Int. J. Heat Mass Transfer. Vol. 28, No. 7, pp. 1337–1344, 1985
Printed in Great Britain

0017-9310/85 $3.00+0.00

Experimental study of mixed convection heat transfer for transitional and turbulent flow between horizontal, parallel plates

D. G. OSBORNE and F. P. INCROPERA
Heat Transfer Laboratory, School of Mechanical Engineering, Purdue University, W. Lafayette, IN 47907, U.S.A.

(*Received 7 August 1984 and in final form 9 January 1985*)

Abstract—Experiments have been performed to determine the effect of buoyancy on convection heat transfer for transitional and turbulent water flow between horizontal parallel plates. In transitional flow, Nusselt numbers at the top and bottom plates decrease and increase, respectively, with increasing heat flux. Results for the top plate are attributed to boundary-layer laminarization, while those for the bottom plate are attributed to heat transfer enhancement by free convection. For turbulent flow, top plate heat transfer data are correlated by an expression for pure forced convection, while bottom plate data are correlated by an expression for mixed convection.

1. INTRODUCTION

DUE TO THE significant effect which buoyancy is known to have on hydrodynamic and thermal conditions, mixed convection has been widely considered for laminar, internal flows. Numerous experimental and theoretical studies for horizontal tubes and channels have revealed the nature of the buoyancy-induced flow and its effect on enhancing heat transfer, reducing the thermal entry length and promoting early transition to turbulence. A recent review of the literature on this subject is provided by Osborne and Incropera [1].

Because buoyancy effects are less significant, fewer studies of mixed convection have been performed for turbulent, internal flow [2]. For vertical tubes, studies have been concerned with the influence of buoyancy on relaminarization and the enhancement (or reduction) of heat transfer [3–6]. For horizontal tubes, heat transfer is enhanced, but by amounts which are much less than those associated with laminar flow.

For horizontal, turbulent flow, Petukhov [2] reported that large Grashof numbers ($Gr \approx 10^8$) are needed to appreciably ($>10\%$) influence the circumferentially-averaged Nusselt number. He also reported large circumferential variations in the Nusselt number, which increased with increasing Grashof number. These results were confirmed in the study of Petukhov *et al.* [7], who also observed a significant effect of buoyancy on the structure of the turbulence. Observations concerning the effect of buoyancy on velocity profiles and heat transfer enhancement [2, 7] have been corroborated by a numerical solution for turbulent air flow in a horizontal tube with uniform heat flux [8]. Experiments have also been performed for the transitional flow of water in a horizontal tube with uniform heat flux [9]. Buoyancy-induced and hydrodynamic forms of turbulence were observed, and the Rayleigh number for which buoyancy effects became significant increased with increasing hydrodynamic turbulence.

Although buoyancy effects are less pronounced for transitional and turbulent flows than for laminar flow, the effects may still be significant and much remains to be learned of their specific nature. This statement is especially true for flow between horizontal, parallel plates. Results have not been reported for this particular geometry, and the objective of the present study has been to determine experimentally the effects associated with heating at the top and bottom plates. While heating from below may induce a buoyancy flow which enhances heat transfer, heating from above can stratify the flow and inhibit heat transfer. Hence, a major objective of the study has been to determine the effects of thermal destabilization and thermal stratification on heat transfer at both the top and bottom surfaces. Experiments have been performed for water flow in a horizontal channel and have involved flow visualization, as well as temperature and heat transfer measurements.

2. EXPERIMENTAL PROCEDURES

Experiments were performed using the water channel of Fig. 1. The channel is 0.305 m wide and has developing and test sections which are 1.05 m and 0.97 m long, respectively. Although two heights (20 mm and 60 mm) were considered, aspect ratios were sufficiently large (15.25 and 5.08) to permit the assumption of parallel-plate conditions for the longitudinal midplane of the channel. All channel walls were constructed from 12.7-mm Plexiglas, except the top and bottom of the test section, which were fabricated from 9.5-mm aluminum plate.

Test section surface temperature measurements were obtained from 19 thermocouples imbedded in the bottom plate and 23 thermocouples in the top plate. Beginning at $z = 51$ mm, plate temperatures were measured at 111-mm intervals along the midline and at selected stations off the midline. Vertical temperature

NOMENCLATURE

c_p	specific heat
D	diameter
g	acceleration due to gravity
Gr_H	Grashof number, $g\beta(T_{bp}-T_m)H^3/\nu^2$
$Gr_{H,q}$	Grashof number, $g\beta q_b H^4/k\nu^2$
H	channel height
h	local convection heat transfer coefficient
k	thermal conductivity
Nu_H	Nusselt number, hH/k
Pr	Prandtl number, ν/α
q	heat transfer rate per unit area
Re_H	Reynolds number, $w_m H/\nu$
T	temperature
w	longitudinal velocity
z	longitudinal coordinate.

Greek symbols

α	thermal diffusivity
β	thermal expansion coefficient
ρ	density
ν	kinematic viscosity.

Subscripts

b	bottom plate
F	pure forced convection
i	inlet condition
m	mean value
N	pure natural or free convection
p	plate condition
t	top plate.

distributions in the water were determined using thermocouples mounted to a traversing mechanism. By coupling a reversible motor to a potentiometer, the thermocouples could be traversed at 10 mm min^{-1} and the temperature and position outputs measured on an x–y recorder.

Heating was maintained by Electrofilm patch heaters bonded to the upper and lower surfaces of the top and bottom aluminum plates, respectively. A uniform heat flux was assumed for the plate–water interface, and the validity of the assumption was tested by performing a two-dimensional analysis of plate conduction. The analysis accounted for longitudinal conduction in the plate and for heat loss at the junction between the plate and the developing section. Deviations in the uniform flux condition were determined to be less than 5% at the first measurement station ($z = 51$ mm) and less than 3% at subsequent stations. Heat fluxes to the two plates were independently controlled and could be varied from 0 to 6000 W m^{-2}. The outer surfaces of the heaters were insulated, and heat loss to the surroundings was estimated to be less than 1%.

A flowmeter, valve and pump were used to control and measure the mean velocity of water flow through the test section, and a flow straightener was used to achieve a uniform velocity profile at the inlet to the developing section. Hydrodynamic development within this section is sufficient to ensure fully-developed flow at $z = 0$ for transitional and turbulent flow conditions. Flow was visualized through the side walls by using the shadowgraph technique, which provides an indication of spanwise average conditions.

At the bottom and top surfaces, convection coefficients were determined from

$$q = h(T_p - T_m) \tag{1}$$

where the heat flux and plate temperature were measured quantities and the mean temperature was determined from the energy balance

$$T_m(z) = T_m(z=0) + \frac{(q_t + q_b)z}{w_m H \rho c_p}. \tag{2}$$

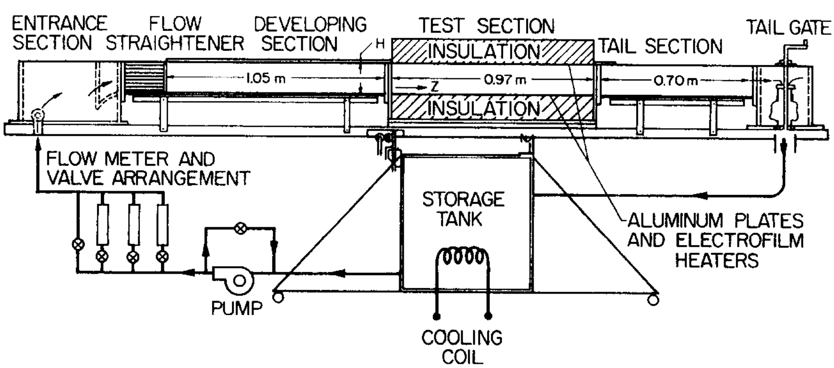

FIG. 1. Schematic of flow channel.

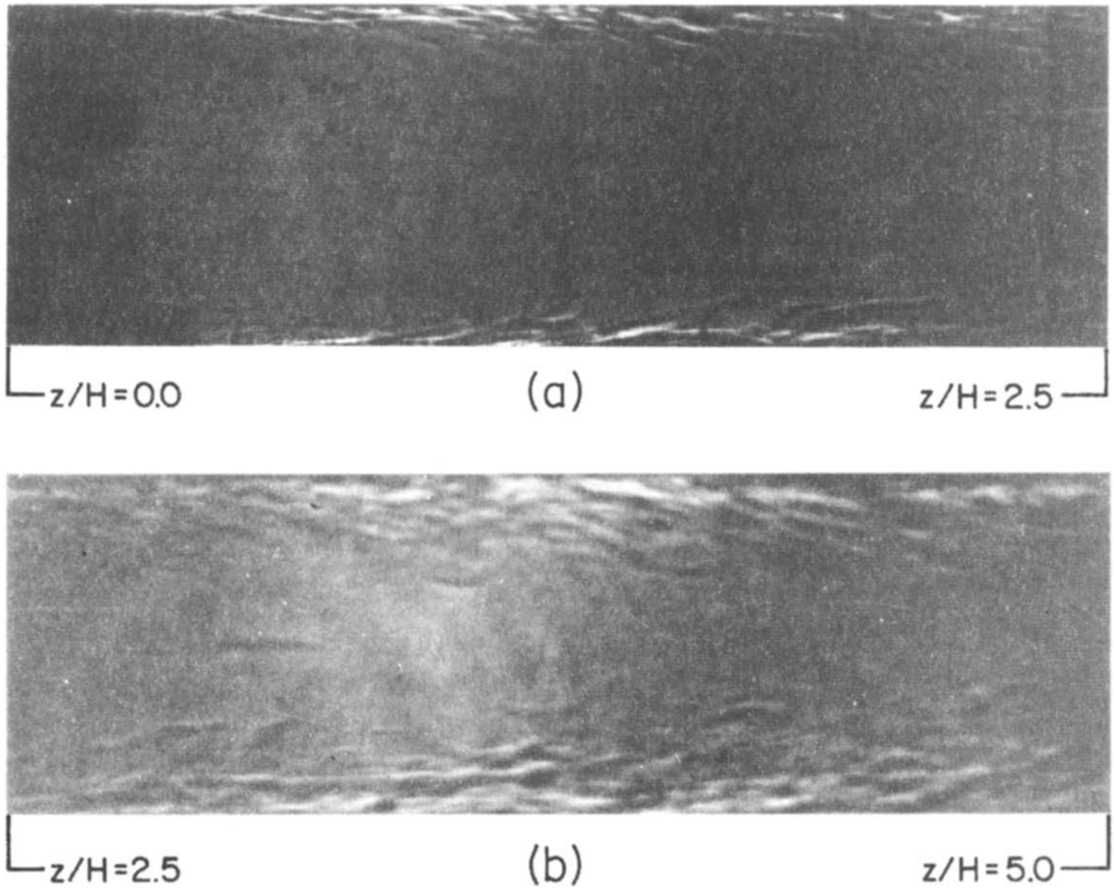

FIG. 2. Shadowgraphs for symmetric heating with $q_t = q_b = 6000$ W m^{-2}, $w_m = 100$ mm s^{-1}, $H = 60$ mm, $Re_H = 6500$ and $Gr_{H,q} = 4.2 \times 10^8$.

Top and bottom heat transfer coefficients were calculated at the eight longitudinal stations corresponding to measurement of T_p, and results were correlated in terms of the Nusselt (Nu_H), Reynolds (Re_H), and Grashof ($Gr_{H,q}$) numbers. All plate temperature measurements used with equation (1) were made along the midline in order to minimize the influence of side wall effects and to maximize applicability of the results to infinite parallel plates. All properties were evaluated at the local mean temperature, $T_m(z)$. Experiments were performed for the range of operating conditions $0 \leqslant (q_b, q_t) \leqslant 6000$ W m^{-2} and $20 \leqslant w_m \leqslant 300$ mm s^{-1}. Grashof and Reynolds numbers were in the range $8.6 \times 10^5 \leqslant Gr_{H,q} \leqslant 2.8 \times 10^8$ and $1400 < Re_H < 6500$. The Reynolds number range corresponds to transitional ($1400 < Re_H < 5000$) and fully turbulent ($Re_H > 5000$) flow [10,11].

Using established procedures [12], uncertainties in Nu_H, were estimated to be in the range from 6% to 30%. The larger uncertainties are due to small temperature differences ($T_p - T_m$) associated with the low heat flux experiments. Uncertainties in $Gr_{H,q}$ and Re_H are in the ranges 8–21% and 2–5%, respectively.

3. RESULTS

3.1. *Flow visualization and temperature distributions*

Representative shadowgraphs are shown in Fig. 2 for fully turbulent flow with symmetric heating. Top and bottom thermal boundary layers are revealed by bright streaks which are associated with changes in boundary-layer density gradients. The results suggest that, to a first approximation, top and bottom flow conditions are symmetrical and that free convection effects are negligible. This behavior is in sharp contrast to the highly asymmetrical conditions associated with laminar flow [1]. In laminar flow, forced convection boundary-layer conditions are associated with the top plate, while mixed convection effects cause the bottom boundary layer to be substantially thicker and to be characterized by random behavior. Careful observation of the shadowgraphs for transitional and turbulent flow conditions revealed bottom boundary-layer thicknesses slightly larger than those for the top boundary layer, suggesting a small influence of free convection effects.

Similar trends are revealed by measured vertical temperature distributions, and representative results

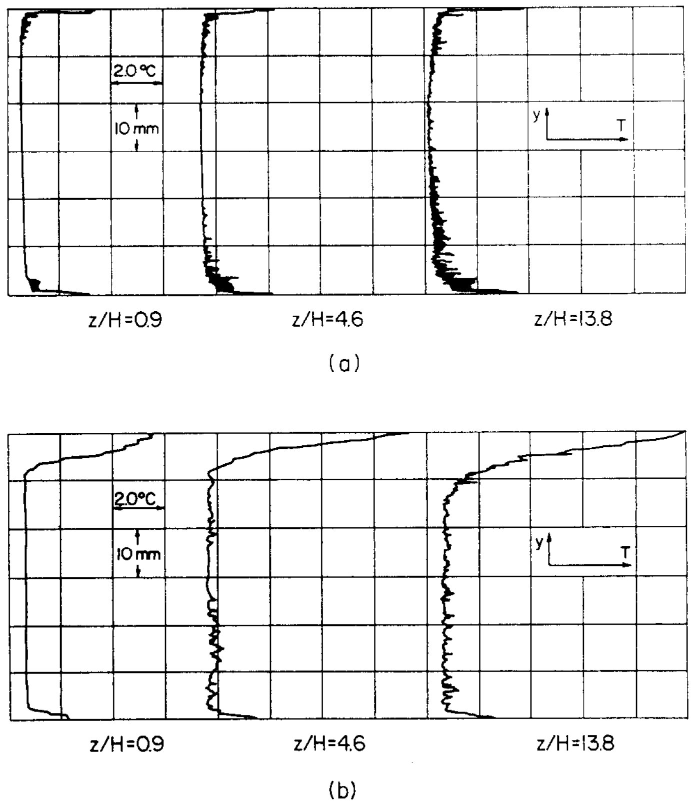

Fig. 3. Vertical temperature distributions for symmetric heating in (a) turbulent flow ($q_t = q_b = 4000\ \mathrm{W\ m^{-2}}$, $w_m = 100\ \mathrm{mm\ s^{-1}}$, $H = 60\ \mathrm{mm}$, $Re_H = 6500$, $Gr_{H,q} = 2.8 \times 10^8$); and (b) laminar flow ($q_t = q_b = 1000\ \mathrm{W\ m^{-2}}$, $w_m = 10\ \mathrm{mm\ s^{-1}}$, $H = 60\ \mathrm{mm}$, $Re_H = 650$, $Gr_{H,q} = 7.0 \times 10^7$).

associated with turbulent and laminar flow are shown in Figs. 3(a) and (b), respectively. In Fig. 3(a) the distributions are nearly symmetrical, except that with increasing z/H, mixing becomes more intense (the amplitude and frequency of temperature fluctuations become larger) near the bottom surface and the top surface temperature becomes slightly larger than that of the bottom surface. The trends are attributed to the enhancement of mixing by buoyancy forces associated with the unstable temperature gradient at the bottom surface. In Fig. 3(b), the asymmetry is much more pronounced, as buoyancy-driven flows significantly enhance heat transfer from the bottom plate, reducing its temperature relative to that of the top plate. Temperature fluctuations are due entirely to random, buoyant flows which originate from the bottom plate. With increasing z/H, buoyancy-induced mixing spreads through most of the channel, although penetration to the top plate is inhibited by stable stratification of the top boundary layer. The smooth portion of the temperature profile at $z/H = 4.6$ is attributed to the time required to complete a vertical temperature scan and the fact that, in inlet regions of the channel, vertical penetration of the buoyancy flow varies with time.

For the transitional and turbulent flows, temperature fluctuation measurements were used to estimate the onset of buoyancy effects at the bottom plate in terms of the mixed convection parameter $[(z/H)^{0.165} Gr_{H,q}^{3/4}/Re_H^{2.4} Pr^{1/4}]$, which is developed in section 3.2. Onset of the effects was found to correspond to a parameter value of approx. 0.001.

3.2. *Heat transfer results*

Transitional flow. Trends associated with heat transfer data obtained for the top plate under transitional flow conditions ($1400 < Re_H < 5000$) differ from results obtained for laminar flow [1]. In laminar flow, the top plate thermal boundary layer is sufficiently stratified to resist significant penetration by buoyancy-driven flows originating from the bottom plate. Hence bottom heating has a negligible effect on top plate boundary-layer conditions, which are dominated by forced convection and are well correlated by an expression developed for forced convection heat transfer between parallel plates [13]. In addition, the top plate convection coefficient is independent of the heat flux. This behavior is shown in Fig. 4 for $Re_H = 650$, where a four-fold increase in q_t has no effect on the value of $Nu_{H,t}$. Similar results are associated with turbulent flow ($Re_H = 6500$).

For transitional flow, Fig. 4 reveals a substantial decrease in $Nu_{H,t}$ with increasing q_t. The trend is associated with all of the transitional flow data of this study and is attributed to the effect which the top plate thermal boundary layer has on flow conditions near the

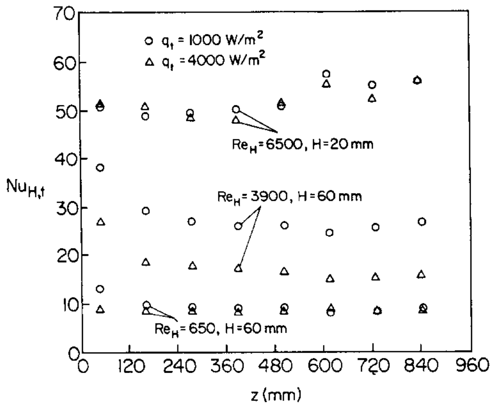

FIG. 4. The effect of top plate heating on top plate Nusselt number for symmetric heating ($q_t = q_b$) and laminar flow, $Re_H = 650$ ($H = 60$ mm), transitional flow, $Re_H = 3900$ ($H = 60$ mm), and turbulent flow, $Re_H = 6500$ ($H = 20$ mm).

top plate. Due to the stable temperature distribution, the density of an upward-moving fluid parcel exceeds that of the surrounding fluid and its motion is retarded. Heating from above therefore has a laminarizing effect on flow conditions near the top plate. This effect was found to increase with increasing heat flux up to $q_t \approx 4000$ W m^{-2}. Nusselt number distributions for $q_t = 4000$ W m^{-2} are in good agreement with an existing correlation for laminar, forced convection between parallel plates [13], suggesting that laminarization of the top plate boundary layer is complete for this heat flux.

Although increasing q_t laminarizes the top plate boundary layer and decreases $Nu_{H,t}$, the opposite effect occurs at the bottom plate with increasing q_b. Buoyancy-driven secondary flows due to bottom heating are known to hasten transition from laminar to turbulent flow and to enhance convection heat transfer [14–16]. This behavior is illustrated in Fig. 5. For symmetrical heating of the top and bottom plates ($q_b = q_t$), $Nu_{H,b}$ increases with increasing q_b and is much larger than $Nu_{H,t}$. Moreover, due to opposing effects on heat transfer of buoyancy-induced flow at the bottom plate and laminarization at the top plate,

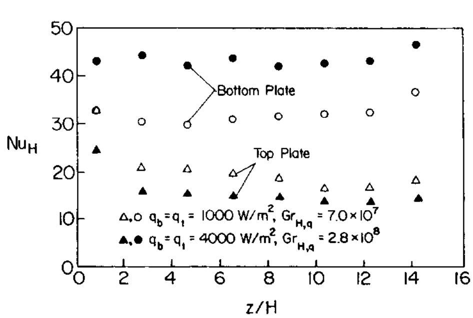

FIG. 5. Top and bottom plate Nusselt numbers for transitional flow ($H = 60$ mm, $Re_H = 2600$) with $q_b = q_t$.

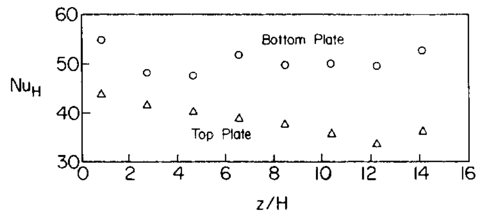

FIG. 6. Top and bottom plate Nusselt numbers for turbulent flow ($H = 60$ mm, $Re_H = 5200$) with $q_b = q_t = 4000$ W m^{-2} and $Gr_{H,q} = 2.8 \times 10^8$.

the difference in the Nusselt numbers ($Nu_{H,b} - Nu_{H,t}$) increases with increasing $q_b = q_t$.

Experiments involving asymmetric heating were also performed to determine the effect of heating at one plate on flow conditions and convection heat transfer at the opposite plate. For the range of conditions considered in this study, the effect was negligible.

Turbulent flow. Although the effects of buoyancy on forced convection are known to be much less pronounced for turbulent than for laminar flow, the effects were discernible for the weakly turbulent conditions of this study. The representative results of Fig. 6 show that bottom plate Nusselt numbers, which are influenced by buoyancy, are as much as 50% larger than top plate Nusselt numbers, which are dominated by forced convection. Moreover, unlike the situation for transitional flow, Nu_t was found to be independent of q_t, as shown by the turbulent flow data of Fig. 4. This result suggests that, although the top plate boundary layer is sufficiently stratified to inhibit disturbances associated with transitional flow, stratification is not sufficient to attenuate mixing associated with turbulent flow. However, it is possible that such attenuation could result from values of q_t larger than those considered in this study.

Heat transfer correlations. The fact that top plate turbulent boundary-layer conditions are uninfluenced by buoyancy suggests the possibility of correlating heat transfer data in terms of forced convection parameters such as z/H, Re_H and Pr. Although existing correlations for turbulent forced convection between parallel plates could not be found in the literature, a correlation is available for turbulent pipe flow [17, 18] and is of the form

$$Nu_D = C_1(D/z)^{0.055} Re_D^{4/5} Pr^{1/3} \tag{3}$$

where $C_1 = 0.034$. The correlation applies for $0.7 < Pr < 100$, for $Re_D > 10{,}000$, and to the entrance region, for which $z/D < 15$ [19, 20]. However, with $C_1 = 0.034$ and $z/D = 15$, equation (3) overpredicts the Colburn equation for fully-developed turbulent flow by approx. 27%. Agreement with the fully-developed result may be obtained by using $C_1 = 0.027$.

Equation (3) may be adapted to a parallel-plate geometry by replacing the tube diameter with the hydraulic diameter, $D_h = 2H$. It follows that

$$Nu_H = C_2(H/z)^{0.055} Re_H^{4/5} Pr^{1/3} \tag{4}$$

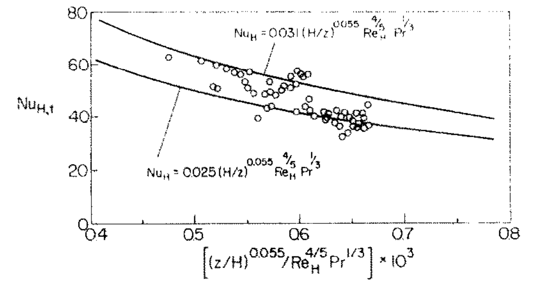

FIG. 7. Comparison of forced convection correlation for turbulent flow with top plate Nusselt number data for $H = 20$, 60 mm, $5200 \leqslant Re_H \leqslant 6500$, $1000 \leqslant q_t \leqslant 4000$ W m^{-2} and $0.25 \leqslant q_b/q_t \leqslant 1.0$.

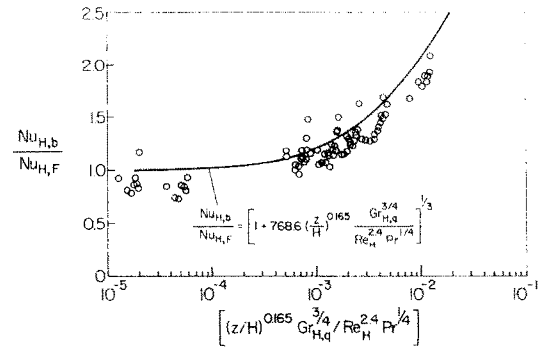

FIG. 8. Comparison of mixed convection correlation for turbulent flow with bottom plate Nusselt number data for $H = 20$, 60 mm, $1400 \leqslant Re_H \leqslant 6500$, $8.6 \times 10^5 \leqslant Gr_{H,q} \leqslant 2.8 \times 10^8$ and $1 \leqslant q_t/q_b \leqslant 4$.

which applies for $z/H < 30$ and $Re_H > 5000$. Values of C_2 equal to 0.031 and 0.025 correspond to C_1 equal to 0.034 and 0.027, respectively. In Fig. 7 equation (4) is compared with all of the top plate heat transfer data obtained for $Re_H > 5000$, and the agreement is reasonable. All of the data are correlated to within 35% with either constant, although most of the data are overpredicted and underpredicted, respectively, for values of C_2 equal to 0.031 and 0.025.

To correlate heat transfer data for the bottom plate, consideration must be given to free and forced convection effects. When a function varies smoothly between two limiting cases which are well defined, an approximate composite relation can be obtained by appropriately summing the limiting expressions. For mixed convection Acrivos [21] and Churchill [22] suggest a superposition of the form

$$Nu^n = Nu_F^n + Nu_N^n \tag{4}$$

where the subscripts refer to pure forced (F) and natural (N) convection. The power n is chosen to provide the best correlation of data, and a value of $n = 3$ was found to be best suited for the conditions of this study. The forced convection component may be obtained from equation (3), and the natural convection component may be obtained from a correlation developed by Fujii and Imura [22] for turbulent free convection from a heated horizontal plate (facing upward).

$$Nu_{H,N} = 0.229(Gr_{H,q}Pr)^{1/4}. \tag{5}$$

Substituting equation (3), with $C_2 = 0.025$, and (5) into equation (4), and using $n = 3$, the correlation for turbulent mixed convection from the bottom plate becomes

$$Nu_H = [1.56 \times 10^{-5}(H/z)^{0.165} Re_H^{2.4} Pr + 0.012(Gr_{H,q}Pr)^{3/4}]^{1/3}. \tag{6}$$

Dividing by equation (3), the ratio of the mixed convection to the forced convection Nusselt number may be expressed as

$$\frac{Nu_H}{Nu_{H,F}} = \left[1 + 769\left(\frac{z}{H}\right)^{0.165} \frac{Gr_{H,q}^{3/4}}{Re_H^{2.4} Pr^{1/4}}\right]^{1/3}. \tag{7}$$

The ratio provides a measure of the extent to which heat transfer is enhanced by buoyancy effects.

Equation (7) is compared with bottom plate heat transfer data obtained for transitional, as well as turbulent, flow conditions ($Re_H > 1400$) in Fig. 8. Inclusion of the transitional flow results is motivated by the fact that, at the bottom plate, buoyancy acts to hasten the onset of turbulence and to enhance turbulent mixing. The agreement is reasonable, with all of the data correlated to within 30% and 91% of the data correlated to within 20%. The agreement is less satisfactory if $C_2 = 0.031$ is used in equation (3), as only 54% of the data are correlated to within 20% and all of the data are correlated to within 47%.

If heat transfer enhancement is presumed to be negligible when $Nu_H/Nu_{H,F}$ is less than 1.1, equation (7) suggests that buoyancy effects may be neglected when the parameter $[(z/H)^{0.165} Gr_{H,q}^{3/4}/Re_H^{2.4} Pr^{1/4}]$ is less than 4.3×10^{-4}. This value is in reasonable agreement with the estimate of 0.001 based on the temperature fluctuations, which corresponds to a heat transfer enhancement of $Nu_H/Nu_{H,F} = 1.21$.

4. SUMMARY

Experiments have been performed to determine the effects of free convection on transitional and turbulent, forced convection flow between horizontal parallel plates which are symmetrically and asymmetrically heated. Major conclusions are as follows.

(1) For turbulent flow ($Re_H > 5000$), the top plate is characterized by a forced convection boundary layer. Nusselt numbers are independent of the heat flux imposed at the top plate and are correlated by equation (4).

(2) For transitional flow ($1400 < Re_H < 5000$), the top plate Nusselt number decreases with increasing heat flux at the top plate. The effect is attributed to boundary-layer laminarization due to a stably stratified temperature distribution.

(3) For both transitional and turbulent flow, the

bottom plate Nusselt number increases with increasing heat flux at the bottom plate. Heat transfer enhancement is due to the effects of buoyancy and is correlated by equation (7).

(4) The effects of buoyancy on convection heat transfer at the bottom plate are small (less than 10%) if the parameter $[(z/H)^{0.165} Gr_{H,q}^{3/4}/Re_H^{2.4} Pr^{1/4}]$ is less than 4.3×10^{-4}.

(5) For the range of conditions in this study, convection heat transfer at one plate is uninfluenced by the magnitude of the heat flux at the opposite plate.

Acknowledgement—We are grateful for support of this work by the National Science Foundation under Grant No. MEA-8316580.

REFERENCES

1. D. G. Osborne and F. P. Incropera, Laminar, mixed convection heat transfer for flow between horizontal parallel plates with asymmetric heating, *Int. J. Heat Mass Transfer* **28**, 207–217 (1985).
2. B. S. Petukhov, Turbulent flow and heat transfer in pipes under considerable effect of thermogravitational forces, in *Heat Transfer and Turbulent Buoyant Convection*, D. B. Spalding and N. Afgan (Editors), pp. 701–717. Hemisphere, Washington (1976).
3. B. S. Petukhov, V. S. Protopopov, and V. A. Silin, An experimental investigation of the regime of impaired heat transfer during the turbulent flow of carbon dioxide at supercritical pressures, *Int. chem. Engng* **13**, 235–240 (1973).
4. D. R. B. Kenning, R. A. W. Shock, and J. Y. M. Poon, Local reductions in heat transfer due to buoyancy effects in upward turbulent flow, *Proc. 5th Int. Heat Transfer Conference*, Vol. 3, pp. 139–143 (1974).
5. J. Khosla, T. W. Hoffman, and K. G. Pollock, Combined forced and natural convection heat transfer to air in a vertical tube, *Proc. 5th Int. Heat Transfer Conference*, Vol. 3, pp. 144–148 (1974).
6. C. R. Kakarala and L. C. Thomas, Turbulent combined forced and free convection heat transfer in vertical tube flow of supercritical fluids, *Int. J. Heat Fluid Flow* **2**, 115–120 (1980).
7. B. S. Petukhov, A. F. Poliakov, Yu. C. Shekhter, and V. A. Kuleshov, Experimental study of the effect of thermogravitation upon turbulent flow and heat transfer in horizontal pipes, in *Heat Transfer and Turbulent Buoyant Convection*, D. B. Spalding and N. Afgan (Editors), pp. 719–727. Hemisphere, Washington (1976).
8. D. Skiadaressis and D. B. Spalding, Prediction of combined free and forced convection in turbulent flow through horizontal pipes, *Lett. Heat Mass Transfer* **4**, 35–42 (1977).
9. H. R. Nagendra, Interaction of free and forced convection in horizontal tubes in the transition regime, *J. Fluid Mech.* **57**, 269–283 (1973).
10. W. M. Kays and M. E. Crawford, *Convective Heat and Mass Transfer*, 2nd edn. McGraw-Hill, New York (1976).
11. R. W. Fox and A. T. McDonald, *Introduction to Fluid Mechanics*, 2nd edn, John Wiley, New York (1978).
12. S. J. Kline and F. A. McClintock, Describing uncertainties in single-sample experiments, *Trans. Am. Soc. mech. Engrs, Mech. Engng* **75**, 3–8 (1953).
13. R. K. Shah, Thermal entry length solutions for the circular tube and parallel plates, *Proc. 3rd National Heat Mass Transfer Conference*, Indian Inst. Tech., Bombay, Vol. I, Paper No. HMT-11-75 (1975).
14. M. Akiyama, G. J. Hwang and K. C. Cheng, Experiments on the onset of longitudinal vortices in laminar forced convection between horizontal plates, *Trans. Am. Soc. mech. Engrs, J. Heat Transfer* **93**, 335–341 (1971).
15. Y. Kamotani, S. Ostrach and H. Miao, Convective heat transfer augmentation in thermal entrance regions by means of thermal instability, *Trans. Am. Soc. mech. Engrs, J. Heat Transfer* **101**, 222–226 (1979).
16. R. R. Gilpin, H. Imura and K. C. Cheng, Experiments on the onset of longitudinal vortices in horizontal Blasius flow heated from below, *Trans. Am. Soc. mech. Engrs, J. Heat Transfer* **100**, 71–76 (1978).
17. W. Nusselt, Der Warmeaustausch zwischen Wand und Wasser im Rohr, *Forsch. Geb. Ingenieurwes.* **12**, 309–315 (1931).
18. M. N. Özişik, *Basic Heat Transfer*, pp. 264–266. McGraw-Hill, New York (1977).
19. J. P. Hartnett, Experimental determination of the thermal entrance length for the flow of water and oil in circular pipes, *Trans. Am. Soc. mech. Engrs* **77**, 1211–1220 (1955).
20. R. G. Deissler, Turbulent heat transfer and friction in the entrance region of smooth passages, *Trans. Am. Soc. mech. Engrs* **77**, 1221–1233 (1955).
21. A. Acrivos, On the combined effect of forced and free convection heat transfer in laminar boundary layer flows, *Chem. Engng Sci.* **21**, 343–349 (1966).
22. S. W. Churchill, A comprehensive correlating equation for laminar, assisting, forced and free convection, *A.I.Ch.E. Jl* **23**, 10–16 (1977).
23. T. Fujii and H. Imura, Natural convection heat transfer from a plate with arbitrary inclination, *Int. J. Heat Mass Transfer* **15**, 755–767 (1972).

ETUDE EXPERIMENTALE DE LA CONVECTION THERMIQUE MIXTE POUR UN ECOULEMENT TRANSITIONNEL ET TURBULENT ENTRE DES PLANS PARALLELES ET HORIZONTAUX

Résumé—Des expériences sont conduites pour déterminer l'effet de la gravitation sur le transfert thermique de convection pour un écoulement transitionnel et turbulent d'eau entre des plaques parallèles et horizontales. Dans l'écoulement transitionnel, les nombres de Nusselt sur les plaques supérieure et inférieure décroissent et croissent respectivement quand le flux thermique augmente. Des résultats pour la plaque supérieure sont attribués à la laminarisation de la couche limite, tandis que ceux pour le plaque inférieure sont attribués à la convection libre. Pour l'écoulement turbulent, le transfert thermique sur la plaque supérieure est représenté par une expression de convection forcée pure, tandis que les résultats sur la plaque inférieure sont représentés par une expression de convection mixte.

EXPERIMENTELLE UNTERSUCHUNG DES WÄRMEÜBERGANGS DURCH MISCHKONVEKTION BEI TURBULENTER STRÖMUNG UND STRÖMUNG IM ÜBERGANGSBEREICH ZWISCHEN HORIZONTALEN PARALLELEN PLATTEN

Zusammenfassung—Es wurden Experimente durchgeführt, um den Einfluß von Auftriebskräften auf den konvektiven Wärmeübergang bei turbulenter Strömung und Strömung im Übergangsbereich von Wasser zwischen horizontalen parallelen Platten zu bestimmen. Bei der Strömung im Übergangsbereich nimmt die Nusselt-Zahl bei wachsender Wärmestromdichte an der oberen Platte ab, an der unteren Platte zu. Die Ergebnisse für die obere Platte werden auf eine Laminarisierung der Grenzschicht, die für die untere Platte auf eine Verbesserung des Wärmeübergangs durch freie Konvektion zurückgeführt. Bei der turbulenten Strömung werden die Wärmeübergangswerte für die obere Platte durch einen Ausdruck für die reine erzwungene Konvektion korreliert, während die Werte für die untere Platte durch einen Ausdruck für die Mischkonvektion korreliert werden.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ТЕПЛООБМЕНА В ПЕРЕХОДНОМ И ТУРБУЛЕНТНОМ РЕЖИМАХ ТЕЧЕНИЙ МЕЖДУ ГОРИЗОНТАЛЬНЫМИ ПАРАЛЛЕЛЬНЫМИ ПЛАСТИНАМИ ПРИ СМЕШАННОЙ КОНВЕКЦИИ

Аннотация—Проведены эксперименты по определению влияния архимедовых сил на конвективный теплоперенос в переходном и турбулентном режимах течения воды между горизонтальными параллельными пластинами. В переходном режиме числа Нуссельта у верхней и нижней пластин, соответственно, уменьшаются и увеличиваются с ростом теплового потока. Результаты для верхней пластины соответствуют ламинаризации пограничного слоя, в то время как результаты для нижней пластины-увеличению теплообмена за счет свободной конвекции. При турбулентном режиме данные по теплопереносу у верхней пластины определяются выражением для чисто вынужденной конвекции, а для нижней-соотношением для смешанной конвекции.