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EXPERIMENTAL ANALYSIS OF NATURAL CONVECTION IN A TILTED CHANNEL

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ABSTRACT

Laminar air natural convection between two inclined parallel plates with uniform, both symmetric and fully asymmetric (one wall heated with the other wall unheated), heat flux has been experimentally investigated. Wall temperatures as a function of the height and the tilting angle are presented, for different aspect ratio and heat flux values. Air temperatures and velocities have been measured. Nusselt number is correlated to the inclination angle in the range 60°-90° and to a channel Rayleigh number including the effect of the aspect ratio. A unique correlation which takes into account all heating modes is also proposed.

NOTATION

a, m	coefficients in equation (10)
b	channel spacing, m
g	acceleration of gravity, m/s^2
Gr	channel Grashof number, equation (6)
h	heat transfer coefficient, W/m^{2}K
k	thermal conductivity, W/mK
L	channel length, m
Nu	average Nusselt number, equation (7)
Pr	Prandtl number
q	heat flux, W/m^2
r	regression coefficient
Ra	channel Rayleigh number
T	temperature, °C
v	velocity of the air, m/s
W	channel width, m
x	coordinate along the length, m
y	coordinate along the spacing, m
z	coordinate along the width, m
Greek symbols	
β	volumetric coefficient of expansion, 1/K
θ	angle of inclination from the vertical, °
ν	kinematic viscosity, m^2/s
Superscripts	
+	dimensionless
Subscripts	
b	refers to bottom wall
c	convective
k	conductive
max	maximum
o	ambient air
r	radiative
t	refers to top wall
w	wall
Ω	ohmic dissipation

A bar over a symbol indicates a mean value on the walls

INTRODUCTION

Air natural convection in isoflux walls tilted channels has been scarcely investigated, particularly for low aspect ratios and angles from horizontal. Nearly-horizontal open ended channels are frequently used in electronic packages, such as telephones, modems, desk top applications. In many of them the thermal performance of the channel is of interest.

As far as air laminar natural convection in an open ended vertical channel with uniform wall heat flux is concerned, maximum wall temperatures and Nusselt numbers were correlated to the channel Rayleigh number by Aung et al. (1972) for both symmetric and asymmetric heating and by Wirtz and Stutzman (1982). An experimental investigation on the effect of inclination from the horizontal on the thermal performance of a channel asymmetrically heated with air natural convection was carried out by Kennedy and Kanhel (1983). Temperature profiles of the uniformly heated bottom plate were presented, which show that the higher the inclination angle the lower the temperature, up to a 60° angle. For angles greater than 20° the flow regime changed from a two-dimensional behaviour into three-dimensional longitudinal vortex rolls. Azevedo and Sparrow (1985) performed heat transfer and flow visualization experiments in water to study the effect of inclination on natural convection in a isothermal, parallel walled channel. Various modes of heating, aspect ratios and Rayleigh numbers were investigated. Nusselt numbers were evaluated for all experimental conditions in the range 0°-45° of the inclination angle from the vertical and they were correlated to within ±10 percent. A two-dimensional recirculating flow was observed near the unheated wall of one sided top heated channels. Local and average air natural convection heat transfer characteristics for a uniformly heated upward-facing horizontal plate, shrouded by a parallel adiabatic surface situated above the plate, were determined experimentally by Sparrow and Carlson (1986). The interplate gap, the wall heat flux and the configuration of the surroundings were varied during the experiments. The local Nusselt number was highest at the edge of the heated plate and decreased along the plate surface. Its distributions, when normalized by the corresponding average Nusselt number, were found to be independent of the parameters for gap heigths above a threshold value.

In this paper laminar air natural convection between two inclined parallel plates, with uniform either symmetric or asymmetric (one wall heated and the other wall unheated) heat flux, has been experimentally investigated. Experiments showed a systematic sharp increase in maximum wall temperatures for inclination angles above 60°. In a previous paper (Manca et al. 1992) the range 0°-60° has been investigated. In the following experimental data are presented in the range 60°-90°. Measured wall temperatures along the channel length are presented for different values of aspect ratio, angle of inclination and channel Rayleigh number based on spacing between the plates. Midplane velocity and temperature of the air were measured and their profiles along the spacing, for 0° and 60° tlting angles, are reported. Heat transfer coefficents have been evaluated and Nusselt numbers are correlated to the inclination angle and to the channel Rayleigh number. A unique correlation is proposed, which takes into account all heating modes and can be easily used in practice, when local temperatures and heat transfer coefficients are unknown.

EXPERIMENTAL APPARATUS

The experimental apparatus is shown schematically in fig.1. The channel was made from two principal parallel walls with uniform heat flux; its side walls were unheated. Each principal wall consisted of two sandwiched phenolic fiberboard plates, 400 mm long and 530 mm wide. The plate facing the channel was 3.2 mm thick and its surface adjacent the internal air was coated with a nickel plated copper layer. The low emissivity of nickel (0.05) minimizes radiation effects on heat transfer. The backside plate was 1.6 mm thick. Its back surface was coated with a copper layer, which was the heater. In order to reduce heat losses, a 150 mm polystyrene block was affixed to the rear face of each plate. The side walls were made of plexiglass rectangular rods. The channel was 400 mm high and 475 mm wide and was open to the ambient along its top and bottom edges, Fig.1b. It was secured to a tilting support frame. Discrete inclination angles of the channel in the range 0°-90° were obtained by brackets and pinned joints, with an accuracy of ±0.5°. A finer adjustment (±0.05°) was obtained in the range 85°-90° from the vertical by a micrometric screw system. The entire apparatus was located within an enclosed room, sealed to eliminate extraneous air currents.

The plates were heated by passing a direct electrical current through the heaters. The dissipated heat flux per board was evaluated with an accuracy of ±2 percent by measuring the voltage drop across the heaters and the current passing through them. Wall temperatures were measured by twelve 0.50 mm OD ungrounded iron-constantan thermocouples embedded in the fiberboard plates and in contact with the outer layer. They were located in the vertical center line of each plate at different heights above the bottom edge. In preliminary tests, carried out at all inclination angles, additional thermocouples placed horizontally outward the xy plane indicated that the wall temperature distribution was symmetric with reference to z=0 within ±0.2°C. Moreover, wall temperatures turned out to be independent of z within ±100 mm, their maximum deviation from the vertical centerline temperature being ±2 percent. Within the range of inclination angle 30°-70°, for which Kennedy and Kanhel (1983) and Azevedo and Sparrow (1985) observed three-dimensional longitudinal vortex rolls, no variation of the wall temperature along the z axis higher than the aforementioned ±2 percent was measured. Therefore, in the following reference has been made to the centerline wall temperature. Fifteen thermocouples were affixed to the rear surface of the plates and embedded in the styrene to enable evaluation of conductive heat losses. Their maximum value was 20 percent of the ohmic heat rate dissipated in each plate, in the range of inclination angle 60°-90°. Ambient air temperature was measured by shielded thermocouples placed near the leading edge of the channel. Thermocouples voltages were recorded to 1 μV. Calibration of the temperature measuring system showed an estimated precision of the thermocouple-readout system of ±0.1°C. The velocity and the temperature of the air in the channel were measured by a hot wire system. In every case the air velocity could be determined wtih a maximum uncertainty of ±10 percent. The air temperature was measured with a maximum uncertainty of ±0.1°C.

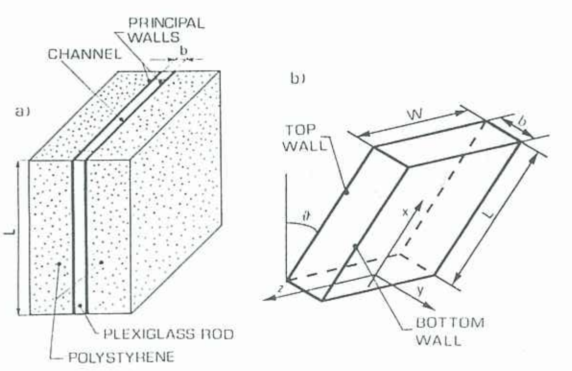

Fig.1. Experimental apparatus: a) test section; b) geometry of the channel.

A more detailed description of the experimental apparatus is reported in Manca et al. (1992).

DATA REDUCTION

The dimensionless coordinates are

$$x^+ = x/L \ ; \quad y^+ = y/b \tag{1}$$

The dimensionless wall temperature is

$$T_w^+ = \frac{T_w - T_0}{\bar{q}_c\, b / k} \tag{2}$$

where $\bar{q}_c$ is the mean value of the spatially-averaged convective heat flux on each heated wall

$$\bar{q}_c = \frac{1}{2L}\left[\int_0^L q_{c,t}(x)\,dx + \int_0^L q_{c,b}(x)\,dx\right] \tag{3}$$

The dimensionless temperature of the air is

$$T^+ = \frac{T - T_0}{T_{w,max} - T_0} \tag{4}$$

where $T_{w,max}$ denotes the maximum surface temperature on the two walls.

The air velocity is made non dimensional with reference to its maximum value at a given x

$$v^+ = v / v_{max} \tag{5}$$

Heat transfer results are presented in terms of the so called channel Rayleigh number

$$Ra = Gr\,Pr = \frac{g\,\beta\,\bar{q}_c\, b^5}{\nu^2\, k\, L}\,Pr \tag{6}$$

Nusselt numbers will be based on the difference between wall and inlet fluid temperatures rather than on that between wall and bulk fluid temperatures, since the last one usually is not available in practice.

An average Nusselt number can be defined as

$$Nu = \frac{\bar{q}_c}{\bar{T}_w - T_o}\,\frac{b}{k} \tag{7}$$

with

$$\bar{T}_w = \frac{1}{2L}\left[\int_0^L T_{w,t}(x)\,dx + \int_0^L T_{w,b}(x)\,dx\right] \tag{8}$$

The thermophysical properties of the air are evaluated at the reference temperature $(\bar{T}_w + T_o)/2$.

Local actual convective heat flux, $q_c(x)$, is not uniform because of radiation and conduction. Experimental data were reduced by first introducing, in the above presented equations, the local heat flux

$$q_c(x) = q_\Omega(x) - q_k(x) - q_r(x) \tag{9}$$

where $q_\Omega(x)$ is the local heat flux due to the ohmic dissipation, which is assumed to be uniform, $q_k(x)$ are the local conduction heat losses from the plate and $q_r(x)$ is the local radiative heat flux from the plate. For each run, the terms $q_k(x)$ were calculated by a numerical procedure, a three-dimensional distribution of the temperature being assumed in the styrene. The predicted temperatures in significant configurations of the system had been previously compared with those measured by thermocouples embedded in the polystyrene insulation and the agreement was very good. The $q_r(x)$ terms were calculated for each temperature distribution of the walls, ambient temperature and channel spacing, according to the procedure described by Webb and Hill (1989). With both walls heated, for θ=0° the minimum and the maximum values of the ratio $q_c(x)/q_\Omega(x)$ were respectively 0.75 at x=350 mm and 0.94 at x=150 mm; for θ=90° they were 0.59 at x=50 mm and nearly 0.85 in the region around x=200 mm.

The uncertainties in Nusselt and channel Rayleigh numbers are estimated, according to the procedure suggested in Kline and McClintock (1953), at 12 and 15 percent, respectively.

RESULTS AND DISCUSSION

Experiments were performed at different angles of inclination (0, 60, 75, 85, 88, 90°) and various spacings (20.00, 32.25 and 40.00 mm). Ohmic heat fluxes varied between 14 and 250 W/m^2. The cooling medium was air, whose inlet temperature into the channel was nearly 26.6°C in all tests; its fluctuations were within ±0.5°C during each test. Three heating modes were studied for each configuration. According to what proposed by Azevedo and Sparrow (1985), they will be designated as heating modes I, II and III and are: I) both walls heated; II) top wall heated with bottom wall unheated; III) top wall unheated with bottom wall heated. The top and bottom wall designations are referred to the relative positions of the walls shown in fig.1b. When the channel is vertical, modes II and III represent the same heating mode: IV) one wall heated with the other wall unheated.

Wall temperature rise above ambient temperature as a function of the length and the angle of inclination, for different spacings, ohmic heat fluxes and heating modes, are presented in fig.2. The figure shows that up to θ=88°, the higher the angle of inclination the higher the temperature profile of both the walls, due to the reduced chimney effect. When the plates are horizontal, there is no chimney effect. In this case the flow originates at both the edges of the bottom plate and the temperature profiles along the walls are symmetric. The temperatures of the bottom wall near the exit of the 88° inclined channel turn out to be always higher than those at 90°, due to air inflow at the leading edge of the bottom plate. For modes I and II the air outflow at the exit edge of the upper plate determines wall temperatures nearly equal at 88° and 90° inclination angles. For mode I (symmetric heating) the difference between the temperatures of the top and the bottom plates increases at increasing tilting angles as a consequence of the continuous enhancement and reduction in convective heat transfer, respectively from the lower and the upper wall. For mode II (top wall heated and bottom wall unheated) the aforementioned temperature difference is independent of inclination angle at the lower spacing value. At the same heat flux and at the higher spacing the wall temperature difference remains constant up to a 75° angle whereas at higher angles it increases at increasing inclination angles. For b=32.25 mm and q_Ω=121 W/m^2 fig.2b shows that at any angle the higher θ the higher the difference between the temperatures of the two walls. This is due to the increased temperature of the top wall while those of the bottom one (not reported in the figure) are nearly equal to those of the intermediate case. For mode III (top wall unheated and bottom wall heated) fig.2c exhibits, at any spacing and heat flux, maximum wall temperatures lower than those for mode II but the slope of temperature profiles is higher for mode III than for mode II. The temperature profile of the heated wall in mode III is similar to that of the bottom wall in mode I, since in both heating modes this plate faces air cooler than in mode II.

The midplane dimensionless velocity and temperature of the air as a function of the dimensionless coordinate along the spacing for b=20.00 mm and symmetric heating at various tilting angles and Rayleigh number are presented, respectively, in figs. 3 and 4. Fig.3 shows that for θ=60° and x^+=0.875 the maximum velocity of the air is still attained near the midplane and its profile is almost symmetric for both the Rayleigh numbers. Near the channel exit (x^+=0.9875) the velocity profiles at both Rayleigh numbers are skewed towards the top wall, due to the rising plume and its temperature higher than that of the opposite wall. The asymmetric velocity distribution determines a chimney effect in the tilted channel smaller than that in the vertical one though the values of the maximum

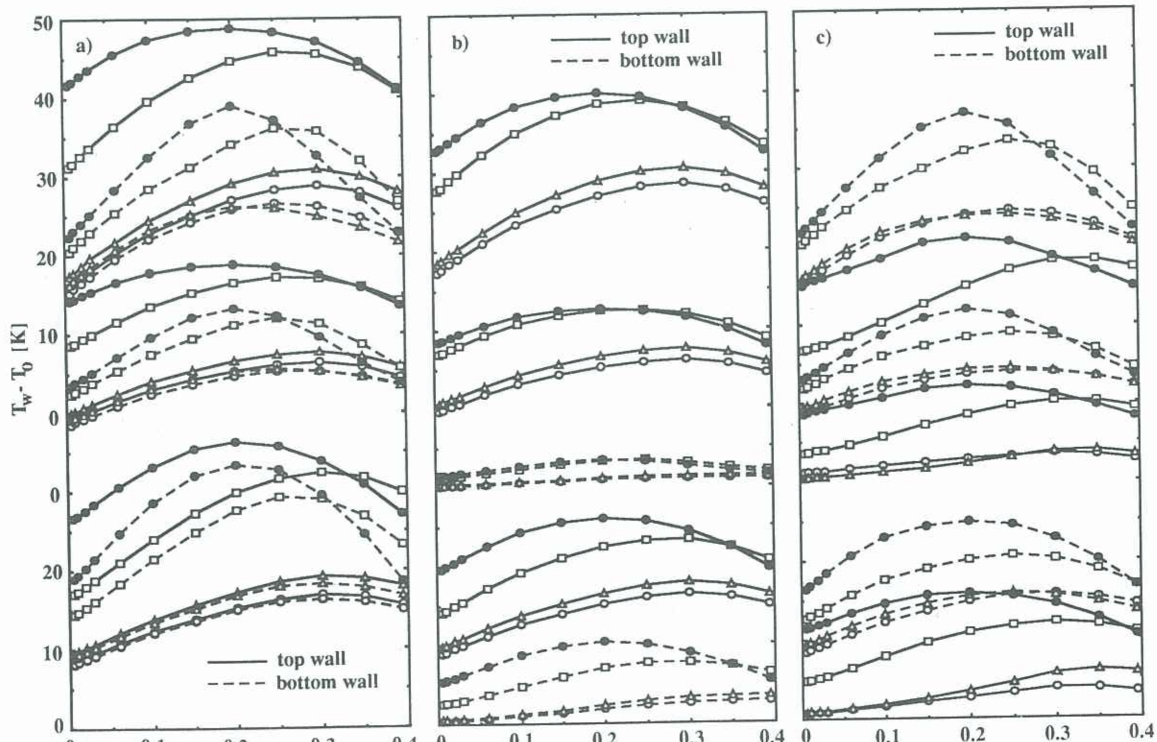

Fig.2. Wall temperature rise vs length. In the lower part: q_Ω=60 W/m^2, b=20.00 mm; in the middle part: q_Ω=60 W/m^2, b=32.25 mm; in the upper part: q_Ω=121 W/m^2, b=32.25 mm. a) both walls heated; b) top wall heated and bottom wall unheated; c) bottom wall heated and top wall unheated. O 60°, △ 75°, □ 88°, ● 90°.

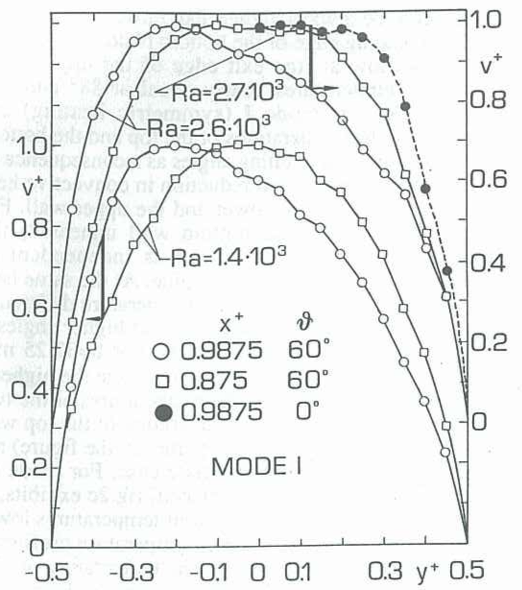

Fig.3. Midplane velocity of the air vs. coordinate along the spacing.

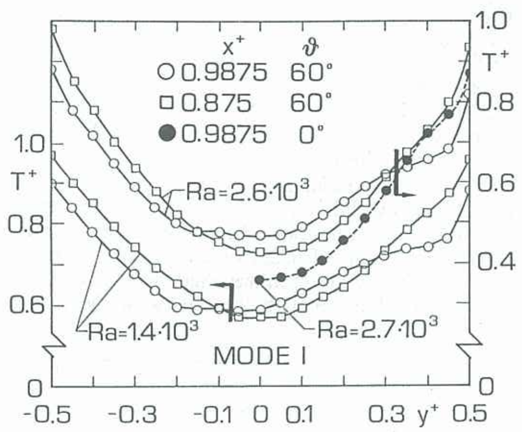

Fig.4. Midplane temperature of the air vs. coordinate along the spacing.

velocity are not very different (0.286 m/s at θ=60° and 0.324 m/s at θ=0°). One can also notice that at x^+=0.875 the velocity profile for the smaller Rayleigh number (q_Ω=60 W/m^2) is more developed than that for the higher one (q_Ω=121 W/m^2). Furthermore, it is worthwile pointing out that velocity field turns out to be developing also at the higher Rayleigh number. The same behaviour was experienced in a vertical channel by Manca and Naso (1990). This is in agreement with the experimental results by Wirtz and Stuzman (1982) but in contrast with the numerical predictions by Aung et al (1972). Fig.3 shows that the higher the Rayleigh number the higher the velocity of the air, its maximum values being respectively: for Ra=$1.4\cdot10^3$, at x^+=0.875 v_{max}=0.171 m/s and at x^+=0.9875 v_{max}=0.215 m/s; for Ra=$2.6\cdot10^3$, at x^+=0.875 v_{max}=0.226 m/s and at x^+=0.9875 v_{max}=0.286 m/s; for Ra=$2.7\cdot10^3$ v_{max}=0.324 m/s. Fig.4 shows that air temperature near the bottom wall is nearly uniform along the spacing. Near the two walls the temperature of the air at x^+=0.875 is higher than that x^+=0.9875 whereas in the middle of the channel the temperatures at the exit are higher than those at x^+=0.875. At θ=60° the air temperature is always higher than that at θ=0°, except that in proximity of the bottom wall. The maximum wall temperature rises above the ambient one were, respectively, 29.8K at θ=0° and 28.9K at θ=60°.

The average Nusselt number was correlated to the channel Rayleigh number and the inclination angle, for any heating mode in the range 60°-90° of the tilting angle, by the equation

$$Nu = a\,[\,Ra\cos(\theta - 2)\,]^m \qquad (10)$$

The coefficients in equation (10) and the regression coefficients are reported in table I. The same equation correlates data for all modes of heating by the coefficients in the last line of the table. A better fit of experimental data was obtained by correlations in terms of Ra·cosθ rather than in terms of Ra. Diminishing the tilting angle by a constant value allows to correlate also experimental data at θ=90°. The value introduced in equation (10) is the one which gives the highest regression coefficients.

Table I. Coefficients in equation (10).

Mode	a	m	r^2
I	0.504	0.251	0.983
II	0.585	0.239	0.986
III	0.467	0.272	0.985
All	0.519	0.253	0.979

CONCLUSIONS

Experiments confirmed that, in the range 60°-90° of the inclination angle from the vertical, the higher the angle the worse the thermal performance of a channel. For a symmetric heating the ratio between the maximum wall temperature rise in the 90° and 60° tilted channels was 1.86 for q_Ω=60 W/m^2 and b=20.00 mm and 1.39 for q_Ω=121 W/m^2 and b=32.25 mm.

Near the outlet, air temperature turned out to increase along the channel in its middle part whereas it decreased in proximity of the walls. Still near the exit, velocity profiles were skewed towards the top wall.

The Nusselt number was correlated to the inclination angle and the channel Rayleigh number by a unique equation for all heating modes. The best fit of experimental data was achieved in terms of Ra·cos(θ-2).

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