INFLUENCE OF BUOYANCY ON MEAN FLOW AND TURBULENCE UNDER CONDITIONS OF MIXED CONVECTION IN AN ANNULUS

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ABSTRACT

LDA measurements are reported of mean flow and turbulence in water as it flows downwards through a long vertical passage of annular cross-section having an inner surface which can be uniformly heated and an outer one which is adiabatic. Under buoyancy-opposed conditions, which were achieved by heating the core and operating at reduced mass flow rate, the flow near the inner surface was retarded, turbulent velocity fluctuations and turbulent shear stress were increased and the effectiveness of heat transfer was enhanced. When the influence of buoyancy was very strong, flow reversal occurred near the inner surface. Under such conditions, turbulence was produced very readily and the heat transfer process remained very effective, even when the Reynolds was reduced to values at which the flow was laminar in the absence of heating. The measurements of turbulence in buoyancy-opposed flow made in this study provide direct confirmation of the validity of the ideas currently used to explain the influences of buoyancy on mixed convection in vertical passages.

NONMENCLATURE

Bo	Buoyancy parameter, $Bo = Gr^*/(Re^{3.425} Pr^{0.8})$
c_p	Specific heat at constant pressure
D_e	Equivalent diameter of the annulus
Gr^*	Grashof number, $Gr^* = \beta g D_e^4 q_w/(k \nu^2)$
g	Acceleration due to gravity
h	Local heat transfer coefficient
k	Thermal conductivity
Nu	Nusselt number, $Nu = hD_e/k$
Pr	Prandtl number, $Pr = \mu c_p/k$
q_w	Heat flux from core surface
r_o	Inner radius of the annulus
Re	Reynolds number, $Re = u_b D_e/\nu$
u_b	Bulk velocity
u	Axial velocity component
v	Radial velocity component
x	Distance from start of heating
β	Coefficient of volumetric expansion
μ	Dynamic viscosity
ν	Kinematic viscosity

INTRODUCTION

Turbulent mixed convection in vertical passages is encountered in a variety of thermal systems and the influence of buoyancy on turbulence and heat transfer in such plant is a matter of considerable practical importance. Consequently, many investigators have studied this problem and much progress has been made in understanding the rather surprising influences of buoyancy on heat transfer which occur. Attention has been concentrated mainly on uniformly heated circular tubes. A comprehensive review of such work can be found in Jackson, Cotton and Axcell (1989). Heat transfer can be either enhanced or impaired as a result of the influence of buoyancy depending on the conditions and the flow direction. However, the trends observed are not always as expected. For instance in the buoyancy-opposed case (the particular mode of mixed convection addressed in this paper) the process of heat transfer is enhanced in spite of the flow near the heated surface being retarded and therefore less effective in terms of its convective capability. Such trends have been reported by numerous investigators, for example, Petukhov and Nolde (1959), Brown and Gauvin (1965), Petukhov and Strigin (1968), Axcell (1975), Jackson and Fewster (1977), Easby (1978), Rouai (1987), Büyükalaca (1990) and Li (1994).

The explanation of the enhancement of heat transfer in buoyancy-opposed flow is that more turbulence is produced due to shear stress being increased in the near wall region (Jackson and Hall (1969)). A simple semi-empirical model based on such ideas (the details of which can be found in Jackson and Hall (1979)) has proved to be successful in describing this enhancement of heat transfer. The following simple equation is yielded by the model

$$\frac{Nu}{Nu_f} = \left[1 + 2.5x10^5 Bo \left(\frac{Nu}{Nu_f} \right)^{-2} \right]^{0.46} \tag{1}$$

Nu_f is the Nusselt number for forced convection (buoyancy-free conditions) under otherwise identical conditions and Bo is a parameter which combines Grashof number Gr^*, Reynolds number Re and Prandtl number Pr in the form

$Gr^{*}/Re^{3.425}Pr^{0.8}$ in a manner which characterises the strength of buoyancy-influences.

Although the explanation of heat transfer enhancement in buoyancy-opposed mixed convection through the action of increased turbulence production and improved turbulent diffusion of heat is generally accepted, very little is available in the way of measurements of mean flow and turbulence quantities under buoyancy-influenced conditions which provide direct confirmation of the ideas involved. The main aim of the present study was to rectify this by making such measurements using laser Doppler anemometry. The particular configuration chosen for the study is downward flow of water in passage of annular cross-section with a uniformly heated inner surface and an adiabatic outer one (it was recognised that the latter condition could be achieved by using an outer casing made of perspex which then enabled laser optical measurements to be readily made within the flow). As indicated earlier, many studies of mixed convection in uniformly heated circular tubes have been reported. However very little work has been done using passages of other geometry (particularly ones involving both heated and unheated surfaces). This was an additional reason for choosing a passage of annular cross-section. It was envisaged that a valuable outcome of the study would be the provision of basic data for the evaluation of computational formulations and turbulence models used for simulating buoyancy-influenced flows.

EXPERIMENTAL INVESTIGATION

The flow loop used in this investigation is shown in Figure 1. Water from the header tank flows to the top of the test section which is a vertical passage of annular cross section, and then downwards through it. On leaving the test section it is returned by a centrifugal pump to the header tank via an orifice plate flowmeter and a shell and tube heat exchanger. To achieve a symmetrical flow in the test section the water is supplied to it through a manifold arrangement. Four tubes 90° apart feed into cylindrical entry section containing flow conditioning grids and straightener tubes. The test section has a diameter ratio 1.94. The outer casing is made from perspex and has an internal diameter of 140mm. The core is made from stainless steel tube of external diameter 76 mm and wall thickness 1.6 mm. It is unheated over a length of 1.5 m and then uniformly heated by resistive means over a length of 3.0 m. About 90 thermocouples are used on the core to measure its temperature distribution. Two sheathed, mineral insulated, thermocouples situated in the flow at inlet and outlet are used to measure the water temperature at those locations. The electrical power to heat the test section is supplied by a thysistor-controlled system. The signals from the instrumentation are all recorded automatically by a data acquisition system, and then processed on line.

The LDA system used consists of a 4W Argon-ion laser generator, a 60×40 transmitter connected by a 10 metre long fibre optic cable to a probe, a two component photomultiplier and two Burst Spectrum Analysers. The four beams are carried to the probe by the fibre optic cable, separated inside it and then focussed to a point in the test section with a lens of focal length of 160mm. The light scattered backwards by particles in the flow is collected by a receiver inside the probe, transmitted to the photomultiplier and converted to electrical signals. These pass to the BSA units, the data from which are finally collected by a further computer-based data acquisition system. A 45° arrangement of the beams was chosen in order to measure the velocity components in the directions ±45° off the direction

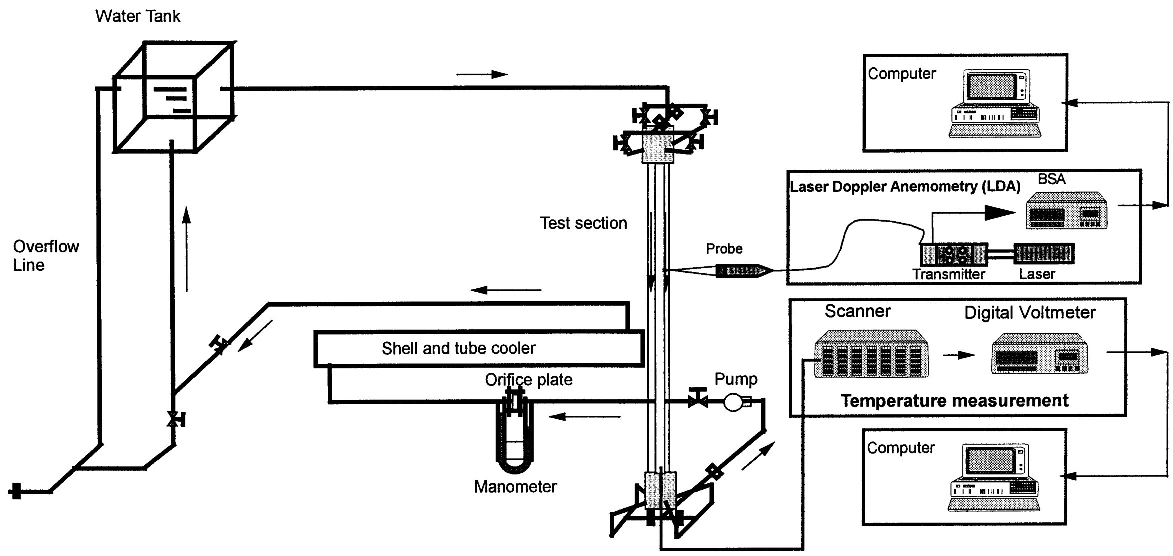

Figure 1 The test facility

of the axis of the annular tube. The vertical and horizontal velocity components were calculated from the two signals.

Using the test facility just described, a comprehensive programme of experiments was carried out which yielded detailed information concerning local heat transfer coefficient and also profiles of mean velocity and turbulence quantities for conditions where the mode of heat transfer varied from buoyancy-free forced convection to buoyancy-dominated mixed convection. Measurements of mean flow and turbulence were also made under isothermal conditions (i.e. without heating). A range of Reynolds number, from 20000 down to 1500 was covered. Grashof number, Gr*, was varied from 2×10^{8} to 1.5×10^{9}. Some sample results obtained in the course of the study are presented here (see Figures 2 to 5). Attention is concentrated on measurements of mean flow and turbulence quantities. In each case, profiles of mean velocity, turbulent velocity fluctuation and shear stress are presented. The heat transfer results have been reported separately (see Wu, He, Kuester and Jackson (2000)).

RESULTS AND DISCUSSION

Turbulent flow with negligible buoyancy influences

Figure 2 shows the results for Re = 20000, the highest value of Reynolds number covered. Figure 2(a) shows the results for conditions of isothermal flow (no heating). The profiles exhibit the characteristics expected for fully developed turbulent flow in an annulus. Figure 2(b) shows the corresponding results for non-isothermal conditions (ie, with heating applied to give a Grashof number Gr* of 1.49×10^{9}). The buoyancy parameter Bo is only 0.59×10^{-6}. It can be seen that the profiles of velocity, turbulent velocity fluctuation and the turbulent shear stress are essentially the same as the corresponding ones in Figures 2(a). The flow is not significantly influenced by buoyancy and the condition is one of forced convection heat transfer.

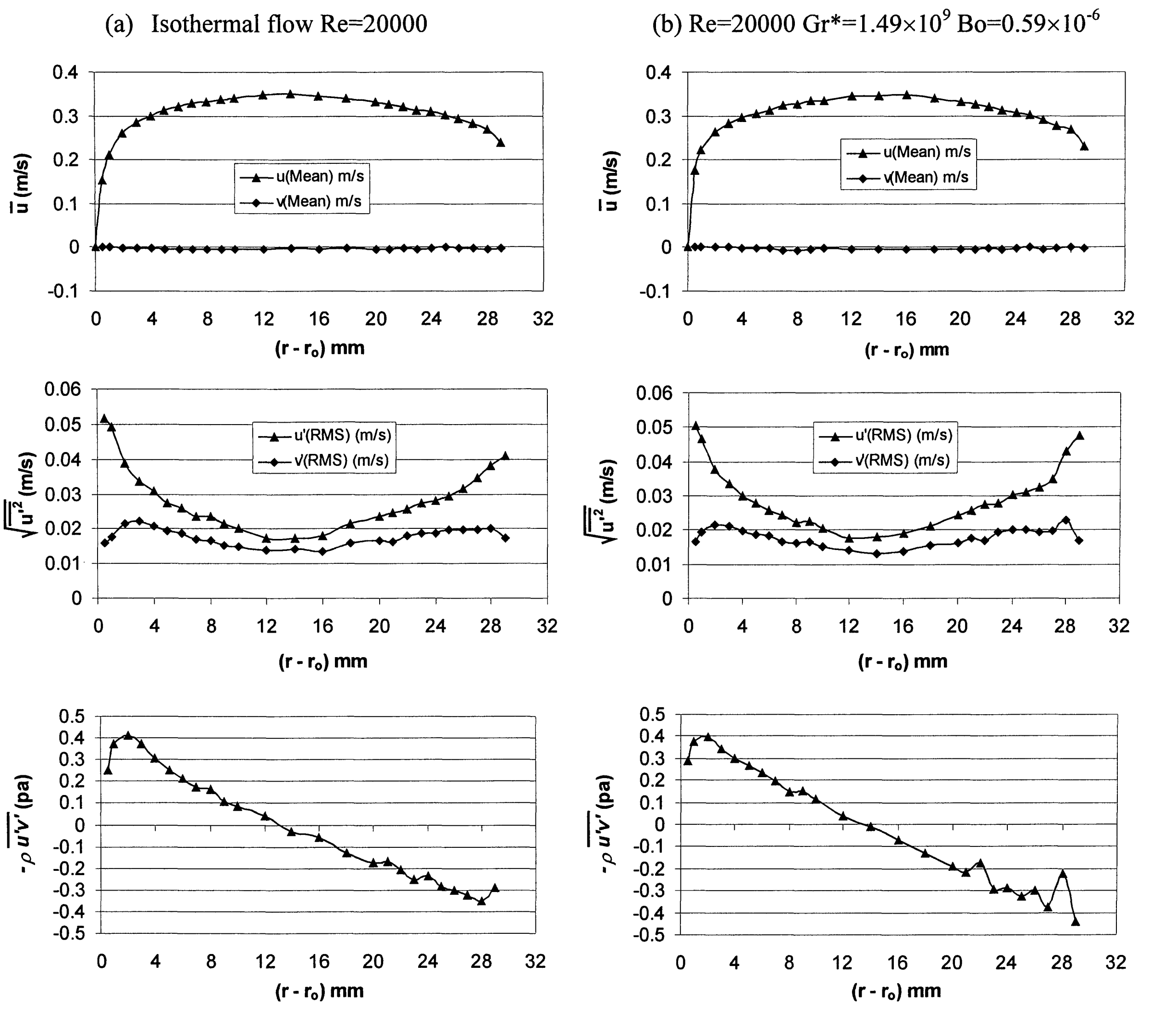

Figure 2 Local mean velocity and turbulent quantities

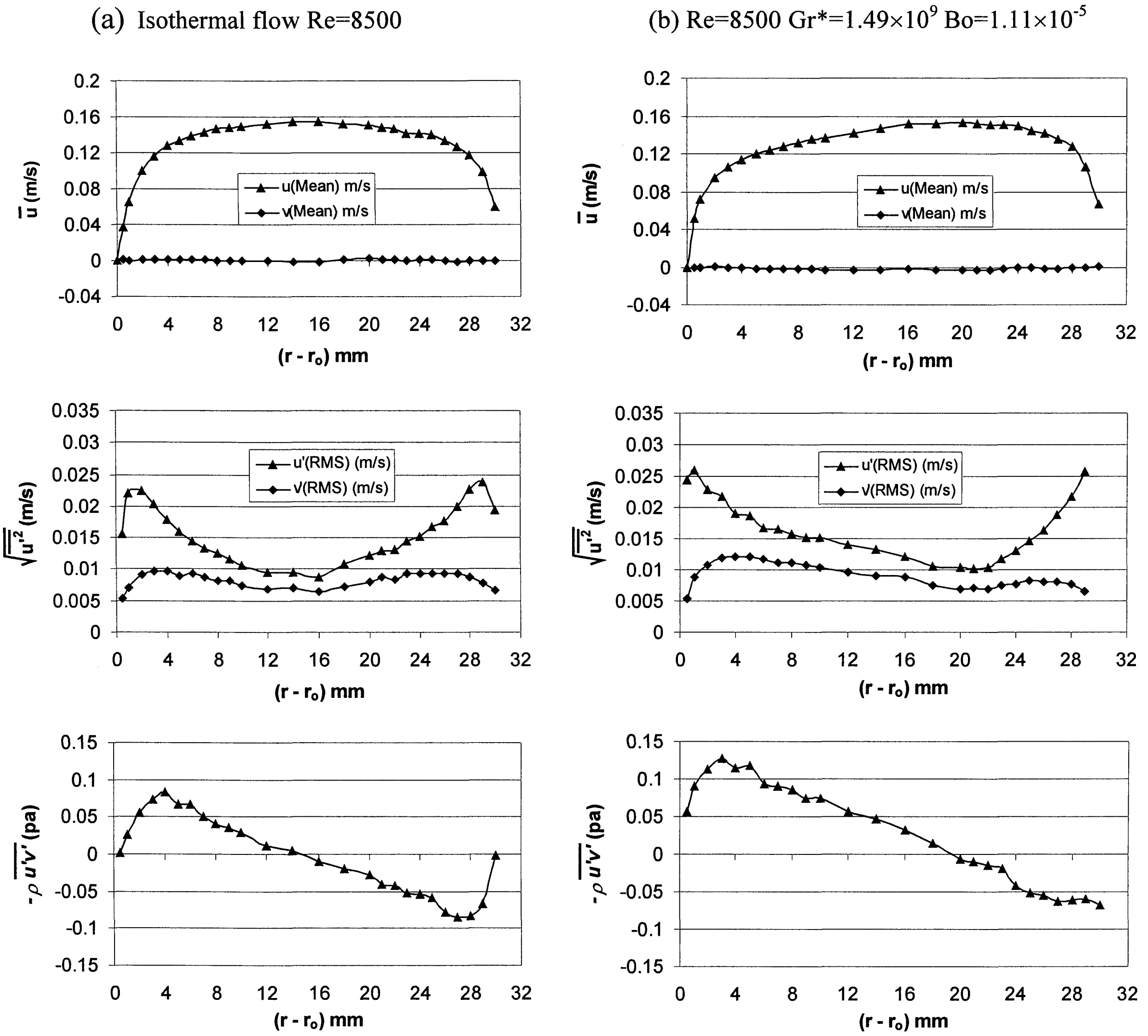

Figure 3 Local mean velocity and turbulent quantities

Turbulent flow with buoyancy influences

Figure 3 shows the results for Re = 8500. From Figure 3(a) it can be seen that the profiles of mean velocity and turbulence quantities for isothermal conditions again exhibit all the characteristics of fully developed turbulent flow. Figure 3(b) shows the results obtained under non-isothermal conditions (i.e. with heating). The Grashof number Gr* is again 1.49×10^{9}, but because the Reynolds number is smaller the buoyancy parameter is increased to 1.11×10^{-5}. As can be seen, the velocity profile is not the same as that for the isothermal case. The flow near the heated surface is retarded due to the buoyancy opposing it. As a result, the peak velocity is increased slightly and it is shifted towards the outer surface. The profiles of turbulent velocity fluctuation are also modified. The peak values on the heated side are increased and they occur nearer to the surface. The locations of the minima on the profiles are shifted towards the outer surface as in the case of the peak value of mean velocity. The turbulent shear stress profile exhibits similar trends. The maximum shear stress is increased in comparison with the isothermal case and the location of zero stress is shifted toward the outer surface. Thus we see that there is a significant influence of buoyancy on the mean and turbulence flow in this case.

Figure 4 shows the results for Re=4000. As can be seen from Figure 4(a) the isothermal flow results still exhibit the characteristics expected for fully developed turbulent flow. Figure 4(b) shows the results obtained with heating applied. The Grashof number is again $1.49\mathrm{x}10^{9}$ but now the buoyancy parameter is increased to 1.47×10^{-4}. The velocity profile is very clearly distorted, the flow on the heated side being strongly retarded and actually stalling near the surface. The peak velocity is significantly greater than for isothermal flow, and its location is shifted far away from the heated surface. The peak values of turbulent velocity

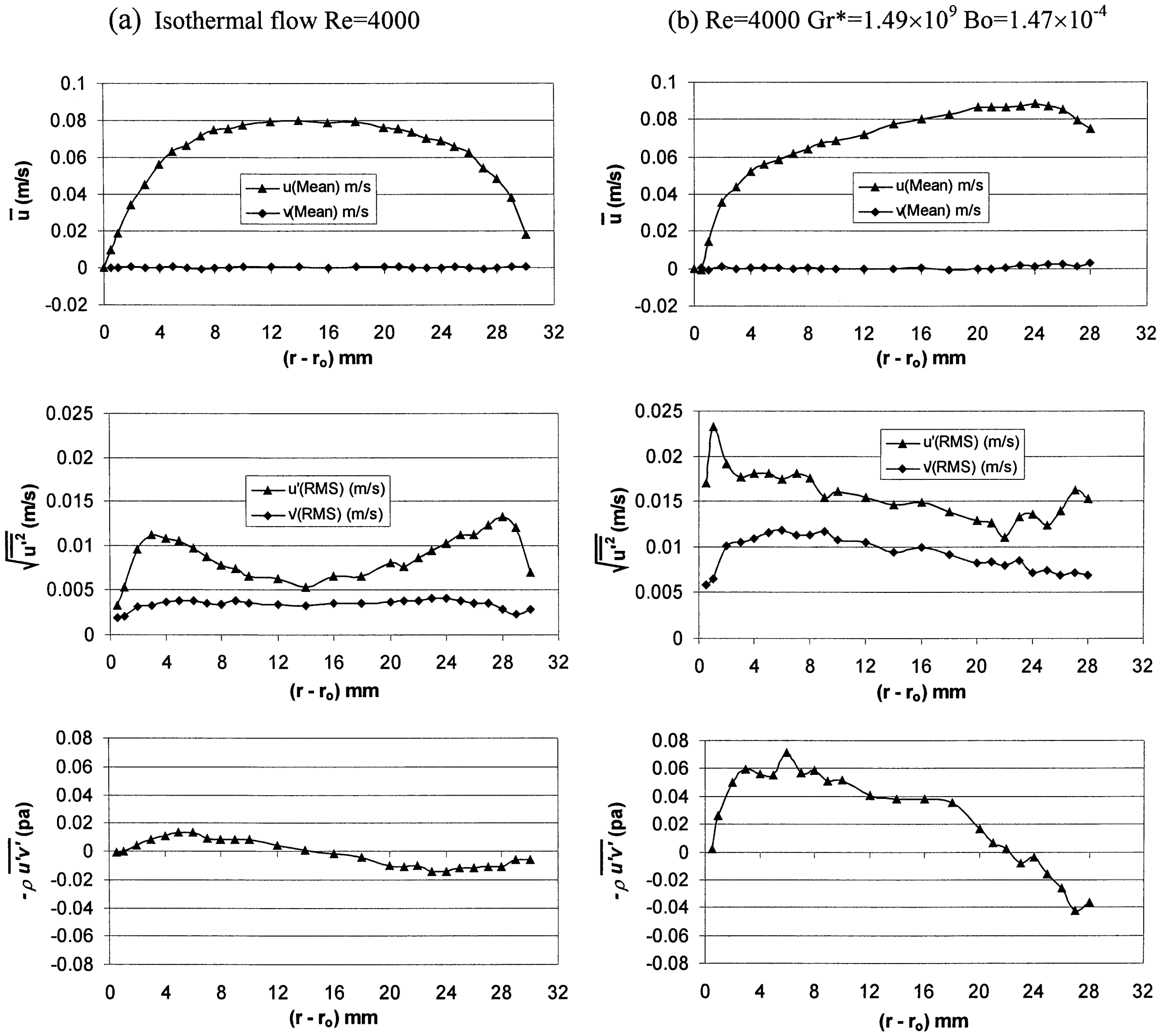

Figure 4 Local mean velocity and turbulent quantities

fluctuation on the heated side are about twice those for the unheated case. The maximum value of turbulent shear stress is increased by a factor of about four. The flow is very strongly influenced by buoyancy and is much more turbulent than for isothermal flow.

Buoyancy-induced transition

Figure 5 shows results for a Reynolds number of 2000. As can be seen from Figure 5(a), the flow is laminar without heating. The velocity profile is parabolic in shape and is in good agreement with that predicted by theory for fully developed laminar flow. However, in the case of the results obtained with heating shown in Figure 5(b) the situation is very different. Here the Grashof number Gr* is again 1.49×10^9, but the buoyancy parameter Bo is now increased to $1.58\text{x}10^{-3}$. Instead of being laminar, the flow is very turbulent. It can be seen that the mean flow is actually reversed near the heated surface. The turbulent velocity fluctuations are greater than those for isothermal flow at a Reynolds number of 8500 but in this case the profiles have only one peak instead of two. The turbulent shear stress is very high. Its maximum value is greater than that for isothermal flow at a Reynolds number of 8500.

CONCLUDING REMARKS

The measurements of turbulence in buoyancy-opposed flow made in the present study provide direct confirmation of the validity of the ideas which are currently used to explain the influences of buoyancy on mixed convection in vertical passages.

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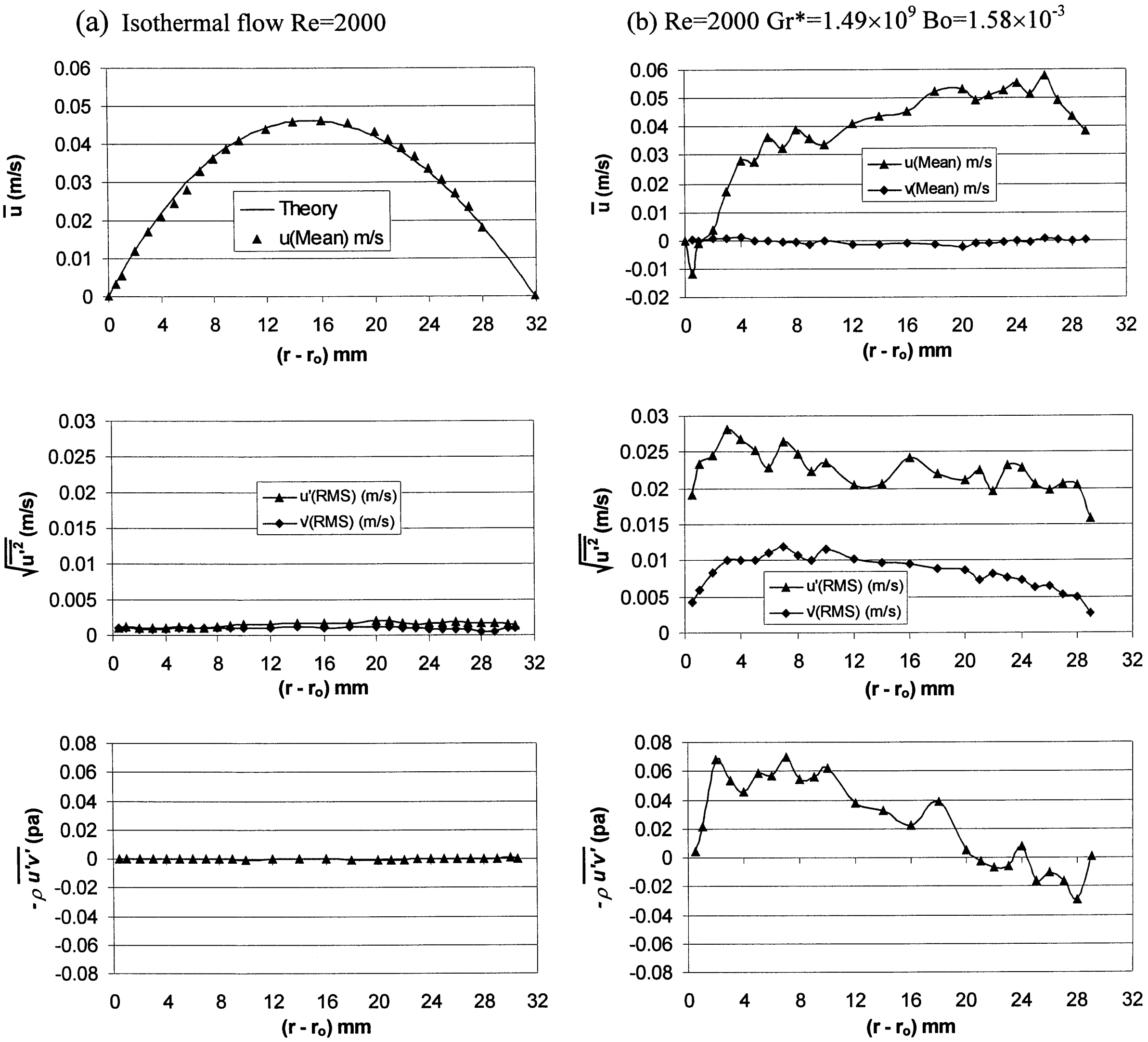

Figure 5 Local mean velocity and turbulent quantities

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ACKNOWLEDGEMENT

The work reported in this paper was carried out under the terms of an IMC research contract funded by British Energy Generation Ltd. entitled 'CFD Quality and Trust – Generic Studies of Thermal Convection'. The authors gratefully acknowledge the financial assistance provided.