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Influences of buoyancy and thermal boundary conditions on heat transfer with naturally-induced flow

J.D. Jackson, J. Li

University of Manchester, United Kingdom

Abstract. A fundamental study is reported of heat transfer from a vertical heated tube to air which is induced naturally upwards through it by the action of buoyancy. Measurements of local heat transfer coefficient were made using a specially designed computer-controlled power supply and measurement system for conditions of uniform wall temperature and uniform wall heat flux. The effectiveness of heat transfer proved to be much lower than for conditions of forced convection. It was found that the results could be correlated satisfactorily when presented in terms of dimensionless parameters similar to those used for free convection heat transfer from vertical surfaces provided that the heat transfer coefficients were evaluated using local fluid bulk temperature calculated utilising the measured values of flow rate induced through the system. Additional experiments were performed with pumped flow. These covered the entire mixed convection region. It was found that the data for naturally-induced flow mapped onto the pumped flow data when presented in terms of Nusselt number ratio (mixed to forced) and buoyancy parameter. Computational simulations of the experiments were performed using an advanced computer code which incorporated a buoyancy-influenced, variable property, developing wall shear flow formulation and a low Reynolds number k~ε turbulence model. These reproduced observed behaviour quite well.

1. INTRODUCTION

Convective heat transfer from the outside surface of the steel containment vessel of a pressurised water reactor to a flow of air induced upwards by buoyancy through the space between the vessel and an external concrete shell has been proposed as a passive method of removing heat from the containment following a severe accident. Whilst there is no doubt that conditions of turbulent flow could be produced by this means, it is probable that the effectiveness of the heat transfer process would be poor. In view of the limited flow rate likely to be achieved, the heat transfer process within the passage will be buoyancy-influenced as well as buoyancy-driven. The mechanism of heat transfer will therefore be mixed free and forced convection. Published work on mixed convection heat transfer in vertical passages has been reviewed in a number of papers (see, for instance, Reference [1]). Some surprising trends have been identified. In the case of upward flow in a heated vertical passage, buoyancy aids the motion but contrary to expectation the values of heat transfer coefficient achieved can be lower than for conditions of turbulent forced convection. This is because the turbulence in the boundary layer is modified through the action of buoyancy with the result that the flow takes on the characteristics of one at lower Reynolds number. Impairment of heat transfer builds up gradually as the buoyancy influence is caused to become stronger, either by increasing the heat loading or reducing the flow rate. Eventually a very sharp reduction in the effectiveness of heat transfer is found to occur. With further increase of buoyancy influence, the process of heat transfer recovers. In contrast, for downward flow in a heated tube buoyancy opposes the motion, turbulence is increased and heat transfer is enhanced. The physical mechanism by which turbulent heat transfer is modified through the action of buoyancy influences in vertical passages was first identified by Hall and Jackson [2]. The contrasting influences on heat transfer for upward and downward flow are well described by a simple semi-empirical model of mixed convection developed by Jackson and Hall [3]. Figure 1 shows the predictions of that model for the case of a uniformly heated tube. These are found to be in good agreement with observed behaviour (see Reference [1]).

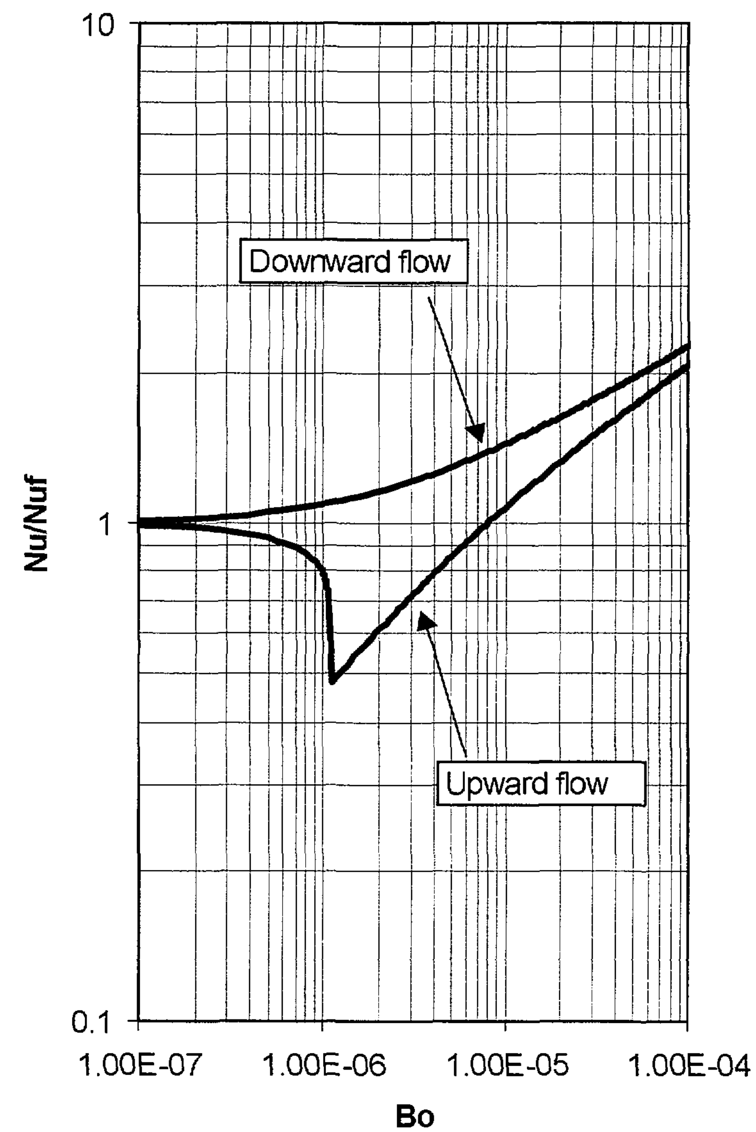

FIG. 1. Description of mixed convection using the semi-empirical model of Jackson and Hall

It is of interest to consider whether impairment of heat transfer would be encountered under the conditions likely to be achieved in a buoyancy-driven flow system of the kind which has been proposed for passively cooling a nuclear reactor containment vessel. In this connection, a further matter needs to be considered. Most of the experimental studies of mixed convection reported to date have been carried out with a thermal boundary condition of uniform wall heat flux. However, in the case of a severe accident in a pressurised water reactor, where steam is released from the core into a steel containment vessel and is condensing on its inside surface, the vessel will take up a uniform temperature. Since the nature of the thermal boundary condition could certainly affect the process of heat transfer to the air, there is a need to consider whether the behaviour with uniform wall temperature will be similar to that with uniform wall heat flux.

The study reported here using a non-uniformly heated test section which can operate at uniform temperature was undertaken to clarify these matters. This naturally-induced cooling experiment (NICE) formed part of the DABASCO project funded by the European Commission to provide an experimental data base for containment thermal hydraulic analysis (see Reference [4]).

2. EXPERIMENTAL FACILITY

The experimental facility used in the present study is shown in Figure 2. The test section, which was made from stainless steel tube of inside diameter 72.9 mm and wall thickness 1.63 mm, is suspended vertically in a space of height 15 m. It can be used for experiments with either naturally-induced flow or pumped flow.

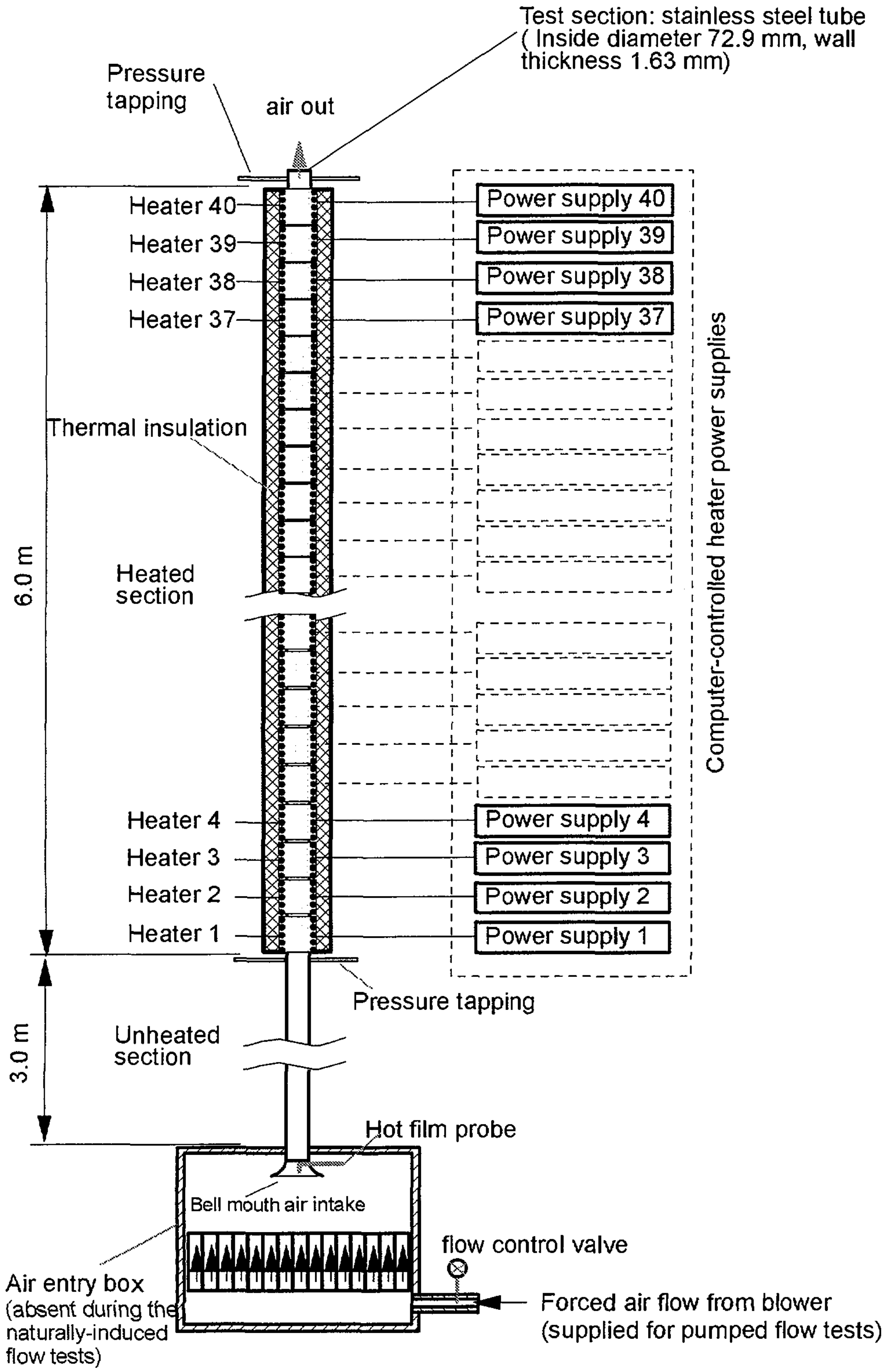

FIG. 2. General arrangement of the test facility

In the case of naturally induced flow, the entry box is absent and air passes from the laboratory into the bellmouth intake and upwards through an unheated flow development section of length 3 m and a heated section above it of length 6 m before discharging at the top. Motion occurs simply as a result of heat being applied to the test section. The velocity at inlet is measured using a calibrated hot film probe mounted in the centre of the bellmouth intake.

In the case of pumped flow, air is supplied to the entry box at the bottom of the test section by a blower and then flows upwards through the test section. The mass flow rate can be measured using a metering nozzle installed at the exit. The hot film probe was calibrated against this flow metering nozzle. Wall static pressure tappings are provided at the top and bottom of the heated section. The pressure difference is measured using a high precision electronic micro-manometer.

Numerous thermocouples attached to the outside of the heated length of the test section enable the wall temperature distribution to be measured in detail. The heated length is well lagged on the outside with pre-formed thermal insulation of low thermal conductivity to minimise heat transfer to the surroundings. The small loses which do occur can be accurately accounted for using data from calibration experiments which were performed at the commissioning stage. Heat can be applied to the test section either uniformly or non-uniformly by means of 40 separate, individually-controlled heaters distributed along its length. These were made using proprietary heater cable which was wound tightly around the outside of the stainless steel tube. Electrical power is supplied to the heaters from the mains via variable auto-transformers through the specially designed computer-based power control and measurement system shown in Figure 3. This supplies power to each heater at a rate needed to maintain the tube at a specified temperature at that location and also enables the power to be measured. The power is controlled by allowing a proportion of the half cycles of the incoming AC supply to pass to the heater. For each of the 40 heaters there is a control signal generator and a zero-crossing solid state relay. The former generates a control signal which enables the latter to pass a programmable number of half cycles from the supply to the heater during a specified time interval of 0.16 s. The signal generators are connected to a computer via a PC interface. The computer can write a number to each of the 40 signal generators under the action of software. The power to each heater is controlled by these numbers. Knowing this number, the voltage of the incoming supply and the electrical resistance of the heater, the power generated can be calculated. In operation, the computer reads temperatures using the Intercole data acquisition system. It then compares each wall temperature with a pre-set value, calculates a new number using the PID technique and sends it to the corresponding signal generator. The power supply system can also be operated in such a manner that a specified distribution of heat input is applied to the test section. A Pentium PC connected to a 208 channel data acquisition system is used for signal collection and processing. The software package which is used to drive the data logger and the power control and measurement system was developed and tested 'in house' using Visual Basic under the Windows 95 environment.

Initially, commissioning tests were performed on the test facility using the pumped flow arrangement to demonstrate that everything was operating satisfactorily. Friction factors determined from pressure drop measurements made under conditions of isothermal flow through the test section were compared with values calculated using the well-established correlation equation of Blasius for fully developed turbulent flow in a smooth tube. The maximum discrepancy was less than 6%.

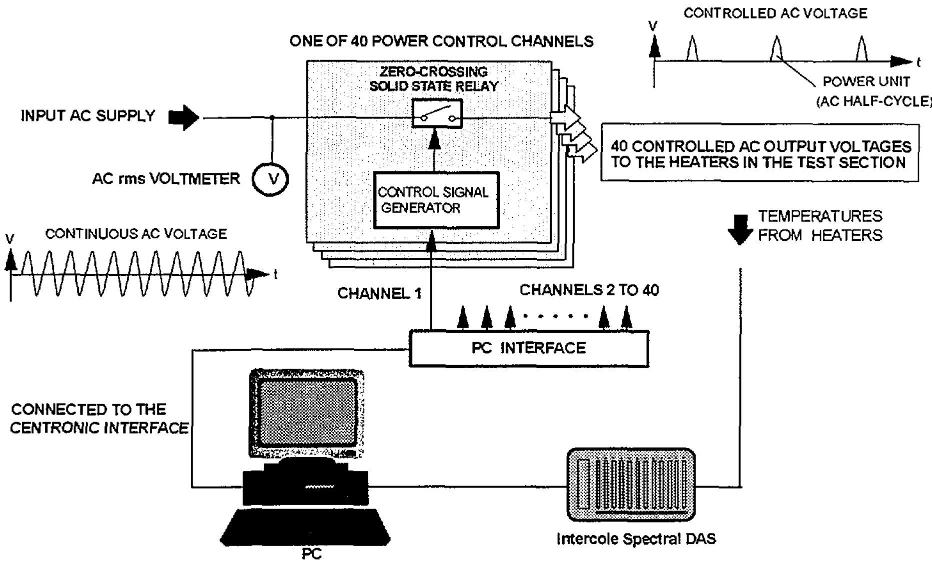

FIG. 3. Computer-based heater power control and measurement system

Local values of Nusselt number were determined at various locations along the tube from experiments performed under conditions of forced convection with uniform heating. In Figure 4(a) the results are compared with the distribution of Nusselt number calculated using the established empirical correlation equation of Petukhov et al [5]. As can be seen, the agreement is very satisfactory.

Results from a mixed convection experiment with uniform wall heat flux are shown in Figure 4(b). The Reynolds number was about 10,000. Under such conditions heat transfer was found to be strongly impaired due to the influence of buoyancy. It can be seen that the local values of Nusselt number lie well below the curve for forced convection calculated using the Petukhov equation. The observed behaviour is very similar to that found in an earlier study by Li [6] using a uniformly heated test section of similar dimensions (see Jackson and Li [7]).

3. EXPERIMENTAL INVESTIGATION

3.1. Experiments with naturally-induced flow

From the results of the commissioning tests with pumped flow presented in preceding section, it was clear that the test facility was operating satisfactorily. Accordingly, a detailed programme of experiments was carried out with naturally-induced flow and uniform wall temperature. The power control and measurement system was used to achieve uniform values of wall temperature ranging from 60 to 300°C. Corresponding experiments with uniform wall heat flux were also performed keeping the total heat input the same as in the uniform wall temperature experiments. The matrix of conditions covered was as follows.

Wall temperature Tw (°C)	60	80	100	150	200	250	300
Corresponding wall heat flux q_w (W/m^2)	91.0	171.4	229.7	428.0	640.0	899.4	1130.0

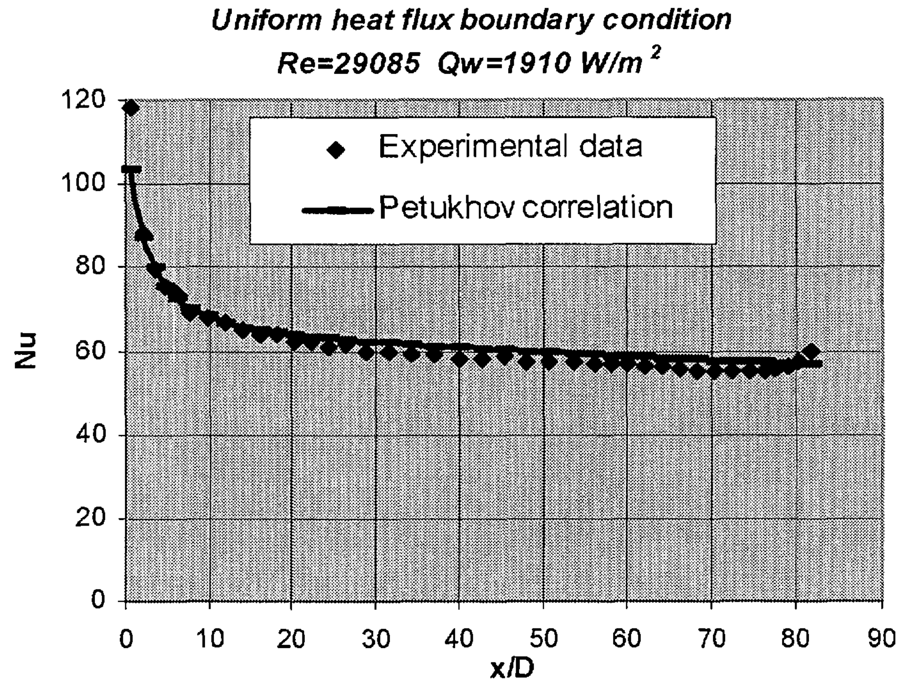

FIG. 4(a). Commissioning test results with uniform heating for forced convection

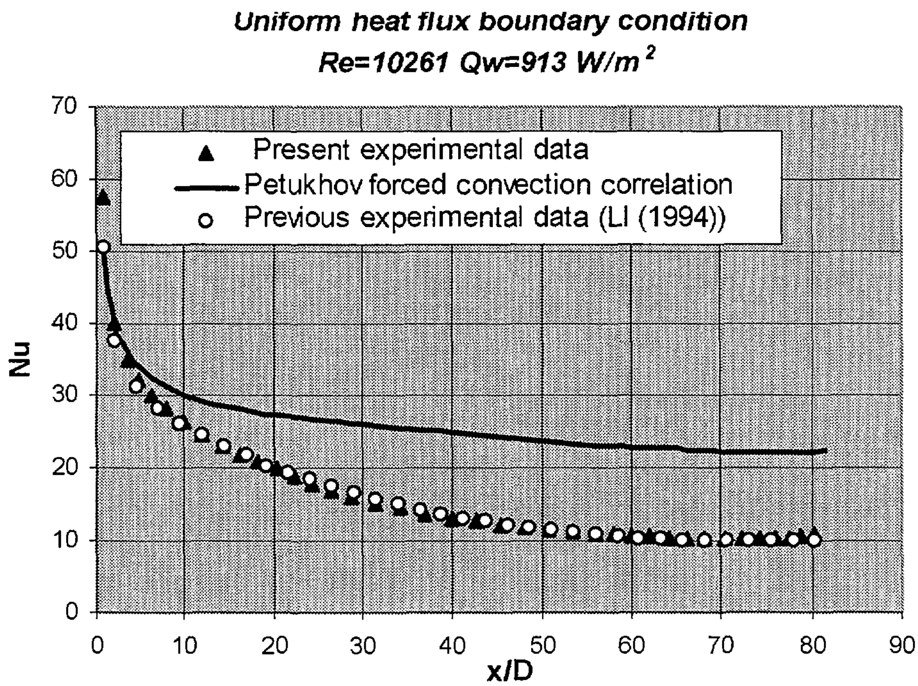

FIG. 4(b). Commissioning test results with uniform heating for mixed convection with impaired heat transfer

In these experiments the flow rate induced through the system increased systematically as the thermal loading was increased. The variation of dimensionless flow rate (characterised by Reynolds number) with dimensionless heat loading (characterised by Grashof number) is shown in Figures 5(a) and 5(b) for the uniform wall heat flux and uniform wall temperature cases, respectively. The physical properties in these dimensionless parameters were evaluated at the inlet temperature and the characteristic dimension used was the tube diameter.

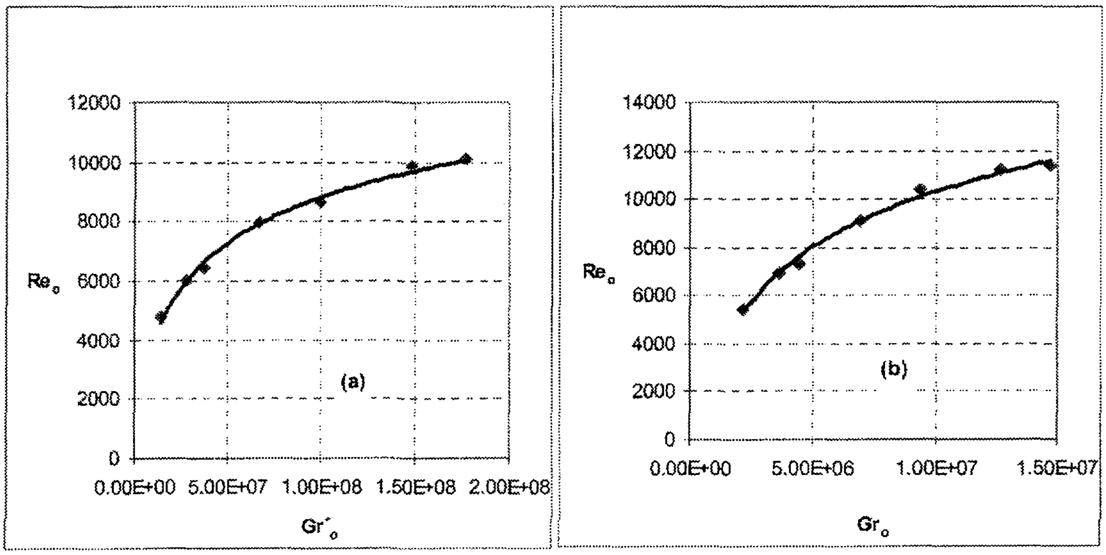

FIG. 5. Relationship between flow rate and thermal loading for naturally induced flow with (a) uniform heat flux, (b) uniform wall temperature

Figures 6(a) to 6(f) show the axial distributions of Nusselt number obtained in these experiments. The flow rates induced through the test section were used to determine the Reynolds numbers shown on the figures. In each case a curve is presented for forced convection with uniform heat flux evaluated at the Reynolds number for that case using the equation of Petukhov et al [5]. The main points to note are:

(i) The experimental values of Nusselt number all follow a similar general pattern. They lie well below the curves for forced convection calculated using the Petukhov equation.

(ii) The values of Nusselt number for the uniform wall temperature case are slightly lower than those for uniform wall heat flux in spite of the fact that the flow rates induced through the system are greater.

(iii) The Nusselt number curves for forced convection calculated using the Petukhov equation decrease slowly at a steady rate in the thermally fully developed region. This is due to the fact that the Reynolds number decreases as the bulk viscosity increases due to the rise of bulk temperature.

Figures 7(a) and 7(b) show the experimental results plotted in terms of local values of local Nusselt number, Nu_x , and Grashof number, Gr_x (evaluated using the distance x from the start of heating as the characteristic dimension). The fluid properties were evaluated at the local bulk temperature, calculated knowing the air temperature at inlet, the heat input and the flow rate of air induced through the system. As can be seen, this method of presenting the results does enable a good correlation of the data to be achieved.

The main conclusions which can be drawn from the naturally-induced flow tests are as follows:

Even though the flow rates achieved were such that in the absence of buoyancy influences the flow would have been turbulent, the effectiveness of heat transfer was seriously impaired in relation to that expected for conditions of turbulent forced convection at those flow rates. It is clear that under the conditions of all the experiments performed the turbulence structure was significantly modified by buoyancy. In the experiments with uniform wall temperature the heat transfer behaviour did not change much as the temperature was increased. Furthermore, the behaviour was very similar in the corresponding experiments with uniform wall heat flux. Clearly, the results obtained in this study highlight the need for care to be exercised in the design of systems for cooling a steel containment shell by naturally-induced flow of air so as to ensure that the kind of buoyancy-influenced conditions which prevailed in the present experiments are avoided.

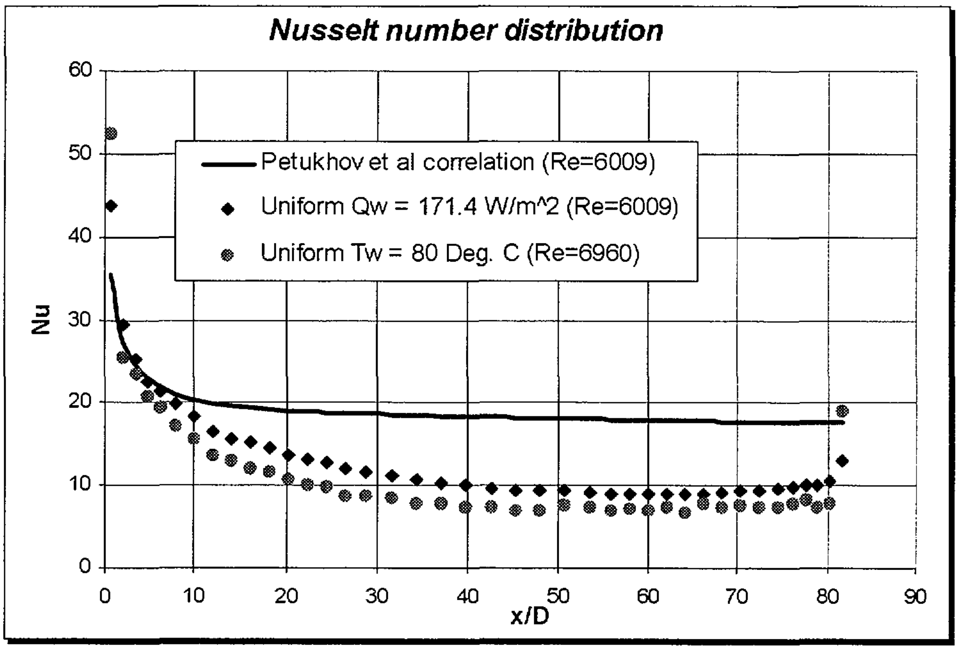

FIG. 6(a). Heat transfer results for naturally-induced flow with Tw = 800C.

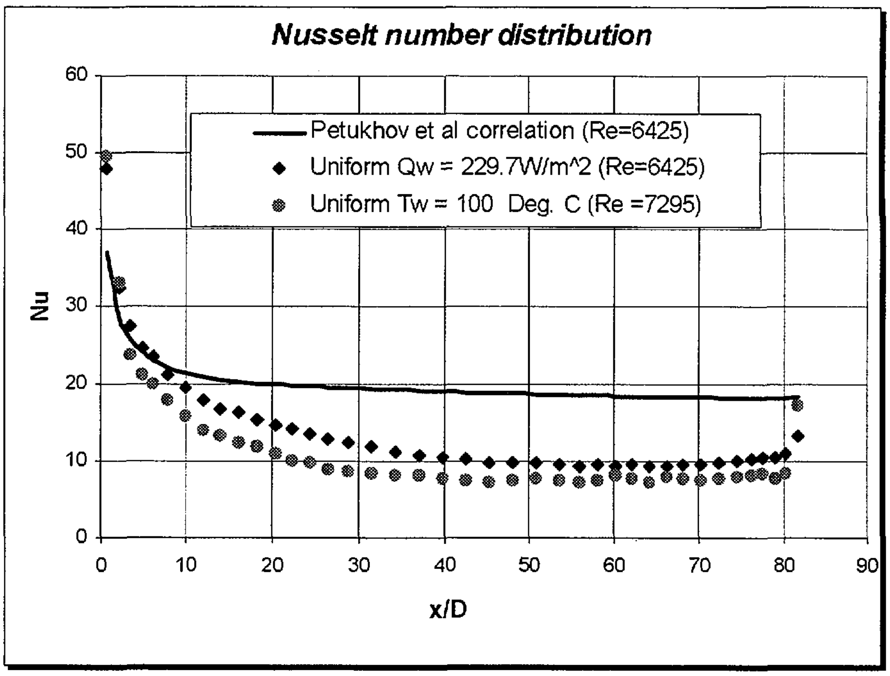

FIG. 6(b). Heat transfer results for naturally induced flow with Tw = 100°C

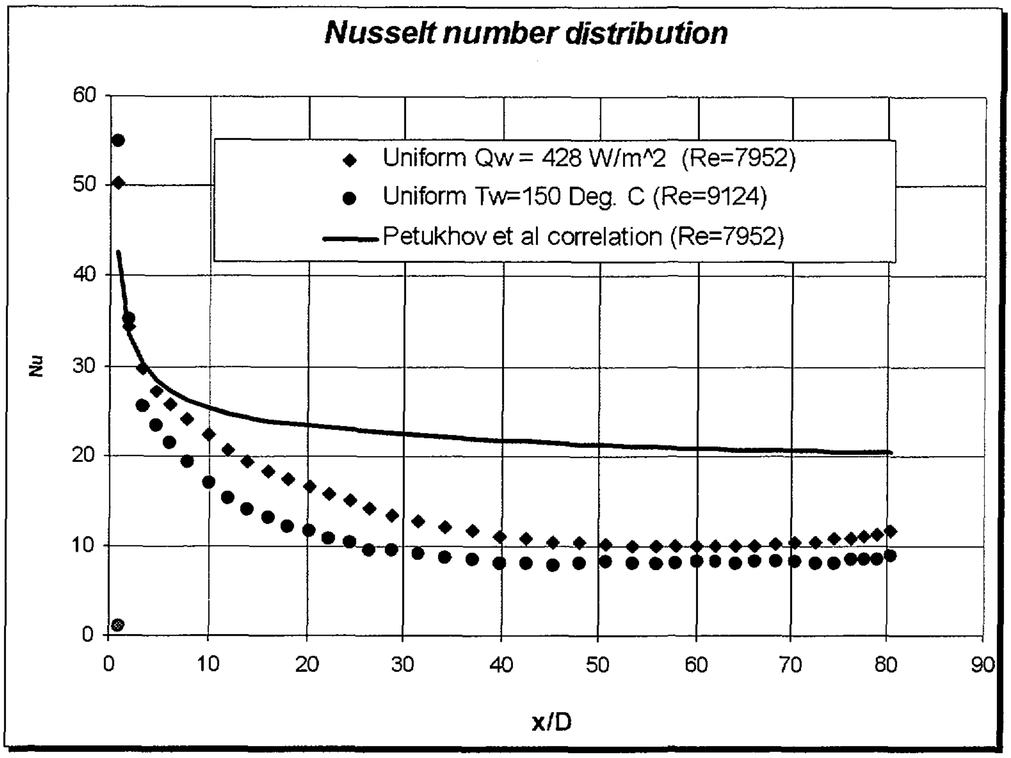

FIG. 6(c). Heat transfer results for naturally-induced flow with Tw = 150^{0}C

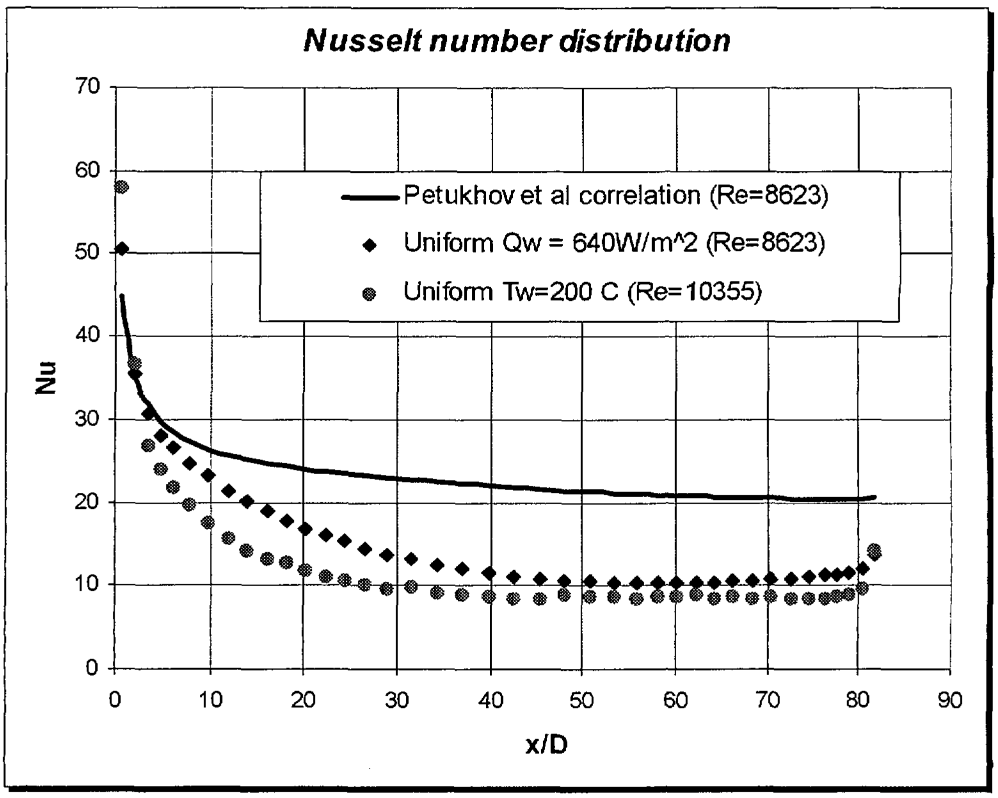

FIG. 6(d). Heat transfer results for naturally induced flow with Tw = 200^{0}C

3.2. Experiments with pumped flow

After completing the naturally-induced flow experiments reported in the preceding sub-section, a programme of pumped flow experiments with uniform wall temperature was carried out. The conditions of heat transfer achieved in those experiments varied from forced convection with negligible influences of buoyancy to mixed convection with very strong influences of buoyancy. Inlet Reynolds number was varied in steps from about 30,000 down to 3,500 and values of wall temperature from 75°C to 300°C were covered. Again, corresponding experiments were performed with uniform heat input.

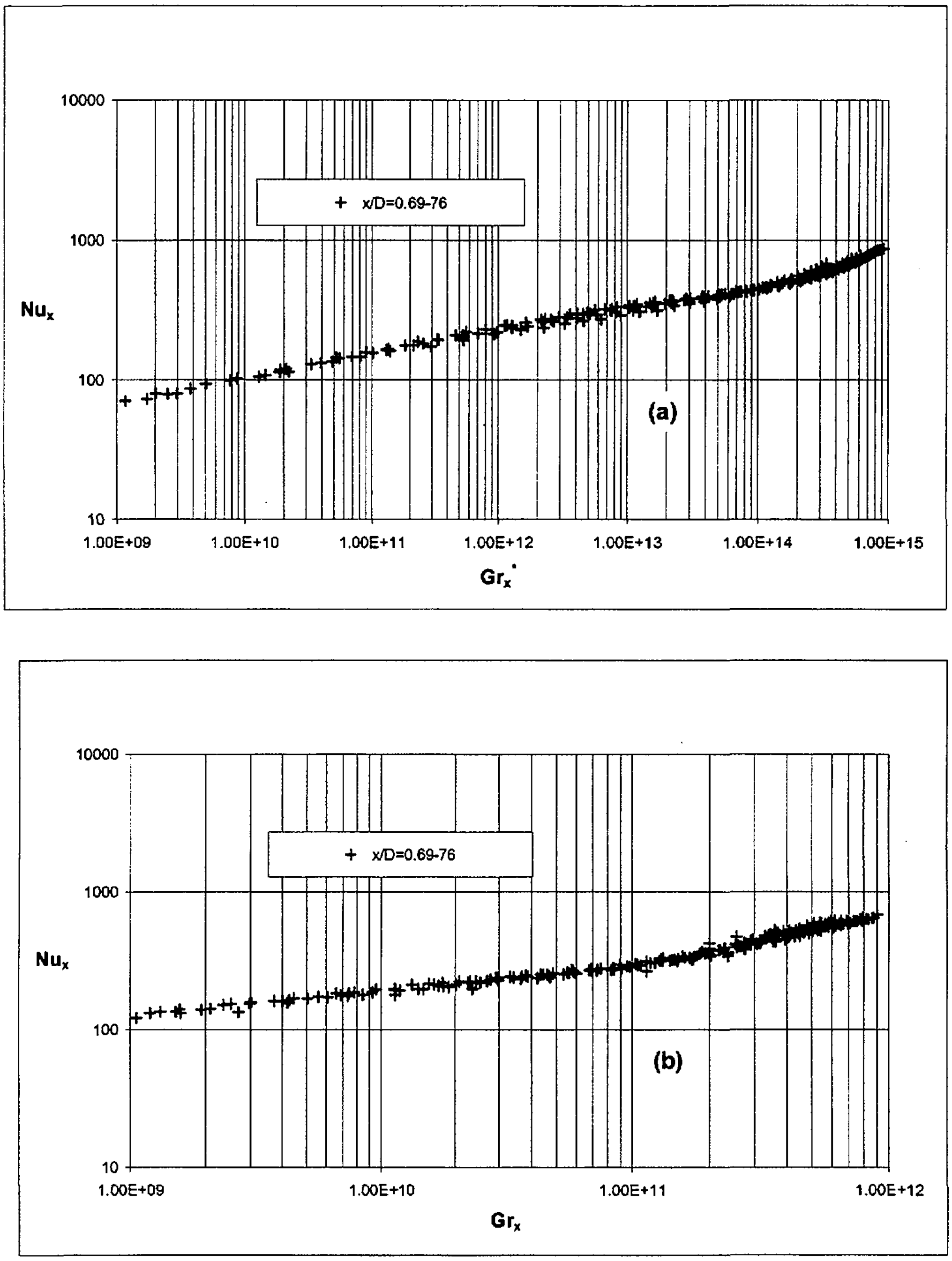

FIG. 7. Correlation of heat transfer results for naturally induced flow with (a) uniform wall heat flux, (b) uniform wall temperature

Some sample results for a wall temperature of 150°C are shown in Figures 8(a) to 8(d) along with the corresponding ones for uniform heat flux. The Reynolds numbers at inlet were about 29,000, 10,500, 5100 and 3500. The main points to note are:

(i) The heat transfer mechanism at the highest Reynolds number is turbulent forced convection. As can be seen from Figure 8(a), the results for the uniform heat flux case lie very close to the curve calculated using the Petukhov equation. Thus there are no significant influences of buoyancy. As expected, the results for uniform wall temperature lie slightly below those for uniform wall heat flux.

(ii) At the Reynolds number of 10,500, Figure 8(b), the mechanism of heat transfer is mixed convection with serious impairment of heat transfer due to influences of buoyancy on turbulence. The effect is much greater for uniform wall temperature than for uniform heat flux. The results for uniform wall temperature are similar to those obtained for naturally-induced flow with a wall temperature of 200°C (Figure 6(d)).

(iii) In the case of the experimental results shown in Figures 8(c) and 8(d) the conditions are those of mixed convection with progressively stronger influences of buoyancy leading to recovery and enhancement of heat transfer. Concentrated non-uniformities appear in the distributions of Nusselt number as a consequence of the development of the flow along the tube under the influence of strong buoyancy forces.

Figures 9(a) and 9(b) shows results from the pumped flow experiments for a large value of x/D presented in terms of Nusselt number ratio and buoyancy parameter. Experimental data for naturally induced flow are also included (in these cases the Reynolds number was evaluated using the value of flow rate induced through the system). The impairment of heat transfer which develops with increase of buoyancy parameter is strikingly apparent in both cases, as also is the recovery of heat transfer (in relative terms) when buoyancy influences become strong enough to restore turbulence production. As can be seen, when presented in this form the naturally-induced flow results map directly onto the pumped flow results.

Although the results obtained in the pumped flow experiments confirm that the general pattern of behaviour in buoyancy-aided mixed convection is the similar with uniform wall temperature and uniform wall heat flux, the impairment of heat transfer which develops with onset of buoyancy influences does occur more readily in the case of uniform wall temperature. The results highlight the fact that care is needed in the design of containment cooling systems to avoid conditions under which buoyancy-induced impairment of heat transfer might occur.

4. COMPUTATIONAL SIMULATIONS OF THE EXPERIMENTS

An advanced computer code CONVERT, developed and validated earlier at the University of Manchester for buoyancy-influenced flow in uniformly heated vertical tubes, was used to perform simulations of the present experiments. This code uses a buoyancy influenced, variable property, developing wall shear flow formulation for turbulent flow and heat transfer in a vertical tube in conjunction with the Launder-Sharma low Reynolds number k-ε turbulence model [9]. The conditions covered in the simulations ranged from forced flow with negligible influence of buoyancy to buoyancy-dominated mixed convection. In each case, simulations were made for thermal boundary conditions of both uniform wall temperature and uniform heat flux. These show that the computational formulation used does enable observed heat transfer behaviour in the mixed convection region to be reproduced. Buoyancy-induced impairment of heat transfer is predicted and satisfactory agreement with experiment is found for such conditions. The non-uniformities which develop in the distributions of Nusselt number under such conditions are well reproduced. It is known from earlier work done by the present authors that other turbulence models are generally less successful than the Launder Sharma model in reproducing the influences of buoyancy on turbulence and heat transfer found in mixed convection. Thus, it is important in the computational modelling of containment cooling systems that an appropriate turbulence model is used.

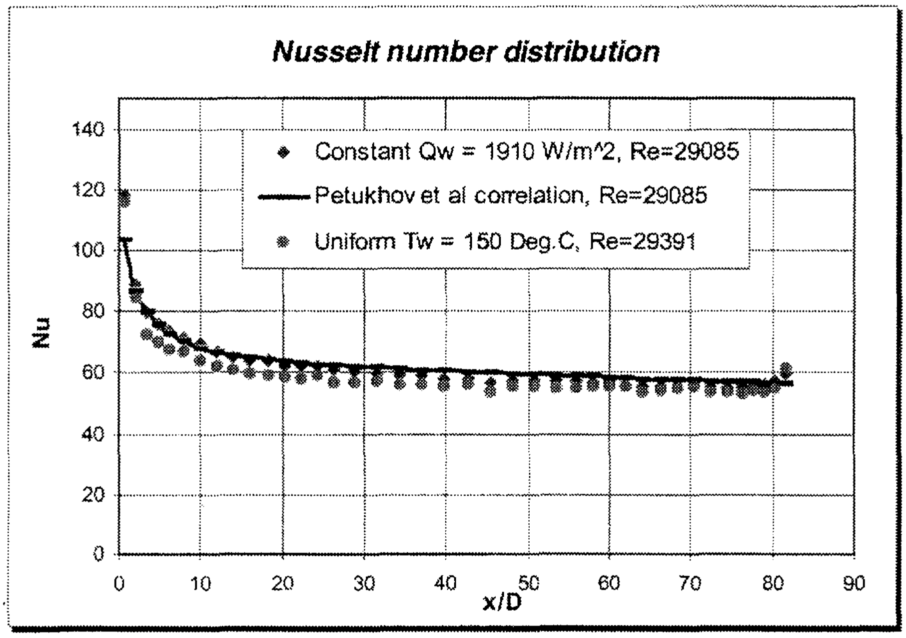

FIG. 8(a). Heat transfer results for pumped flow with Tw = 150°C at Re = 29,000

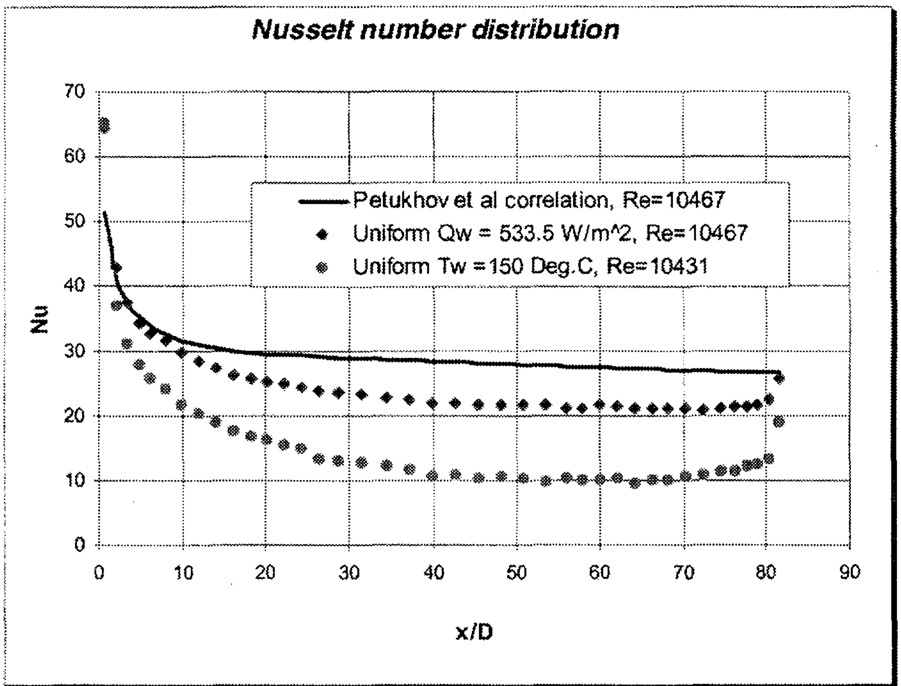

FIG. 8(b). Heat transfer results for pumped flow with Tw = 150°C at Re = 10,000

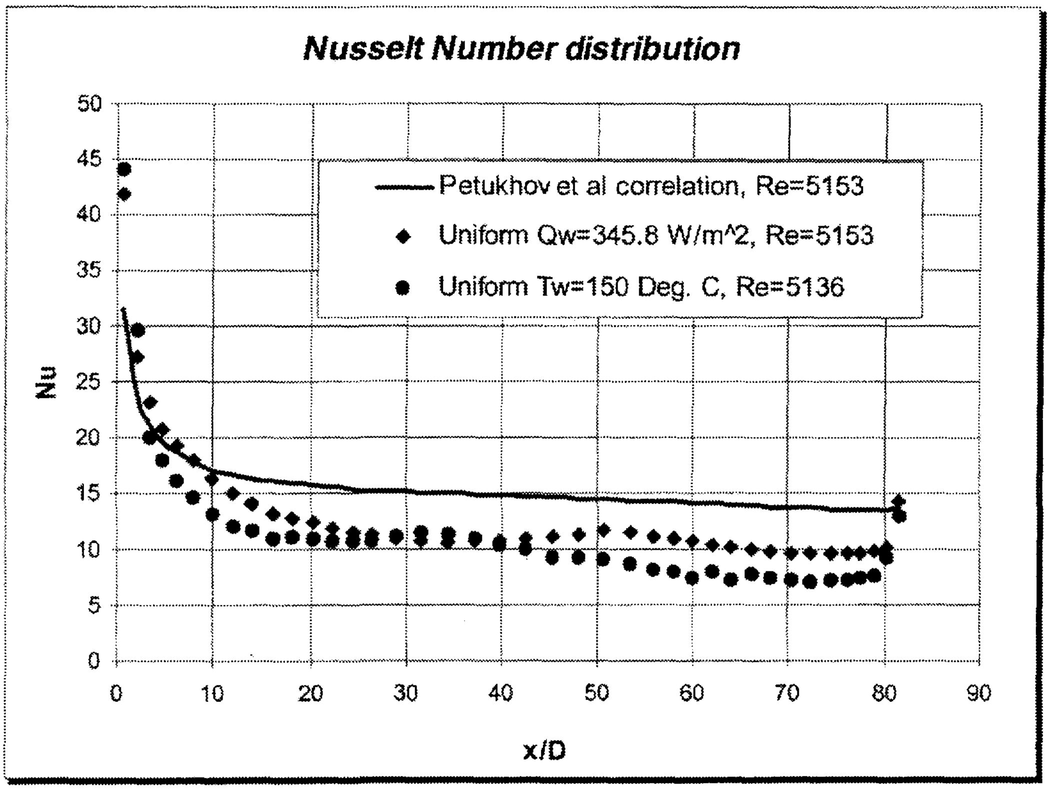

FIG. 8(c). Heat transfer results for pumped flow with Tw = 150°C at Re = 5,400

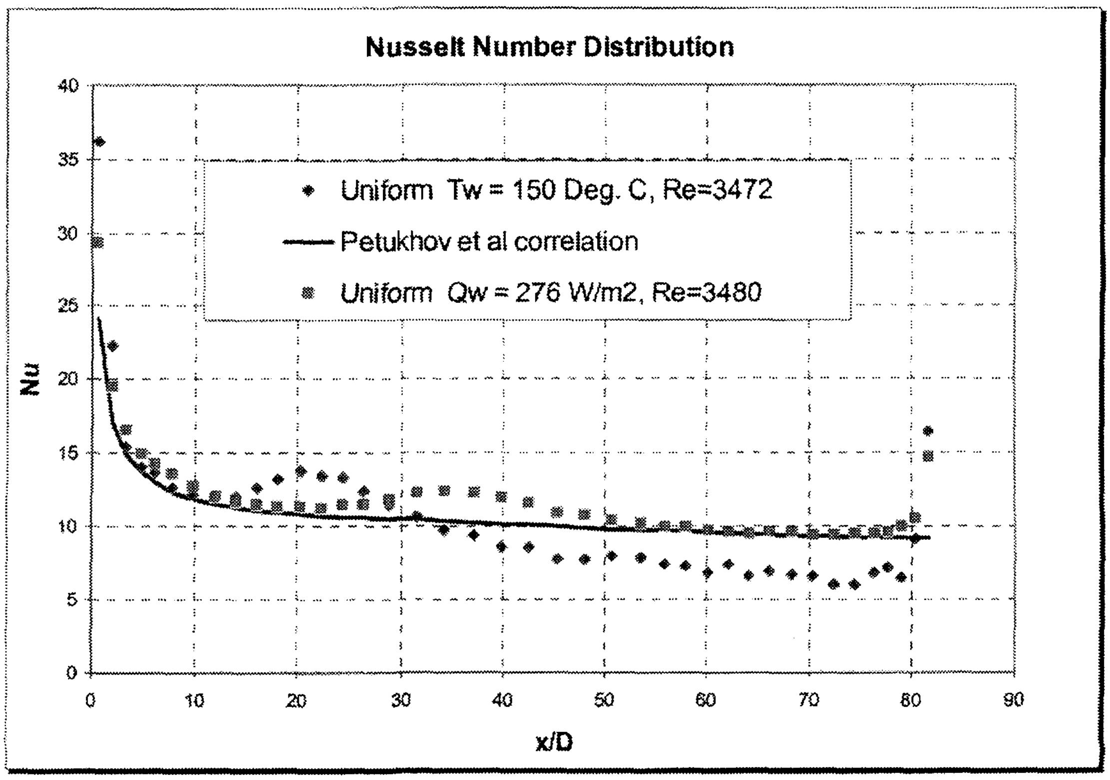

FIG. 8(d). Heat transfer results for pumped flow with Tw = 150°C at Re = 3,500

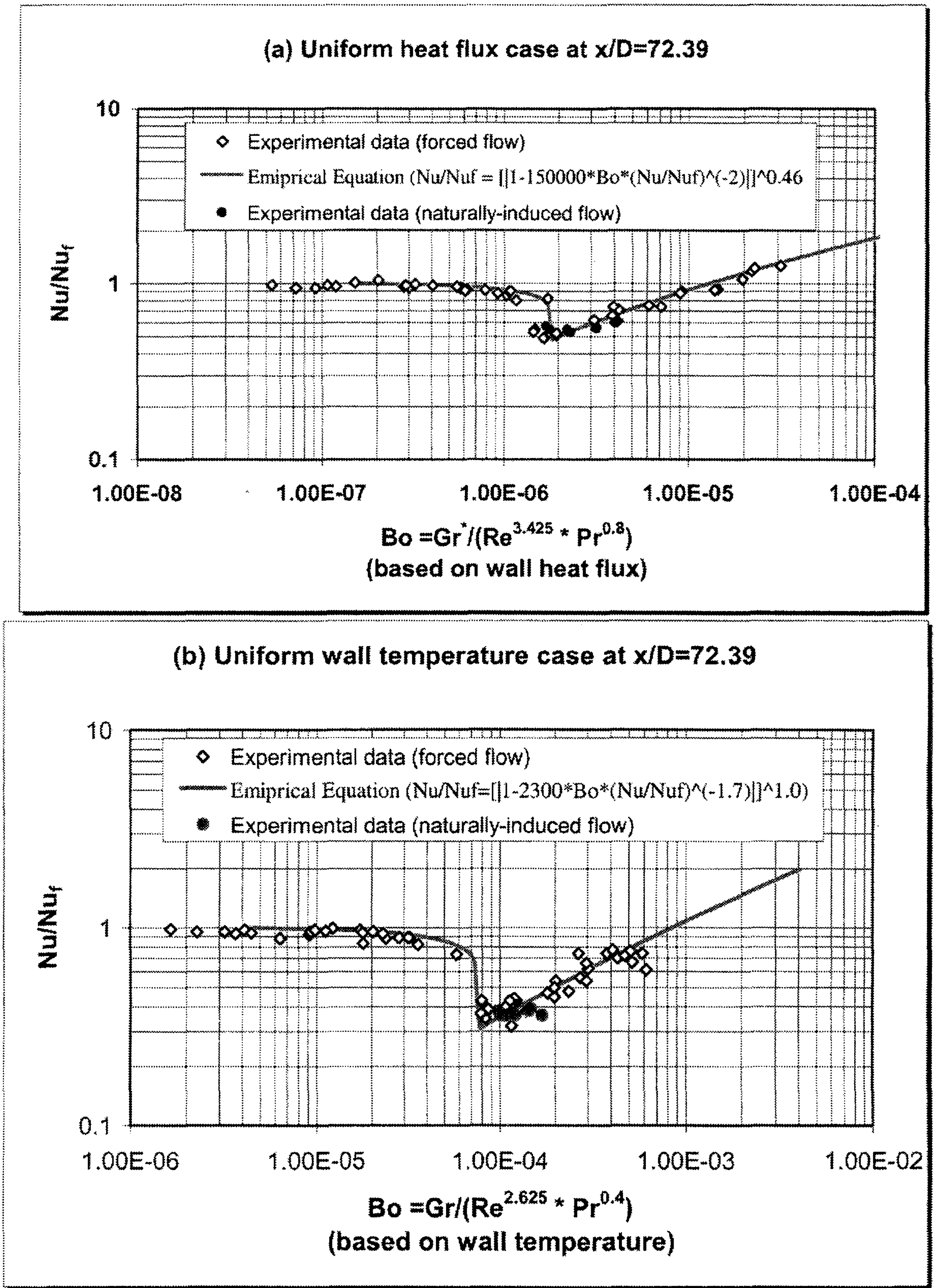

FIG. 9. Correlation of heat transfer data for large x/D in terms of Nusselt number ratio and buoyancy parameter for (a) uniform wall heat flux, (b) uniform wall temperature

5. CONCLUSIONS

Even though the flow rates achieved in the naturally-induced cooling experiments with uniform wall temperature were such that the flow would have been turbulent in the absence of buoyancy influences, the effectiveness of heat transfer was seriously impaired in relation to that for turbulent forced convection. It is clear that under the conditions of all the experiments performed in this study turbulence was strongly impaired as a result of the influence of buoyancy. The heat transfer behaviour did not change very much as wall temperature was raised and was very similar in corresponding experiments with uniform wall heat flux. The results obtained highlight the need for care to be taken in the design of systems for cooling a steel containment shell by a naturally-induced flow of air over it so as to ensure that the buoyancy-influenced conditions which prevailed in the present experiments are avoided.

Although the results obtained in the pumped flow experiments confirm that the general pattern of behaviour in buoyancy-aided mixed convection with uniform wall temperature is similar to that with uniform wall heat flux, impairment of heat transfer with onset of buoyancy influences was found to occur more readily with uniform wall temperature.

The computational simulations showed that the formulation used does enable observed heat transfer behaviour to be satisfactorily reproduced throughout the mixed convection region. It is known from earlier work (Reference [10]) that other turbulence models are generally less successful in reproducing the influences of buoyancy on turbulence and heat transfer found in vertical passages. Thus, in the computational modelling of containment cooling systems it is important to use an appropriate turbulence model and to be aware of the limitations of such models.

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