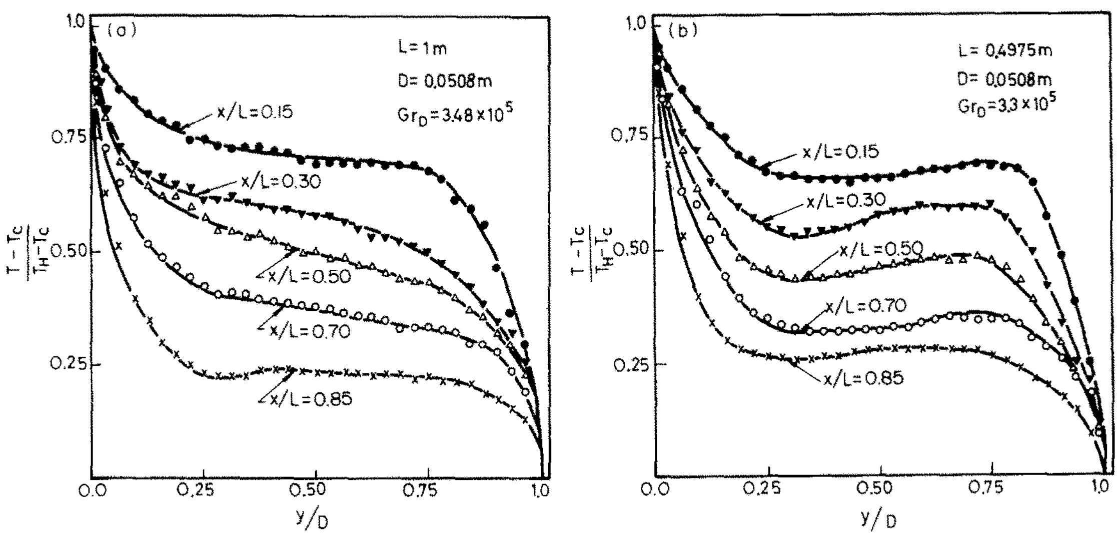


FIG. 7. Temperature profiles for $D = 0.0508$ m (a) $L/D = 19.7$, (b) $L/D = 9.8$.

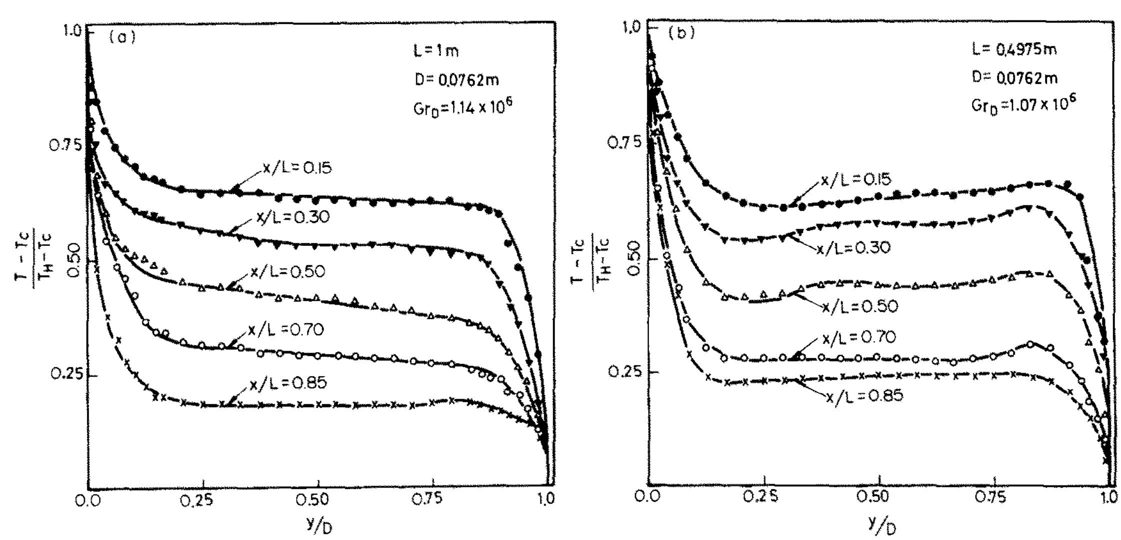


FIG. 8. Temperature profiles for $D = 0.0762$ m (a) $L/D = 13.1$; (b) $L/D = 6.5$.

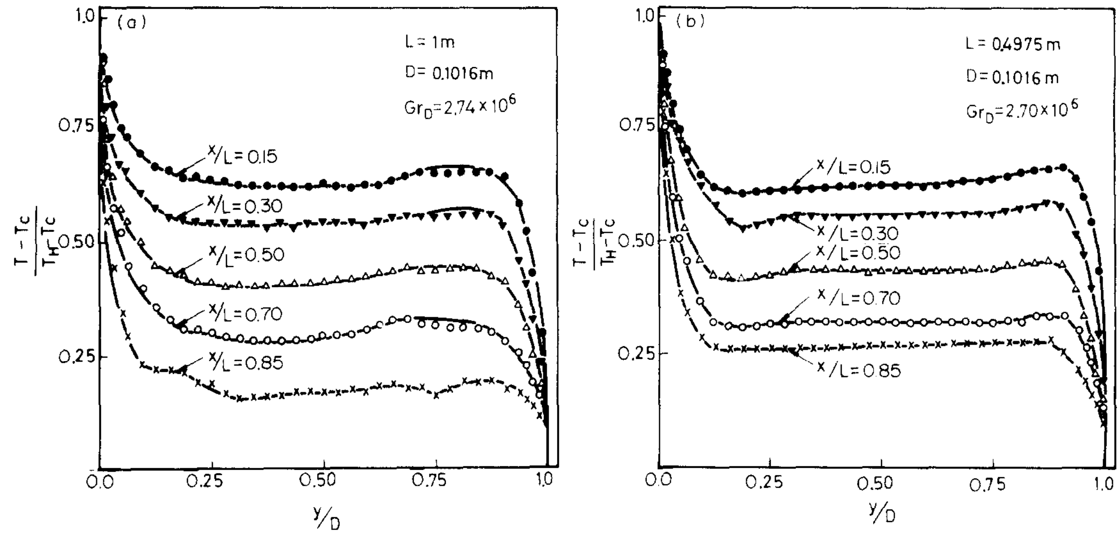


FIG. 9. Temperature profiles for $D = 0.1016\,\text{m}$ (a) $L/D = 9.8$; (b) $L/D = 4.9$.

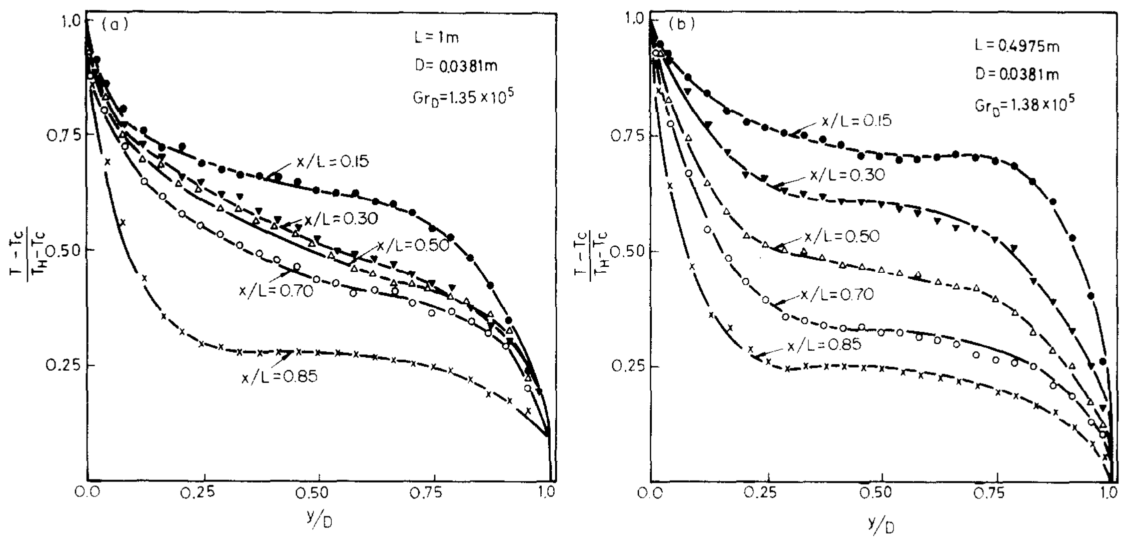


FIG. 10. Temperature profiles for $D = 0.0381$ m (a) $L/D = 26.2$; (b) $L/D = 13.1$.

versions were observed in the central regions in Figs. 7(b), 8(b) and 9(b). These inversions are a manifestation that the flow immediately adjacent to the vertical plates is so strong that it produces a high rate of tangential convection of heat along the surface of the hot plate relative to vertical transport of heat from the surface of the hot plate across to the cold plate. Conduction in the central region of the cavity is actually opposite to the overall direction of heat flow. Such inversions have not been observed by [6]; but it was predicted via numerical simulations by Davis [12]. On the other hand, these temperature inversions which we found may also tend to confirm the flow studies of Elder [13] where he found numerous secondary vortices to occur in the core flow at higher Rayleigh numbers. We feel that such vortices could also account for temperature inversions such as we have observed. Although Elder's work was for large Prandtl numbers, our results indicate that such vortices also occur for smaller Prandtl numbers. When a flow field under such temperature distribution with a core of uniform temperature in direction along horizontal lines exists, it is referred to as "boundary-layer regime". In other words, this flow regime is characterized by its temperature field with the value of $dT/dy$ at cavity centre being larger or equal to zero. It is interesting to point out that all the inversions were observed in the cavities with small aspect ratio, $L/D$. This may illustrate, for one thing, that the effect of tangential convection will be reduced in the cavities with high aspect ratio.

The temperature profiles reveal a precipitous drop along the hot and cold plates with a linear temperature drop in the central core. This can be found in Figs. 10(a) and (b). It indicates that in addition to the heat transported by tangential convection along the surface of the both plates, there is heat conduction through the central core of the layer. The flow field regime will be called "transition regime" in which the temperature gradient $dT/dy$ at the cavity centre is ranged between

−1 and 0. Examining Fig. 10, an interference between temperature distribution lines was found in the upper-central region for the one with larger aspect ratio $L/D$. This appearance at this kind of temperature distribution is thought to be due to some rotation of the core fluid occurring in the air layer. It would seem that boundary layer "interference" occurs throughout the conduction regime, and in the boundary-layer regime no boundary-layer interference occurs and the core fluid is relatively stagnant. Thus, in the transition regime the boundary layers will not interfere, but they will almost meet near the center of the cavity. This will cause some rotation of the core fluid, which prevents the fluid in the core from being isothermal in the horizontal direction. On the other hand, the velocity here is much less than that in the boundary layer, which prevents the linear temperature profile.

It should be noticed that strong temperature fluctuations were observed in the present investigation when temperature differences reached 30°C for the values of each aspect ratio of $L/D > 30$. However, for small $L/D$ this kind of temperature fluctuation was not detected until $\Delta T$ reaches 60°C. If the fluctuations are interpreted as the result of unsteady flow, it means that the unsteady flow will take place at the lower values of Grashof number for layers with larger values of aspect ratio $L/D$. This was the case found in spherical annuli by Yin *et al.* [14].

The classification of flow regimes can be found in Fig. 11. The experimental runs of the present investigation in which a linear temperature drop exists in the major-portion of the flow field indicated by black circles. The conditions for which a horizontal temperature line exists at least over part of the field are indicated by crosses. The other experimental runs are indicated by open squares. The present investigation obtains a lower slope of the limits of regimes than Eckert and Carlson [6] did. It means that the aspect ratio gives stronger influence on the limits of flow regimes in our studies. The explanation for this difference is that Eckert and Carlson did not have the runs, during their test, for $Gr_D > 10^4$ with $L/D > 30$ because the dimension of their test apparatus was not big enough.

## 4. HEAT-TRANSFER RESULTS

The heat-transfer results obtained experimentally are analyzed in terms of the following independent parameters: Grashof number, $Gr_D$, Prandtl number, $Pr$, and aspect ratio, $L/D$. When considering natural convection heat transfer within cavities, the literature, in general, states that the heat transfer taking place can be determined in functional notation as

$$Nu_D = f(Gr_D, Pr, L/D). \tag{7}$$

Where $Nu_D$ is the Nusselt number and defined as

$$Nu_D = \frac{h_c D}{k}. \tag{8}$$

Since air was the only fluid used, and since its Prandtl number does not vary significantly within the range of temperatures involved in this investigation (15–100°C), $Pr$ is considered to have been constant. Thus, the functional relationship

$$Nu_D = f(Gr_D, L/D) \tag{9}$$

was sought. The foregoing dimensionless parameters were calculated using properties evaluated at a mean temperature $T_m$ which is defined by

$$T_m = (T_H + T_C)/2. \tag{10}$$

In order to make the data available in a wide range of aspect ratios and Grashof numbers, two different absolute test heights were used for the present investigation. All the data obtained in the current test were presented on Fig. 12 using the plots of $Nu_D$ vs $Gr_D$ in log–log coordinates. It was observed that the Nusselt number has only a weak dependence on $L/D$. Employing the least-squares technique, the overall correlation equation was given by

$$Nu_D = 0.091\ Gr_D^{0.307} \tag{11}$$

with an average deviation of 7.8%, and 91.3% of the data within ±20% of values predicted by the equation. As far as the effect of $L/D$ is concerned, the resulting correlating equation was obtained of the form

$$Nu_D = 0.210\ Gr_D^{0.269}(L/D)^{-0.131} \tag{12}$$

which correlated all the data with an average deviation

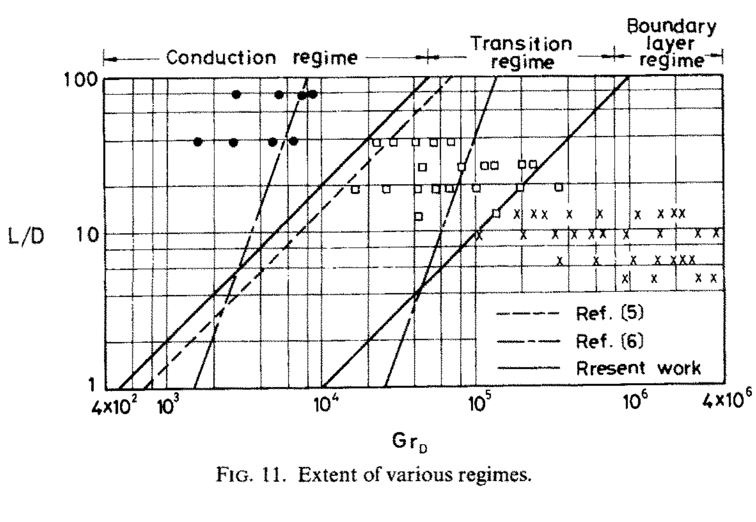


FIG. 11. Extent of various regimes.

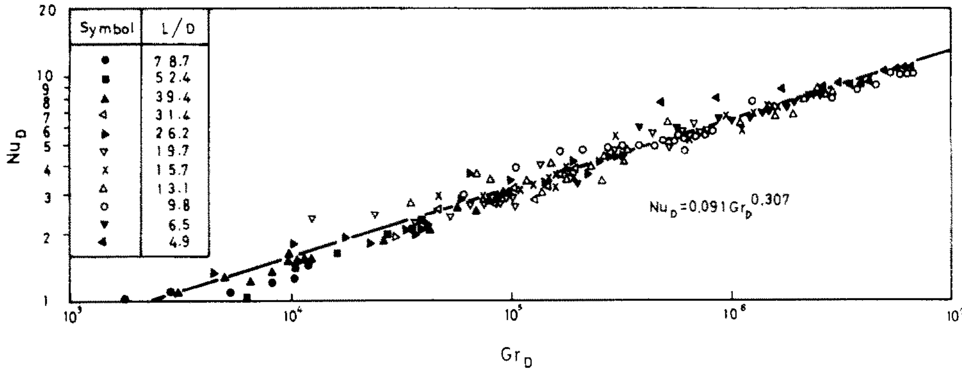


FIG. 12. Overall correlation of heat-transfer data.

Table 1. Empirical constants and range of $L/D$ and $Gr_D$ for equation (13)

| Investigator | $C$ | $a$ | $b$ | Range of $L/D$ | Range of $Gr_D$ |
|---|---|---|---|---|---|
| Present work | 0.210 | 0.269 | −0.131 | 4.9 to 78.7 | $1.5\times10^3$ to $7.0\times10^6$ |
| Jakob | 0.180 | 0.25 | −0.111 | 3.12 to 42.2 | $2\times10^4$ to $2\times10^5$ |
| Jakob | 0.065 | 0.333 | −0.111 | 3.12 to 42.2 | $2\times10^5$ to $1\times10^7$ |
| Eckert and Carlson | 0.119 | 0.3 | −0.1 | 10.0 | $8.0\times10^4$ to $2.0\times10^5$ |
| Newell and Schmidt | 0.155 | 0.315 | −0.265 | 2.5 to 20 | $4\times10^3$ to $1.4\times10^5$ |

of less than 7.6% and also with 94.4% of the data within ±20% of values given by it. (The deviation at a particular data point is defined as the absolute difference between the data and the equation value divided by the data value; the average per cent deviation is the sum of these individual deviations divided by the total number of data points.) Equations (11) and (12) are restricted to $4.9 \leqslant L/D \leqslant 78.7$ and $1.5\times10^3 \leqslant Gr_D \leqslant 7.0\times10^6$.

For comparison purpose a listing of the expression for Nusselt number

$$Nu_D = C\,(Gr_D)^a(L/D)^b \qquad (13)$$

which have been previously reported is presented in Table 1. The exponent on $Gr_D$ for the present study is lower than those given by all previous investigators except the one given by Jakob supposed for laminar flow conditions. The exponent on $L/D$ obtained here is −0.131 which is only a little higher value than the values obtained by Jakob or Eckert and Carlson (−0.111 and −0.1, respectively) but considerably lower than the value given by Newell and Schmidt (−0.265). The values predicted by equation (12) will be higher than those given by Jakob within the range of 9–41%.

## 5. CONCLUSION

This paper reports the results of an experimental study of natural convection in an air layer enclosed within rectangular cavities. Both temperature distributions and heat-transfer rates are reported for a wide range of aspect ratios and Grashof numbers. The measured temperature profiles were found to be relatively independent of the temperature difference between the two vertical plates for each aspect ratio. Temperature inversions, believed to be caused by a high rate of tangential convection of heat relative to the horizontal transport, were observed in several of the test conditions.

Two Nusselt–Grashof relations, equations (11) and (12), were obtained which reasonably correlate the measured heat-transfer data and are in tolerable agreement with previous experimental results.

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## CONVECTION NATURELLE DANS UNE COUCHE D'AIR ENFERMEE DANS DES CAVITES RECTANGULAIRES

**Résumé**– On étudie expérimentalement la convection naturelle dans une couche d'air contenue dans une cavité rectangulaire ayant des parois latérales à température différente et des facteurs de forme variant de 4,9 à 78,7. Des profils de température sont analysés en rapport avec les régimes d'écoulement. Deux corrélations Nusselt–Grashof sont présentées pour les résultats, avec un écart moyen inférieur à 7,8 pour cent. Le nombre de Grashof basé sur l'épaisseur de la couche varie de $1{,}5 \times 10^3$ à $7{,}0 \times 10^6$.

## FREIE KONVEKTION IN EINER LUFTSCHICHT, DIE IN EINEM RECHTWINKLIGEN BEHÄLTER EINGESCHLOSSEN IST

**Zusammenfassung**—In einer Luftschicht, die sich in einem rechtwinkligen Behälter mit Seitenwänden verschiedener Temperaturen befindet, wurde freie Konvektion experimentell untersucht und zwar für verschiedene Höhe-Breite-Verhältnisse im Bereich von 4, 9 bis 78,7. Die gemessenen Temperaturprofile werden zusammen mit den Strömungen ausgewertet. Für die gemessenen Wärmeübertragungsdaten werden zwei Nusselt–Grashof-Beziehungen dargestellt, deren durchschnittliche Abweichung von den Daten weniger als 7,8% beträgt. Die Grashof-Zahl, die mit der Schichtdicke gebildet wird, lag im Bereich von $1{,}5 \cdot 10^3$ bis $7{,}0 \cdot 10^6$.

## ИССЛЕДОВАНИЕ ЕСТЕСТВЕННОЙ КОНВЕКЦИИ ВОЗДУХА В ПРЯМОУГОЛЬНОЙ ПОЛОСТИ

**Аннотация** — Экспериментально исследовалась естественная конвекция в воздушном слое в прямоугольной полости, боковые стенки которой поддерживались при различной температуре, с отношением сторон от 4,9 до 78,7. Измеренные профили температуры соотносились с режимами течения. Представлены две зависимости Нуссельта-Грасгофа для измеренного теплообмена, которые дают расхождение с опытными данными меньше, чем на 7,8%. Число Грасгофа, отнесенное к толщине слоя, изменялось от $1{,}5 \,.\, 10^3$ до $7{,}0 \,.\, 10^6$.