Int. J. Heat Mass Transfer. Vol. 17. pp. 1209–1217. Pergamon Press 1974. Printed in Great Britain

NATURAL CIRCULATION IN AN INCLINED RECTANGULAR CHANNEL HEATED ON ONE SIDE AND COOLED ON THE OPPOSING SIDE

HIROYUKI OZOE,* KAZUMITSU YAMAMOTO,* HAYATOSHI SAYAMA* and STUART W. CHURCHILL†

(Received 21 November 1973)

Abstract—Two-dimensional natural circulation in an inclined, confined box heated on one side and cooled on the opposing side was modelled and solved by finite-difference methods. The angle of inclination was varied from 0 to 180° for four aspect ratios. Experimental results were obtained which are in good agreement with the theoretical predictions. The preferred mode of circulation was observed to change with the angle of inclination and the aspect ratio. A minimum in the heat flux occurred at the point of transition. A maximum in the heat flux occurred as the angle was further increased.

NOMENCLATURE

c_p, specific heat;
g, acceleration due to gravity;
h, local heat-transfer coefficient, $= q/(\theta_h - \theta_c)$;
H, height of channel in y-direction;
k, thermal conductivity;
L, length of channel in z-direction;
Nu, Nusselt number $qH/k(\theta_h - \theta_c)$;
$\overline{Nu}$, average Nusselt number $\frac{1}{H} \cdot \int_0^H Nu(Y)\,dX$;
p, pressure;
p^*, pressure perturbation above the static state;
p_0, static pressure;
Pr, Prandtl number, $= c_p\mu/k$;
q, heat flux density;
Ra, Rayleigh number, $= \rho_0^2 g c_p \beta(\theta_h - \theta_c)H^3/k\mu$;
t, time;
T, dimensionless temperature, $= (\theta - \theta_0)/(\theta_h - \theta_c)$;
u, velocity component in the x-direction, $= \partial\Phi/\partial y$;
U, dimensionless velocity in the X-direction, $= uH/\kappa$;
v, velocity component in the y-direction, $= -\partial\Phi/\partial x$;
V, dimensionless velocity in the Y-direction, $= vH/\kappa$;
W, width of the channel in the x-direction;
x, distance from left side of channel;
X, dimensionless coordinate, $= x/H$;
ΔX, grid-size in X-direction;
y, distance from top of the channel;
Y, dimensionless coordinate, $= y/H$;
ΔY, grid-size in Y-direction.

Greek symbols

β, volumetric coefficient of expansion with temperature;
ζ, vorticity, $= \partial v/\partial x - \partial u/\partial y$;
$\hat{\zeta}$, dimensionless vorticity, $= \zeta H^2/\kappa$;
θ, temperature;
κ, thermal diffusivity, $= k/\rho c_p$;
μ, viscosity;
ρ, density;
τ, dimensionless time, $= t\kappa/H^2$;
Φ, stream function;
Ψ, degree of inclination of the hot plate from the horizontal plane.

Mathematical symbols

$\frac{D}{Dt}$, substantial derivative, $= \frac{\partial}{\partial t} + u\frac{\partial}{\partial x} + v\frac{\partial}{\partial y}$;
$\frac{\mathscr{D}}{\mathscr{D}\tau}$, dimensionless substantial derivative, $= \frac{\partial}{\partial \tau} + U\frac{\partial}{\partial X} + V\frac{\partial}{\partial Y}$;
∇^2, two-dimensional Laplacian in dimensionless coordinates, $= \frac{\partial^2}{\partial X^2} + \frac{\partial^2}{\partial Y^2}$;

*Department of Industrial and Mechanical Engineering, School of Engineering, Okayama University, Okayama, Japan.
†School of Chemical and Biochemical Engineering, University of Pennsylvania, Philadelphia, Pennsylvania, U.S.A.

Subscripts

- 0, value at mean temperature;
- h, value at hot plate;
- c, value at cold plate.

INTRODUCTION

HYDRODYNAMIC stability and natural convection in a horizontal fluid layer heated from below has been the subject of theoretical research since the time of Rayleigh [1]. More recent studies including references [2–5] have considered the critical state for a horizontal fluid layer bounded by the rigid boundaries. Free convection between a heated vertical plate and the ambient fluid has also received continued attention. In recent times attention has turned to natural convection within enclosures as exemplified by references [6] and [7]. Investigations have also been extended to non-Newtonian fluids as illustrated by references [8–10] because of their industrial importance. Studies of inclined enclosures are more limited in number and scope. Natural convection in an inclined porous media was investigated by Holst and Aziz [11, 12]. Hart [13] investigated convective stability of air and water in differentially heated, inclined boxes with height-to-width ratios of 25 and 37, and observed longitudinal roll-cells with their axes in the upslope direction at angles of inclination of the hot plate from 0 to 80° even at $Ra < 10\,000$ and also such rolls at 165°. He further reported (14) on the structure of thermal convection in a slightly slanted slot. Konicek and Hollands (15) made similar measurements and measured the critical value of Ra and Nu at slightly higher values of Ra in a differentially heated inclined layer of air for a height-to-width ratio of 44 and inclinations from 0 to 90° and developed a correlation for Nu for Ra slightly above Ra_c.

The main difference of the current investigation from the above is in the aspect ratio. The work of Hollands and Konicek and of Hart is for a shallow box with large aspect ratios. Our work is for small aspect ratios and hence takes into primary count the effect of the side walls on the two-dimensional circulation.

Recently, Ozoe, Sayama and Churchill [16] investigated natural convection in a long inclined channel with a square cross section and found a minimum and a maximum in the heat flux during rotation of the hot plate from the horizontal to the vertical plane about the long axis. They reported good agreement between the experimental average Nusselt number and the one predicted by numerical integration. The minimum heat flux occurred as the angle of inclination was decreased to about 10°, coincident with a change in the mode of circulation from a long roll-cell to a series of roll-cells with axes normal to the long dimension. The maximum heat flux occurred at an inclination of about 50° for all of the Rayleigh numbers which were studied.

In the work reported herein the circulation of the fluid in a rectangular channel with various aspect ratios was modelled and solved by finite difference methods for angles of inclination from 0 to 180°. The theoretical predictions were tested experimentally. The geometry is shown schematically in Fig. 1. The aspect

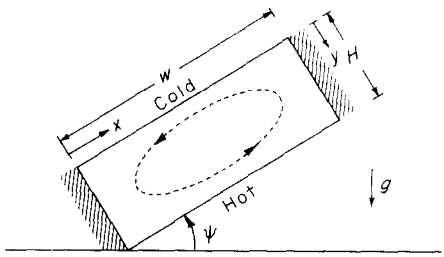

FIG. 1. Geometrical configuration of model and experiments.

ratio is presumed to be characterized by W/H only, i.e. L/H is presumed to be long enough so that the end walls in the z-direction do not affect the behavior significantly. The upper surface of the channel was maintained at a uniform temperature and the opposing surface at a uniform higher temperature. The other sides were thermally insulated. The whole channel was then rotated about the z-axis (long dimension).

MATHEMATICAL MODEL

The usual Boussinesq approximations were made with

$$\rho = \frac{\rho_0}{1+\beta(\theta-\theta_0)} \tag{1}$$

and

$$\theta_0 = (\theta_h+\theta_c)/2. \tag{2}$$

The physical properties of the fluid other than the density were assumed to be constant and were calculated at θ_0. The velocity, pressure and temperature fields were assumed to be independent of z. The following equations of conservation are then applicable [16].

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y} = 0 \tag{3}$$

$$\rho_0\frac{\mathrm{D}u}{\mathrm{D}t} = -\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right)-\rho g\sin\Psi \tag{4}$$

$$\rho_0\frac{\mathrm{D}v}{\mathrm{D}t} = -\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+\rho g\cos\Psi \tag{5}$$

$$\rho_0 c_p\frac{\mathrm{D}\theta}{\mathrm{D}t} = k\left(\frac{\partial^2\theta}{\partial x^2}+\frac{\partial^2\theta}{\partial y^2}\right). \tag{6}$$

The boundary conditions corresponding to Fig. 1 are

$$\theta(x,0,t) = \theta_c$$
$$\theta(x,H,t) = \theta_h$$
$$\frac{\partial\theta}{\partial x}(0,y,t) = \frac{\partial\theta}{\partial x}(W,y,t) = 0$$
$$u(0,y,t) = u(W,y,t) = u(x,0,t) = u(x,H,t) = 0$$
$$v(0,y,t) = v(W,y,t) = v(x,0,t) = v(x,H,t) = 0.$$

The total pressure may be represented as the sum of a static and a perturbed component:

$$p = p_0 + p^*. \tag{7}$$

Substituting for p in equations (4) and (5), taking derivatives with respect to y and x, respectively, and subtracting, then introducing the vorticity and stream function defined as

$$\zeta = -\nabla^2\Phi = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \tag{8}$$

yields

$$\frac{D\zeta}{Dt} = -g\beta\left(\frac{\partial\theta}{\partial y}\sin\Psi + \frac{\partial\theta}{\partial x}\cos\Psi\right) + \frac{\mu}{\rho_0}\left(\frac{\partial^2\zeta}{\partial x^2} + \frac{\partial^2\zeta}{\partial y^2}\right). \tag{9}$$

Dedimensionalizing by the procedure of Hellums and Churchill [17] yields

$$\frac{1}{Pr}\frac{D\zeta}{D\tau} = -Ra\left(\frac{\partial T}{\partial Y}\sin\Psi + \frac{\partial T}{\partial X}\cos\Psi\right) + \nabla^2\zeta \tag{10}$$

$$\frac{DT}{D\tau} = \nabla^2 T. \tag{11}$$

The boundary conditions become

$$T(X,0,\tau) = -1/2$$
$$T(X,1,\tau) = 1/2$$
$$\frac{\partial T}{\partial X}(0,Y,\tau) = \frac{\partial T}{\partial X}\left(\frac{W}{H},Y,\tau\right) = 0$$
$$U(0,Y,\tau) = U\left(\frac{W}{H},Y,\tau\right) = U(X,0,\tau) = U(X,1,\tau) = 0$$
$$V(0,Y,\tau) = V\left(\frac{W}{H},Y,\tau\right) = V(X,0,\tau) = V(X,1,\tau) = 0.$$

SOLUTION OF THE MODEL

The above mathematical model was solved by finite-difference methods using essentially the same unsteady-state technique as Ozoe and Churchill [9] and Ozoe, Sayama and Churchill [16]. The non-dimensional X and Y coordinates were divided into segments

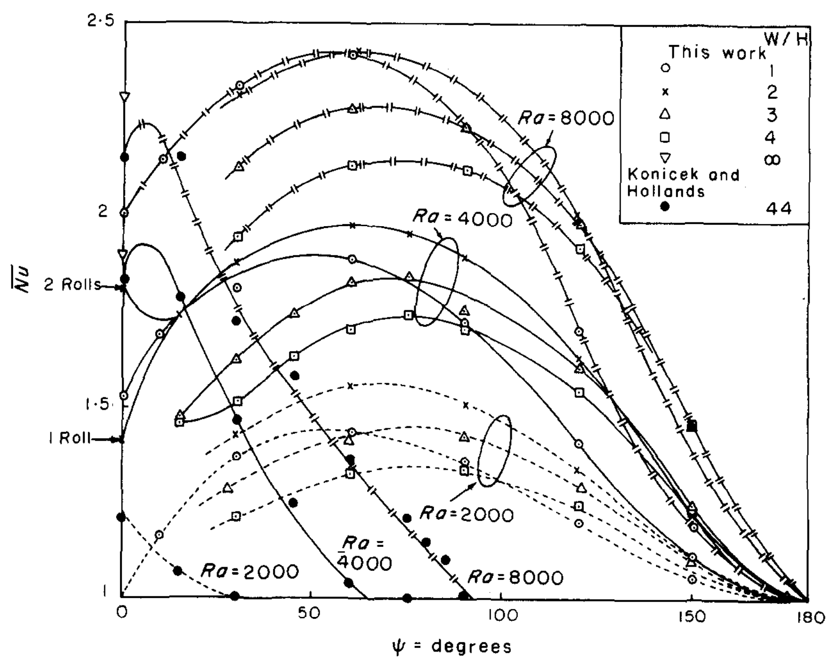

FIG. 2. Computed effect of angle of inclination on rate of heat transfer at $Pr = 10$. This work, by interpolation of values at $\Delta X = \Delta Y = 0{\cdot}1$; Hollands and Konicek, by equation (12).

of equal size to minimize computational errors. $\Delta X = \Delta Y = 0{\cdot}1$ was adopted for the basic computations. The maximum non-dimensional time step, $\Delta\tau$, required for stability was chosen by trial and error.

The critical problem in finite-difference calculations is usually the extrapolation of the results to zero grid-size. The maximum allowable time-step and the computational time increase very rapidly as the grid-size is decreased. Ozoe and Churchill [9] and Ozoe, Sayama and Churchill [16] minimized the computational time by constructing an empirical correction. This construction required a complete computation for three grid-sizes and sufficient values of the parameters to bound the other conditions. Such a construction would have been prohibitive for the problem herein because of the additional parameter of aspect ratio. An alternative procedure was therefore adopted. The calculations were first carried out for the basic grid-size of $\Delta X = \Delta Y = 0{\cdot}1$. A sufficiently large non-dimensional time-step for stability was found to be 0·001. The steady-state temperature and stream-function fields were obtained and $\overline{Nu}$ was calculated at the mid-plane. The grid-size was then cut in two and the values of the temperature and stream-function were estimated for the added points by linear interpolation. $\overline{Nu}$ was recalculated using the interpolated and original values. The resulting value was found by test calculations to be within 2 per cent of that obtained by extrapolation of complete calculations for three grid-sizes to zero grid-size. Calculations were carried out for $W/H = 1, 2, 3, 4$ and ∞ and $Ra = 2000$, 4000 and 8000. The results for the limiting case of zero angle of inclination agreed well with the previously computed values of Samuels and Churchill [18] and Kurzweg [19] for multiple roll-cells.

RESULTS FROM NUMERICAL CALCULATIONS

Representative results obtained by the above procedure are plotted in Fig. 2, which indicates the effect of angle inclination, aspect ratio and Ra for $Pr = 10$. In each case a maximum heat flux was observed at an intermediate inclination. For aspect ratios of 3 and 4 the circulation shifted to multiple roll-cells at small angles of inclination and the computations then became unstable.

As the angle of inclination approaches 180° the circulation ceases and $\overline{Nu}$ correctly approaches unity corresponding to pure conduction at all aspect ratios.

Hollands and Konicek [15] proposed the following equation to represent their experimental data

$$Nu = 1 + K\left[1 - \frac{Ra_c}{Ra}\right]. \tag{12}$$

Values of K and Ra_c from their paper were used to calculate the indicated points and to construct the indicated curves in Fig. 2. Their aspect ratio of 44 may be considered to be essentially infinite. The heat-transfer rate decreases monotonically with increasing angle of inclination up to 90° for this channel. Their Nu is apparently the local value in the central part of the long channel. On the other hand our value is the average one including the effects of mixing in the corners. The radical difference in the results indicates a change in the circulation pattern as the aspect ratio goes from small to large values.

The computed values are plotted vs aspect ratio with angle of inclination as a parameter in Fig. 3. The computed values of Wilkes and Churchill [6] for $Ra = 14\,660$ and $Pr = 0{\cdot}733$ are in qualitative agreement.

The effect of Ra on $\overline{Nu}$ is illustrated in Fig. 4 for several aspect ratios. The effect of the angle of inclination on the velocity field is illustrated in Fig. 5 for $Ra = 4000$ and $Pr = 10$ and an aspect ratio of 1·0.

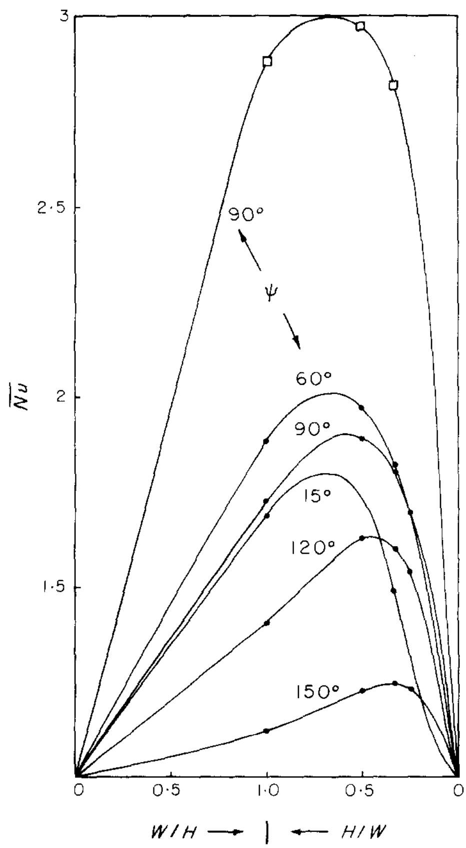

FIG. 3. Computed effect of aspect ratio on rate of heat transfer ●, This work: $Ra = 4000$, $Pr = 10$, by interpolation of values at $\Delta X = \Delta Y = 0{\cdot}1$. □, Wilkes and Churchill, $Ra = 1{\cdot}466 \times 10^4$, $Pr = 0{\cdot}733$, $\Delta X = \Delta Y = 0{\cdot}1$.

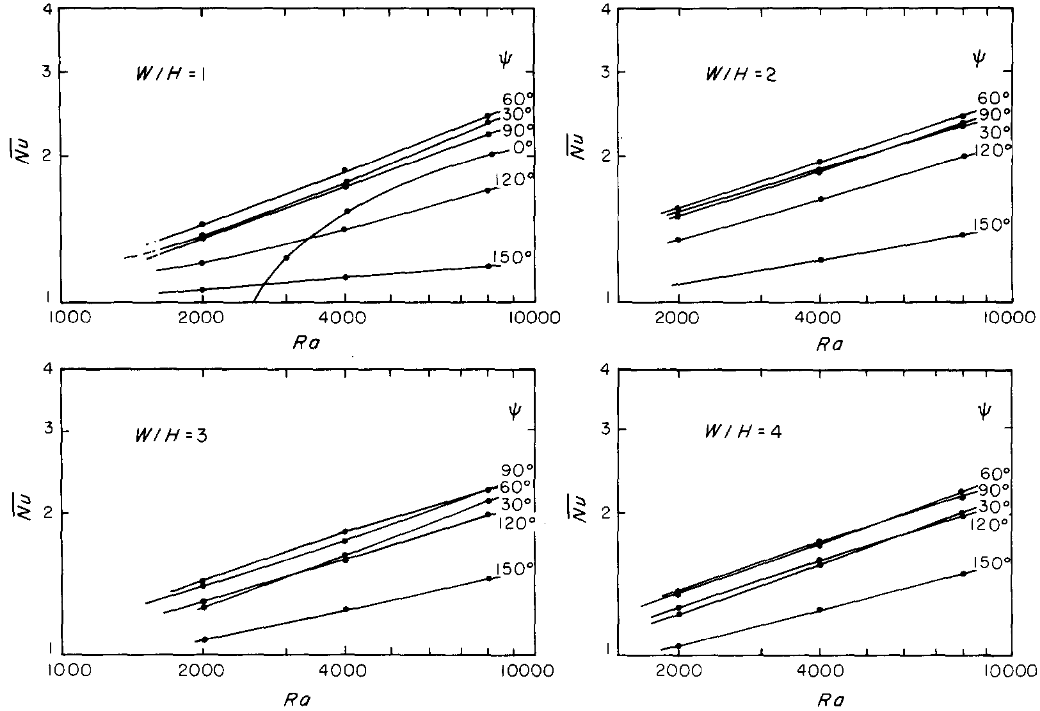

FIG. 4. Computed effect of Rayleigh number on rate of heat transfer ●, This work: by interpolation of values at $\Delta X = \Delta Y = 0{\cdot}1$.

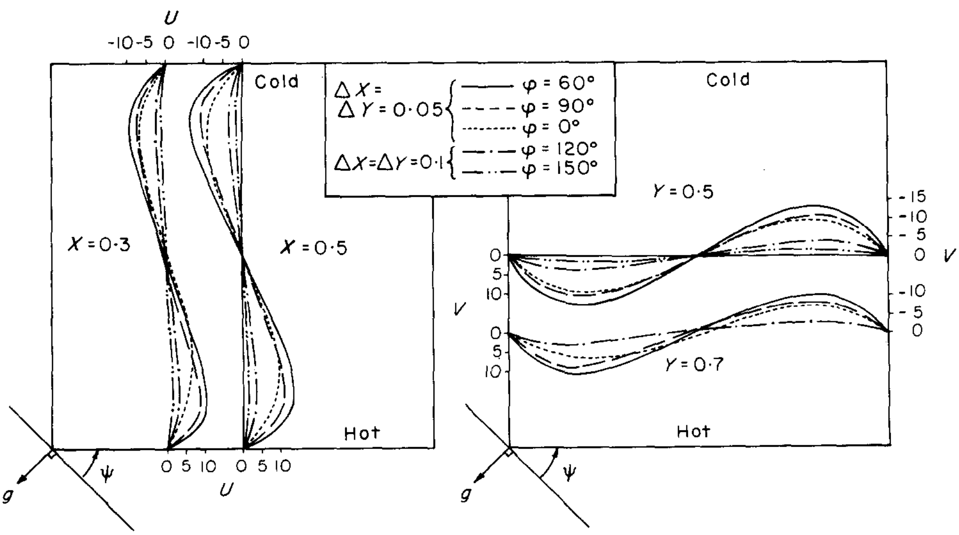

FIG. 5. Computed effect of angle of inclination on velocity field at $Ra = 4000$, $Pr = 10$, $W/H = 1{\cdot}0$.

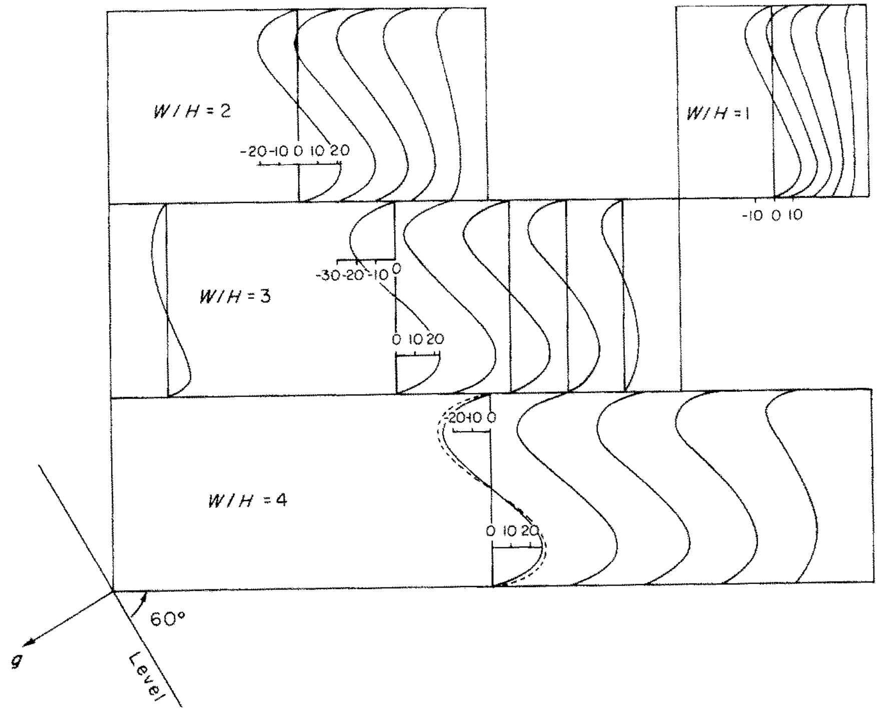

FIG. 6. Computed U-component of velocity at $Ra = 4000$, $Pr = 10$, $\Psi = 60°$ and $\Delta X = \Delta Y = 0{\cdot}1$

$$------, \quad U = \frac{Ra}{6}\left[(0{\cdot}5 - Y)^3 - \frac{0{\cdot}5 - Y}{4}\right]\sin\Psi.$$

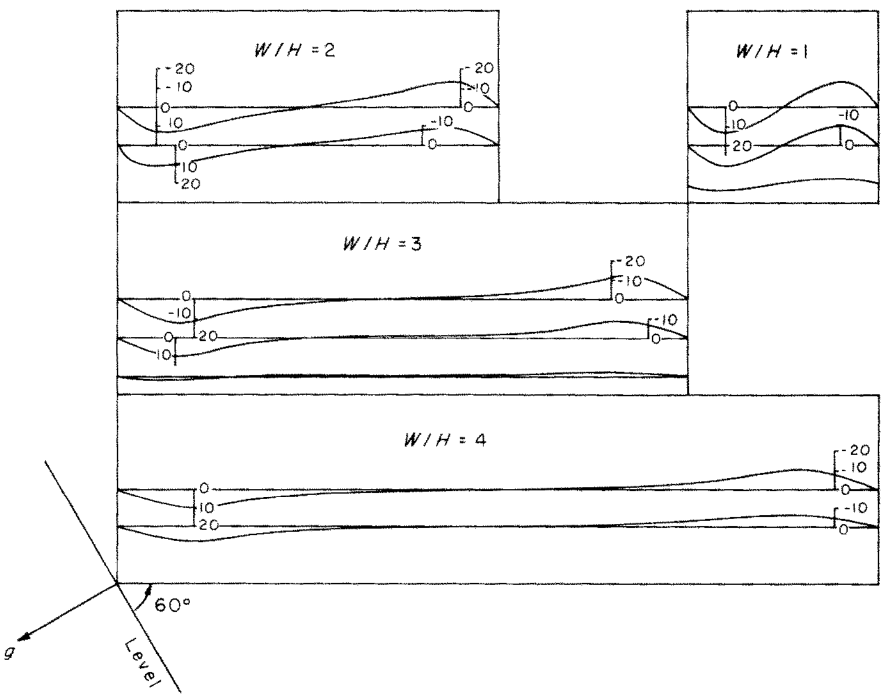

FIG. 7. Computed V-component of velocity at $Ra = 4000$, $Pr = 10$, $\Psi = 60°$ and $\Delta X = \Delta Y = 0{\cdot}1$.

The highest velocity is at 60° which agrees with the maximum in $\overline{Nu}$. The effect of aspect ratio on the velocity field is illustrated in Figs. 6 and 7. The theoretical expression for infinite, inclined vertical plates ($W/H \to \infty$) is (13)

$$U = \frac{Ra}{6}\left[(0{\cdot}5 - Y)^3 - \frac{(0{\cdot}5 - Y)}{4}\right]\sin\Psi. \quad (13)$$

This expression is compared with the computed curve for $W/H = 4$ and $\Psi = 60°$ in Fig. 6. Surprisingly good agreement is apparent.

The contribution of convection to the local heat flux in the Y-direction at several levels is illustrated in Fig. 8 for $Ra = 4000$, $Pr = 10$, $\Psi = 60°$ and $W/H = 2$. Heat transfer at the cooled wall is necessarily wholly by conduction but at the center convection is the primary mechanism.

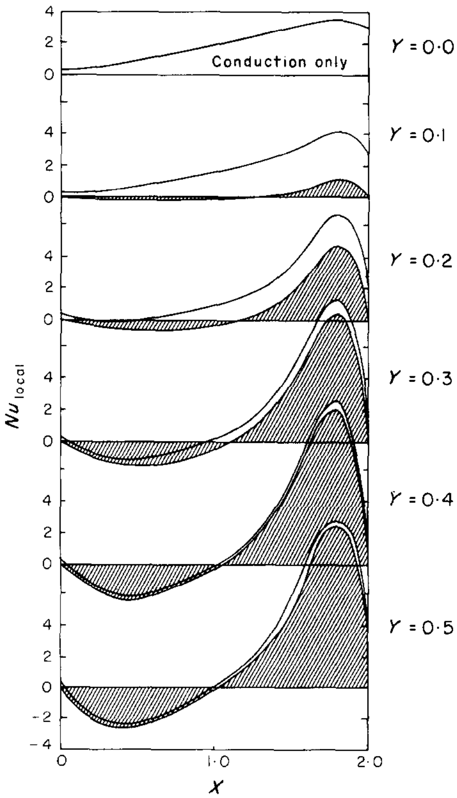

FIG. 8. Computed contribution of circulation of heat flux at $Ra = 4000$, $Pr = 10$, $W/H = 2$, $\Psi = 60°$.

EXPERIMENTAL METHOD

Convection channels with an aspect ratio, W/H, of 1 and 3 were constructed of Plexiglas. The height of the channel was 0·02 m and the length along the z-axis was 0·24 m in both cases. The upper and lower plates were made of 0·01-m-thick copper to ensure a uniform temperature. The principal experimental difficulty was to detect the heat transfer rate precisely because the computed value of $\overline{Nu}$ is sensitive to heat losses. To minimize the effect of ambient changes a constant-temperature chamber, $2 \times 2 \times 1{\cdot}5$ m was constructed and maintained at $301{\cdot}15 \pm 0{\cdot}2$°K by an air-conditioner and heater with a precision mercury regulator. The temperature of the upper plate was established by a cooling jacket through which cooling water at 301·15°K was circulated from a constant-temperature bath of about $5 \times 10^{-2}\,m^3$. The lower plate was heated by Nichrome wire and the heat input was measured with a watt-meter. The rate of heat transfer with no circulation was measured by turning the whole system upside down. From the known thermal conductivity of the working material the heat loss can then be estimated by subtracting the heat flux for pure conduction from the measured heat input. The same heat loss was assumed to occur during the convective heat-transfer experiments for the same total heat input to the system. The temperature difference between the upper and lower plates was measured by copper–constantan thermocouples. A steady state was attained in 18–30 ks.

EXPERIMENTAL RESULTS

Representative experimental results are shown in Fig. 9 for silicone oil ($\mu/\rho = 5 \times 10^{-4}\,m^2/s$ at 298·15°K) and in a channel 0·02-m high × 0·06-m wide × 0·24-m long ($W/H = 3$). The total heat input to the lower plate was 2·38 W.

The experimental values follow the curve representing the computed values except at very low angles of inclination. This discrepancy is associated with the visual observation of a transition from a single roll-cell with its axis parallel to the long dimension of the channel to a series of oblique roll-cells with their axes parallel to the upslope plane but not parallel to each other. For the chosen conditions, $\overline{Nu}$ at $\Psi = 25°$ decreases 30 per cent from the computed value of 1·66 to the experimental value of 1·31.

The decreasing of rate of heat transfer as the angle of inclination from the horizontal plane is increased slightly is presumably due to the greater drag of the oblique cells and hence to a lower rate of circulation.

SUMMARY

Experiments in rectangular channels with aspect ratios of 1 and 3 revealed first a minimum and then a maximum rate of heat transfer as the angle of inclination was increased from 0° to 180°. The behavior changes quantitatively but not qualitatively with $Ra (< 10\,000)$ and aspect ratio.

The maximum rate of heat transfer occurred at $H/W = 0{\cdot}65$ and $\Psi = 60$. The minimum rate of heat

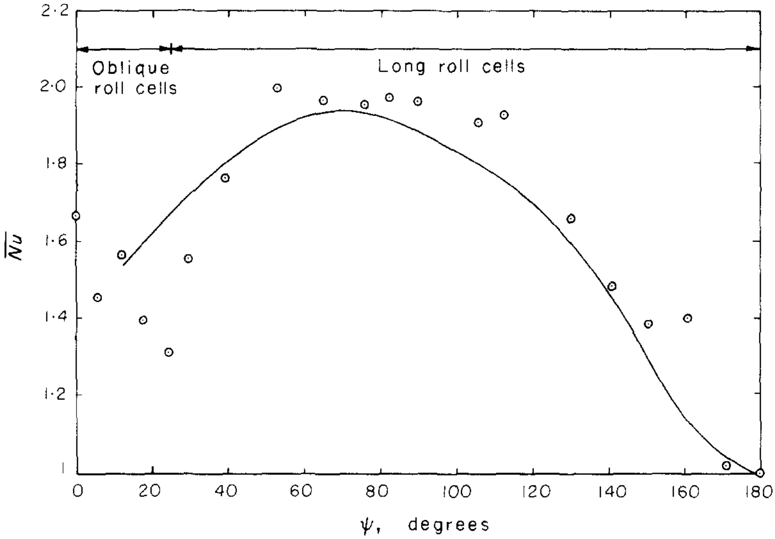

FIG. 9. Comparison of experimental and computed rates of heat transfer at $W/H = 3$. ○, silicone oil ($Pr = 4045$, $Ra = 4770$); —, computed ($Pr = 10$, $Ra = 4800$).

transfer occurred at a small angle of inclination which is a moderate function of aspect ratio and a slight function of Ra. A change in the mode of circulation was observed at the minimum in the heat-transfer rate. The circulation pattern for angles of inclination below this point of transition was observed to be a series of oblique roll-cells.

A mathematical model for a single long roll-cell was constructed and solved numerically for aspect ratios of 1, 2, 3 and 4. The computed value of $\overline{Nu}$ showed general agreement with the experimental results except at small angles of inclination for which a different mode of circulation prevails.

Acknowledgement—We would like to express out thanks for the advice of Professors T. Mizushina and T. Fujii.

REFERENCES

1. Lord Rayleigh, On convection currents in a horizontal layer of fluid, when the higher temperature is on the under side, *Phil. Mag.* Ser. 6, **32**, 529–546 (1916).
2. S. H. Davis, Convection in a box: linear theory, *J. Fluid Mech.* **30**, 465–478 (1967).
3. I. Catton, The effect of insulating vertical walls on the onset of motion in a fluid heated from below, *Int. J. Heat Mass Transfer* **15**, 665–672 (1972).
4. D. K. Edwards, Suppression of cellular convection by lateral walls, *J. Heat Transfer* **91C**, 145–150 (1969).
5. P. A. Jennings and R. L. Sani, Some remarks on thermoconvective instability in completely confined regions, *J. Heat Transfer* **94C**, 234–236 (1972).
6. J. O. Wilkes and S. W. Churchill, The finite-difference computation of natural convection in a rectangular enclosure, *A.I.Ch.E. Jl* **12**, 161–166 (1966).
7. R. K. MacGregor and A. F. Emery, Free convection through vertical plane layers—moderate and high Prandtl number fluids, *J. Heat Transfer* **91C**, 391–403 (1969).
8. A. Acrivos, A theoretical analysis of laminar natural convection heat transfer to non-Newtonian fluids, *A.I.Ch.E. Jl* **6**, 584–590 (1960).
9. H. Ozoe and S. W. Churchill, Hydrodynamic stability and natural convection in Ostwald-de Waele and Ellis fluids: the development of a numerical solution, *A.I.Ch.E. Jl* **18**, 1196–1207 (1972).
10. L. V. McIntire and W. R. Schowalter, Hydrodynamic stability of viscoelastic fluids: importance of fluid model overstability and form of disturbance, *A.I.Ch.E. Jl* **18**, 102–110 (1972).
11. P. H. Holst and K. Aziz, A theoretical and experimental study of natural convection in a confined porous medium, *Can. J. Chem. Engng* **50**, 232–241 (1972).
12. P. H. Holst and K. Aziz, Transient three-dimensional natural convection in confined porous media, *Int. J. Heat Mass Transfer* **15**, 73–90 (1972).
13. J. E. Hart, Stability of flow in a differentially heated inclined box, *J. Fluid Mech.* **47**, 547–576 (1971).
14. J. E. Hart, A note on the structure of thermal convection in a slightly slanted slot, *Int. J. Heat Mass Transfer* **16**, 747–753 (1973).
15. K. G. T. Hollands and L. Konicek, Experimental study of the stability of differentially heated inclined air layers, *Int. J. Heat Mass Transfer* **16**, 1467–1476 (1973).
16. H. Ozoe, H. Sayama and S. W. Churchill, Natural convection in an inclined square channel, *Int. J. Heat Mass Transfer* **17**, 401–406 (1974).
17. J. D. Hellums and S. W. Churchill, Simplification of the mathematical description of boundary and initial value problems, *A.I.Ch.E. Jl* **10**, 110–114 (1964).
18. M. R. Samuels and S. W. Churchill, Stability of a fluid in a rectangular region heated from below, *A.I.Ch.E. Jl* **13**, 77–85 (1967).
19. U. H. Kurzweg, Convective instability of a hydromagnetic fluid within an enclosed rectangular cavity, *Int. J. Heat Mass Transfer* **8**, 35–41 (1965).

CONVECTION NATURELLE DANS UN CANAL RECTANGULAIRE INCLINE CHAUFFE D'UN COTE ET REFROIDI SUR LE COTE OPPOSE

Résumé—On modélise et détermine par la méthode des différences finies, la circulation naturelle bidimensionnelle dans une boite inclinée, chauffée sur une face et refroidie sur la face opposée. L'angle d'inclinaison varie depuis zéro jusqu'à 180° pour quatre rapports de forme. On obtient des résultats expérimentaux en accord avec les estimations théoriques. On observe que le mode préférentiel de circulation change avec l'angle d'inclinaison et le rapport de forme. Il apparait un minimum au point de transition dans le flux thermique et un maximum lorsque l'angle est plus important.

FREIE ZIRKULATION IN EINEM GENEIGTEN RECHTECKKANAL MIT BEHEIZUNG AUF DER EINEN UND KÜHLUNG AUF DER ANDEREN SEITE

Zusammenfassung—Die zweidimensionale, freie Zirkulation in einem geneigten, schmalen Kasten, der auf der einen Seite beheizt und auf der anderen Seite gekühlt ist, wurde als Modell nachgebildet und mit der Methode der finiten Elemente gelöst. Der Neigungswinkel wurde zwischen 0 und 180° für vier Längenverhältnisse verändert. Die erhaltenen experimentellen Ergebnisse stimmen gut mit den theoretischen Vorausberechnungen überein. Es konnte beobachtet werden, daß die Zirkulation sich in erster Linie mit dem Neigungswinkel und dem Längenverhältnis änderte. Ein Minimum der Wärmestromdichte trat im Umschlagspunkt auf, ein Maximum stellte sich bei weiterer Zunahme des Winkels ein.

ЕСТЕСТВЕННАЯ КОНВЕКЦИЯ В НАКЛОННОМ ПРЯМОУГОЛЬНОМ КАНАЛЕ, НАГРЕВАЕМОМ С ОДНОЙ СТОРОНЫ И ОХЛАЖДАЕМОМ С ПРОТИВОПОЛОЖНОЙ

Аннотация — С помощью методов конечных разностей получена модель и решение двумерной естественной конвекции в наклонном ограниченном канале, нагреваемом с одной стороны и охлаждаемом с противоположной. Угол наклона изменяется от 0 до 180° для четырех значений отношений сторон. Полученные экспериментальные данные хорошо согласуются с теоретическими расчетами. Замечено, что преимущественная мода движения изменяется с изменением угла наклона и отношения сторон. В точке перехода имеет место минимальный тепловой поток. При дальнейшем увеличении угла наклона наблюдается максимальный тепловой поток.