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Temperature Measurements in Buoyancy-Opposed Turbulent Mixed Convection

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Abstract: Fluid temperature fluctuations were measured during heat transfer to water at Reynolds numbers of 2750 to 57000 and Grashof number of $1.5x10^8$. Experiments were performed for downward flow in a vertical, uniformly heated, stainless steel pipe having a diameter of 48 mm. The measurements show that the influences of buoyancy forces on the profiles of mean and fluctuating temperature are minimal for the range of parameters investigated.

Key Words: Temperature fluctuations, temperature profile, mixed convection, buoyancy

Türbülanslı Karışık Konveksiyonda Sıcaklık Ölçümleri

Özet: Grashof sayısının $1.5x10^8$ değerinde Reynolds sayısı 2750 ile 57000 arasında değiştirilerek suya ısı transferinde akışkan sıcaklıkları ölçülmüştür. Deneyler aşağıya doğru akışta 48 mm çaplı, paslanmaz çelikten imal edilmiş, ısıtılmış düşey bir boru içerisinde gerçekleştirilmiştir. Ölçümler, göz önüne alınan parametre sınırları içerisinde kaldırma kuvvetlerinin ortalama ve çalkantı sıcaklığı profilleri üzerindeki etkisinin ihmal edilebilecek düzeyde olduğunu ortaya koymuştur.

Anahtar Sözcükler : Çalkantı sıcaklıkları, sıcaklık profili, karışık konveksiyon, kaldırma kuvvetleri

Introduction

Under certain conditions, the influences of buoyancy forces on forced convection heat transfer are not negligible, and in such cases the thermo-fluid regime is usually called "mixed convection".

Heat transfer to turbulent flow in a vertical tube under conditions of mixed convection can exhibit marked departures from the behaviour found in forced convection.

A number of experimental and computational investigations of turbulent mixed convection in vertical tubes have been reported (see Jackson et al., 1988, for a detailed review). Based on the results of these studies, it is well established that in heated descending flows (buoyancy-opposed), heat transfer is always enhanced relative to the forced convection value, whereas in the case of heated ascending flows (buoyancy-aided), it may be either increased or decreased depending on the axial location and the magnitude of buoyancy forces.

Measurements of mean and fluctuating components of the temperature field mainly in air pipe flow under buoyancy-aided flow conditions have been reported (Byrne and Ejiogu, 1971; Carr et al., 1973; Polyakov and Shindin, 1988) to obtain an insight into the influences of buoyancy on the mechanism of turbulent heat transfer and an explanation for the above-mentioned trends.

In this study, buoyancy-opposed mixed convection is investigated. For this case, there have been only few experimental studies in which temperature field has been considered.

Easby (1978) performed experiments for downward flow of nitrogen in a heated pipe for Reynolds numbers between 2000 and 10000 and Grashof numbers up to 10^6.

Axcell and Hall (1978) obtained data using air for a Reynolds number and Grashof number range of $2x10^4$ - $1.3x10^5$ and $9.3x10^8$ - $2.17x10^9$, respectively. Neither Easby (1978) nor Axcell and Hall (1978) presented temperature fluctuations. They reported only mean temperature profiles.

Temperature fluctuation measurements were presented only by Brown and Gauvin (1966).

They studied heat transfer with air under the conditions of both buoyancy-aided and buoyancy-opposed mixed convection. Their Reynolds number range was between 270 to 6900.

In addition to these experimental studies, Swanson and Catton (1987) analyzed the unsteady behaviour of a clump of fluid for two-dimensional parallel flow near a wall. As an outcome of this study, they proposed a correlation equation for the temperature profiles in the case of buoyancy-opposed mixed convection.

Although mixed convection has received intensive attention and there exist various correlation equations for the heat transfer coefficient under mixed convection, understanding of the heat transer mechanism under buoyancy influences has remained rather incomplete. To the author's knowledge, in the open literature there has been no study with water for buoyancy-opposed mixed convection in which temperature measurements in the fluid have been made.

This study has been carried out to fill this gap. Measurements were made for downward flow of water in a vertical tube with uniform heat flux.

Experimental arrangement

The apparatus used in this study is shown in Figure 1. It consists of a vertical test pipe, two water storage tanks, a flow control valve, a flow-meter, a circulating pump and other piping and instrumentation. The water was circulated from the top tank via the test pipe, the flow control valve, the flow-meter, the bottom tank and then back to the top tank.

The test tube was made of a stainless steel pipe of 48.34 mm ID and 1.29 mm wall thickness. The length of the test tube was 7.45 m, and 3 m of it was heated by alternating current. The first 76 diameters of the tube served as a hydrodynamic developing section and the heated length was about 62 diameters. The test pipe was thermally insulated using glass fibre to minimize heat losses to the surroundings.

Measurements of the wall and fluid temperature were made at a distance of 2.625 m (54 diameters) from the point where heating started. The wall temperature was measured by a chromel-alumel thermocouple resistance welded to the outer surface of the tube wall. The welding process was performed under a binocular microscope, and very careful attention was paid to precision in terms of axial location to minimize AC voltage pick-up.

A traversing mechanism enabled temperature distributions in the fluid side to be measured by means of a 0.2 mm diameter thermocouple which was capable of responding to the turbulent fluctuations of temperature. Measurements are made at 19 different radial positions. The bulk temperature of the water was measured at inlet to the test section and also at outlet from the test section by thermocouple probes inserted into the flow.

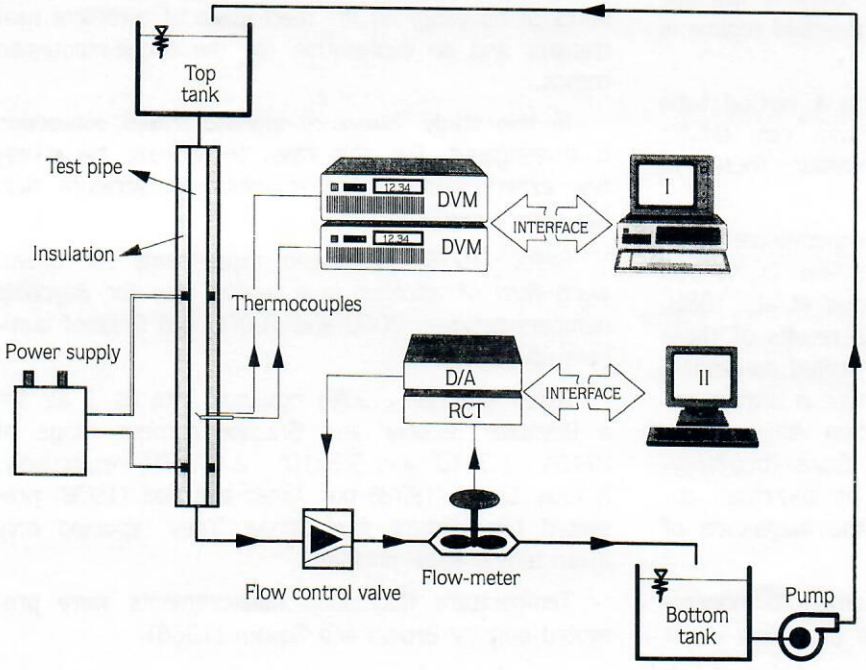


Figure 1. Experimental setup.

The flow-meter signals were read by a computer (II) and the same computer also supplied the signals to activate the flow control valve. The thermocouple emf's were first converted to digital signals using a pair of high-accuracy microprocessor digital voltmeters (one for wall temperature and one for fluid temperature) and then read by a second computer (I).

The electrical power input to the test section was kept constant at about 8800 W/m^2 during the experiments, and flow rate was varied from one experiment to another between 0.4 m^3/h to 8 m^3/h. These values correspond to a Reynolds number of 2700-57000 and a Grashof number of $1.5x10^8$ at inlet.

The uncertainty analyses of the overall experimental procedures using the method described by Moffat (1988) showed that there was ±0.2 °C uncertainty in temperature measurement and ±1.5 percent in flow rate measurement. The uncertainty in the heat flux which arose from uncertainty in estimating heat losses to the surroundings and from uncertainty in measuring electrical power input to the test tube was ±2 percent. The uncertainty in setting radial distance from the tube wall was ±0.01 mm.

Results and Discussion

Increase of the Grashof number for a constant Reynolds number or decrease of the Reynolds number for a constant Grashof number leads to larger buoyancy forces within the flow (Büyükalaca, 1995a). In order to evaluate the relative magnitude of the buoyancy forces, the parameter $BP=Gr^*/(Re^{3.425}\ Pr^{0.8})$, suggested by Jackson and Büyükalaca (1993), was calculated for each case tested in this study and is shown in Table 1. The increase of the heat transfer coefficient with respect to the corresponding forced convection value, which was determined from a correlation proposed by Büyükalaca (1995b), is also shown in this table. Büyükalaca (1995b) studied influences of buoyancy on the heat transfer coefficient in a relatively larger diameter tube (98.3 mm ID) using water.

Table 1. Main experimental parameters.

Run No	Re_{in}	BP_{in}	Increase of Nu [%]
1	2750	$5.5x10^{-5}$	60
2	4300	$1.1x10^{-5}$	25
3	5700	$4.3x10^{-6}$	12
4	6700	$2.5x10^{-6}$	8
5	7400	$1.8x10^{-6}$	6
6	9600	$7.0x10^{-7}$	2
7	11600	$3.8x10^{-7}$	1
8	13300	$2.3x10^{-7}$	-
9	14600	$1.7x10^{-7}$	-
10	18800	$7.1x10^{-8}$	-
11	22200	$4.0x10^{-8}$	-
12	31100	$1.3x10^{-8}$	-
13	38000	$6.4x10^{-9}$	-
14	44900	$3.6x10^{-9}$	-
15	51200	$2.3x10^{-9}$	-
16	57500	$1.6x10^{-9}$	-

In Figure 2, measured temperature profiles are illustrated. The magnitude of T_w-T_f for a fixed heat flux

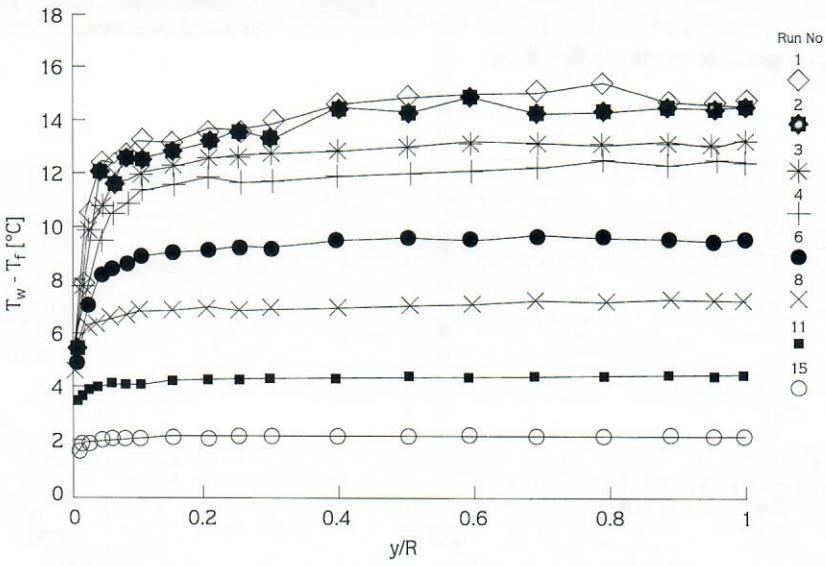


Figure 2. Measured mean fluid temperature profiles.

(q_w =8800 W/m^2), as expected, decreases with the increase of the flow rate.

It is seen that as Re increases (BP decreases), the temperature profiles become flatter over most of the pipe cross section. With the increase of Re, the thickness of the viscous sublayer decreases and the eddy diffusivity increases in the turbulent region. The combined effect of variation in thickness of the viscous sublayer and eddy diffusivity leads to more square temperature profiles. As can be seen from Figure 2, at high flow rates the resistance to heat transfer is increasingly in the sublayer, whereas at low flow rates it is distributed over the entire cross section. At the highest flow rate, the heat transfer resistance is almost entirely in the sublayer and the temperature profile in the fully turbulent part of the flow is virtually flat.

The variation of temperature profiles with Reynolds number (or BP) is monotonic. The profiles do not appear to be greatly affected by buoyancy forces. This can be seen more clearly when the profiles are plotted on a nondimensional basis using $(T_w-T_f)/(T_w-T_c)$. In this case (Figure 3), all of the profiles lie nearly on the same curve over most of the cross section of the tube.

Temperature profiles similar to those in the present experiment were measured by Brown and Gauvin (1966) and Axcell and Hall (1978) for air. Their temperature profiles also showed no significant distortions. As buoyancy forces increase, in contrast to the profiles of temperature, both Axcell and Hall (1978) and Easby (1978) noted a change in the shape of velocity profiles in the sense that the velocity is reduced near the wall region and increased at the tube axis.

Although the shapes of the temperature profiles are not influenced considerably by increasing buoyancy forces, the points tend to scatter with the increase of BP. This is believed to be real fluctuation and can be attributed to the forming of circulating flow cells which are a consequence of increased shear stress due to buoyancy forces within the flow. The increased shear stress leads to enhanced heat transfer (Jackson and Hall, 1979). The presence of circulating cells when buoyancy forces are large enough is also reported by Swanson and Catton (1987) and Petukhov (1976).

The influence of the buoyancy forces on the root-mean-square (RMS) fluctuations of fluid temperature is illustrated in Figure 4. At high flow rates, the fluctuations die out very rapidly near the wall, showing that the buoyancy forces are not sufficiently strong to penetrate to the core region. The magnitude of the fluctuations increases considerably as the flow rate decreases (buoyancy forces increases). This observation is in agreement with Brown and Gauvin's (1966) data for air. It is clear that the shape of the RMS fluctuations of fluid temperature is not affected by the buoyancy, as in the case of mean fluid temperature.

As seen in Figure 4, RMS temperature fluctuations exhibit a peak near the wall. The position of the peak depends on the Reynolds number. With the increase of the Reynolds number the position of the peak shifts closer to the wall. This behaviour again is in good

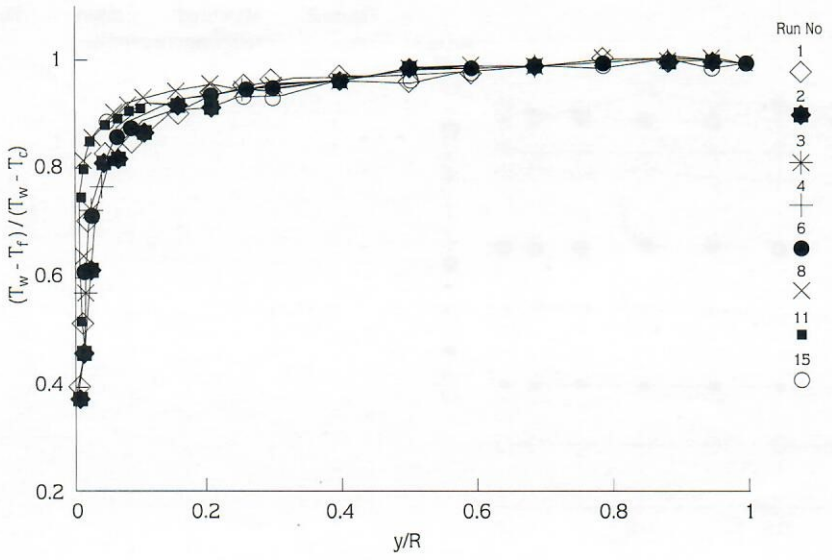


Figure 3. Nondimensional mean fluid temperature profiles.

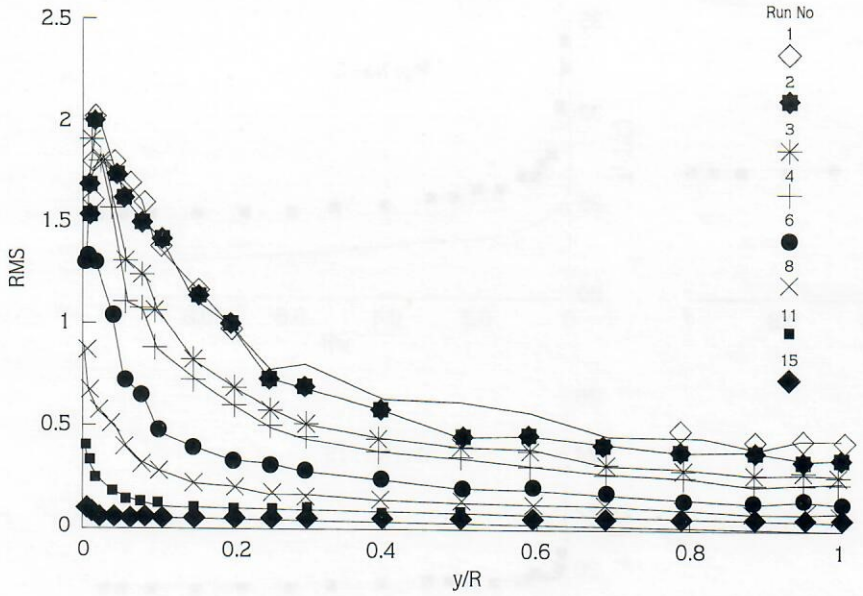


Figure 4. RMS fluctuations of fluid temperature.

agreement with the air data of Brown and Gauvin (1966). However, due to limitations of the temperature measurement arrangement, it was not possible to make measurements very close to the wall. Thus, it was not possible to detect the position of the peaks at high Reynolds numbers (Re > 12000).

Brown and Gauvin (1966) and Tanimoto and Hanratty (1963) noted that there is a very strong resemblance between both the shape and the magnitude of temperature fluctuations and the axial velocity fluctuations, down to a Reynolds number of about 10000 for air. This resemblance may possibly be connected with the fact that the Prandtl number of air is about 1, thus leading to a similarity between velocity and temperature fluctuations (Brown and Gauvin, 1966).

However, in the case of water the similarity between temperature and velocity fluctuations can no longer be seen. The data of He (1992) for water clearly shows that with the increase of Reynolds number, the magnitude of all three components (axial, radial and circumferential) of velocity fluctuation increases.

Kays and Crawford (1993) developed the following expression using a two layer model for the fully developed temperature distribution in a smooth pipe in the case of pure forced convection:

$$T^+ = 2.2 \ln y^+ + 13.39 \, Pr^{2/3} - 5.66 \qquad (1)$$

Equation 1 was reported to be valid for fluids with Prandtl numbers between 0.6-6. It can not be used inside the viscous sublayer and it is not precisely valid at the tube centerline.

The temperature profiles computed from Equation 1 for four different values of the Reynolds number are plotted in Figure 5 together with the data of the present study.

Equation 1 fails to predict the temperature profiles for low Reynolds numbers (large buoyancy forces) such as Run 1 and 2. For these cases the heat transfer process and, as a consequence, the temperature field are greatly affected by buoyancy forces, and the equation underpredicts the measured temperatures unacceptably. The increase of the heat transfer coefficient over the pure forced convection value is about 60% and 25%, respectively, according to Table 1.

However, as the buoyancy force decreases (Reynolds number increases), Equation 1 gets closer and closer to the experimental points and at a BP of about 4.3×10^{-6} (Run 3) a good agreement between the two is seen except near the wall region. In this region Equation 1 produces more flattened temperature profiles than the experimental ones. The discrepancy between the two is an expected result in view of the fact that Equation 1 is only valid in the fully turbulent core region.

The same data is plotted on a nondimensional basis in Figure 6. As can be seen in Figure 6, over BP 4.3×10^{-6}, the data is well presented by Equation 1 down to about $y^+=10$, where the effects of the viscous sublayer begin to be felt, although some discrepancies are evident between the two in some cases. This can easily be attributed to the uncertainties involved in the experiments and the assumptions made when obtaining the equation.

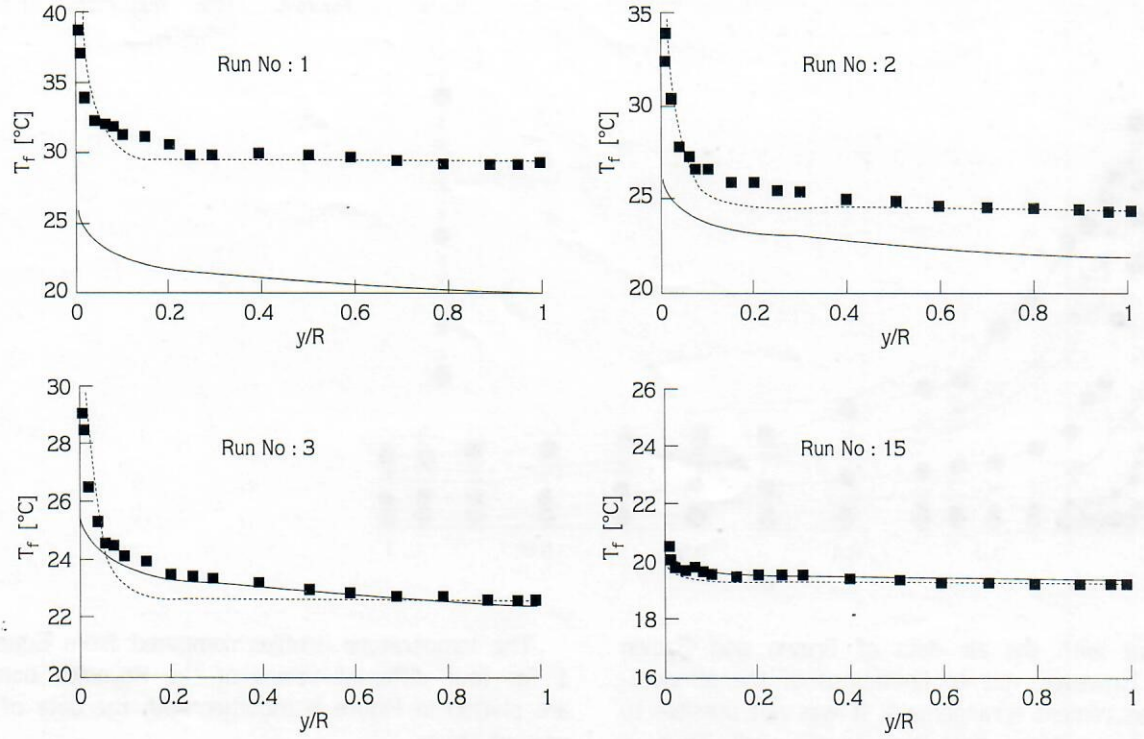


Figure 5. Comparison of experimental (symbols) temperature profiles with those computed from Kays and Crawford, Equatin 1 (solid line) and Swanson and Catton, Equation 2 (dashed line).

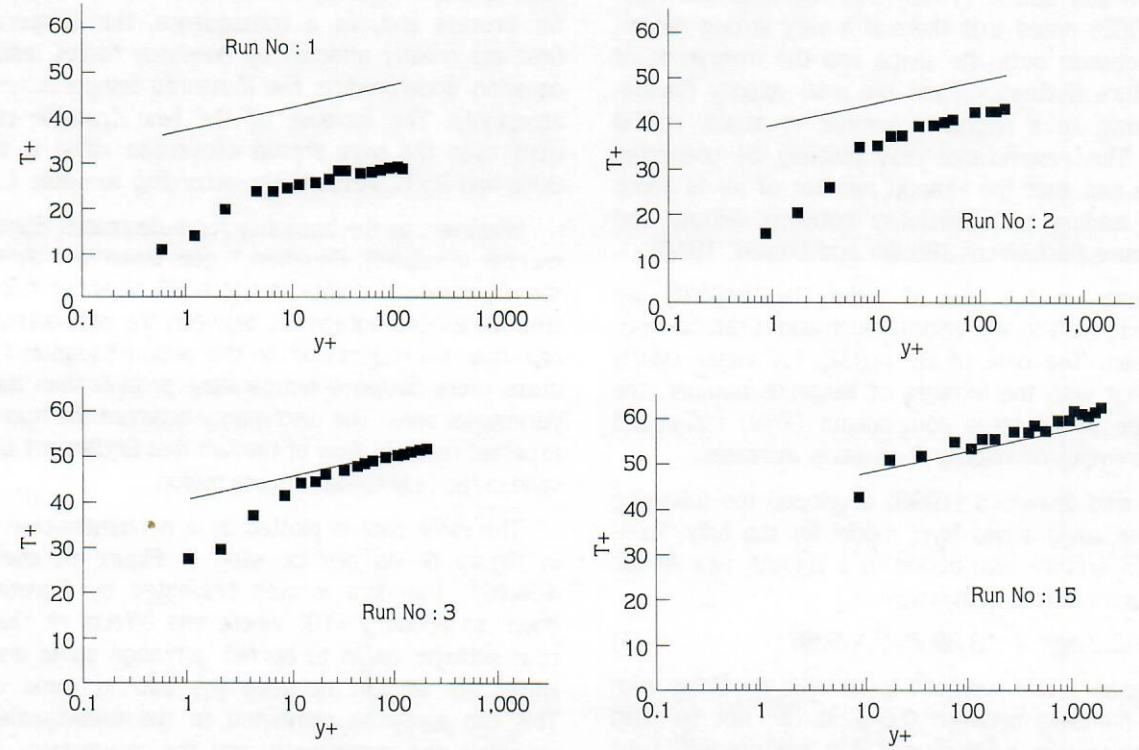


Figure 6. Comparison of experimental (symbols) temperature profiles with those computed from Kays and Crawford, equation 1 (solid line).

It should be noted that, although the measurements are made at fixed radial positions (constant y) for each experiment, with the increase of the Reynolds number, the corresponding y^+ values increase. Thus at higher Reynolds number values, it was not possible to make measurements within the viscous sublayer.

The only attempt to obtain the velocity and the temperature profiles analytically in the case of mixed convection was made by Swanson and Catton (1987), utilizing the surface renewal theory proposed originally by Higbie (1935), Danckwerts (1951) and Einstein and Liu (1956). The surface renewal theory is based on the flow mechanism described as follows: a clump of fluid from the core region enters the wall layer and strikes the wall, where it remains stationary for awhile. After absorbing energy, the clump then bursts off the wall and passes through the layer, returning to the core region. Since most of the thermal resistance occurs near the wall, an accurate description of the heat transfer phenomena in this layer can produce a good estimation of the overall process (Swanson and Catton, 1987).

Swanson and Catton (1987) applied the surface renewal theory to the problem of buoyancy-opposed mixed convection and derived the following equation for temperature distribution:

$$T^* = e^{-yNu} \qquad (2)$$

in which Nu is the heat transfer coefficient and has been obtained by

$$Nu = 0.0115\ Re^{0.8}\ Pr^{0.5} \left[1+\left(1-\frac{696}{Re^{0.8}}+8300\ \frac{Gr}{Re^{2.6}\,(Pr^{0.5}+1)}\right)^{0.39}\right]$$

This equation was reported to be valid only near the wall and less dependable as the core region is approached. It should be noted that in deriving Equation 2, Swanson and Catton (1987) did not take into account changes in the friction factor under the influence of buoyancy, instead utilised an empirical equation which is valid for pure forced convection.

The nondimensional temperature profiles calculated from Equation 2 for some of the experimental conditions of the present study are plotted in Figure 7. An increase in thermal boundary layer thickness is evident from the figure with a decrease in the Reynolds number. However, for the conditions of Run 1 and Run 2, Equation 2 produces almost the same temperature profile. This can be explained by considering the assumption on which Equation 2 was based. The heat transport mechanism was assumed to be controlled by fluid mixing and it depends on the relative magnitudes of two forces, namely viscous shearing force and buoyancy force. With the decrease of the Reynolds number the viscous shearing force decreases, but it is balanced by an increase in buoyancy force, and as a result the same thermal field is predicted for Run 1 and Run 2.

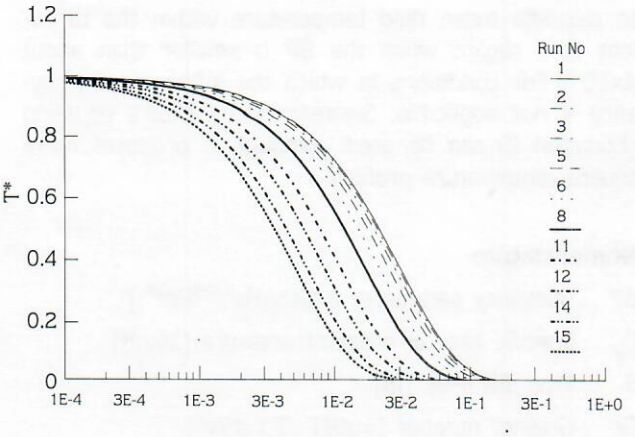


Figure 7. Fluid temperature profiles calculated from Swanson and Catton, Equation 2.

A comparison of Equation 2 and the present experimental data is shown in Figure 5. Kays and Crawford (1993) and Swanson and Catton (1987) defined nondimensional temperature differently. In order to compare the results of Equation 1 and Equation 2 with the present experimental data on the same graph, in Figure 5 dimensional temperature is used.

Agreement between the data and Equation 2 is seen to be close even for conditions where the influence of buoyancy is significant (Run 1 and 2). However, Equation 2 can not reproduce precisely the variation of mean fluid temperature with radial distance. It produces a more square temperature profile than the experimental data. This is much more pronounced for lower buoyancy cases.

Conclusions

It has been shown that buoyancy forces have little effect on the profiles of mean and RMS fluctuation of fluid temperature when buoyancy forces oppose the forced flow.

Comparison of the present data with the equation of Kays and Crawford (Equation 1) for pure forced convection reveal that Equation 1 can safely be used

to calculate mean fluid temperature within the turbulent core region when the BP is smaller than about $4x10^{-6}$. For conditions in which the influence of buoyancy is not negligible, Swanson and Catton's equation (Equation 2) can be used, although it produces more square temperature profiles.

Nomenclature

BP : Buoyancy parameter $[=Gr^*/(Re^{3.425}Pr^{0.8})]$

C_p : Specific heat at constant pressure [J/kgK]

d : Pipe diameter [m]

Gr : Grashof number $[=g\beta(T_w-T_c)\ d^3/\nu^2]$

Gr* : Modified Grashof number $[=g\beta q_w d^4/k\nu^2]$

h : Heat transfer coefficient $[=q_w/(T_w-T_b)]$

k : Thermal conductivity [W/mK]

Nu : Nusselt number [=h d/k)

Pr : Prandtl number

q : Heat flux [W/m^2]

r : Radial distance measured from pipe centre [m]

R : Pipe radius [m]

Re : Reynolds number [=U d/ν]

T : Temperature [°C]

T^+ : Nondimensional temperature $[=(T_w-T_f)U_\tau/(q_w/\rho C_p)]$

T* : Nondimensional temperature $[=(T_f-T_c)/(T_w-T_c)]$

U_τ : Friction velocity $[=\sqrt{\tau_w/\rho}]$

y : Radial distance measured from pipe wall [m]

y^+ : Universal radial distance from pipe wall $[=(yU_\tau)/\nu]$

Greek:

β : Coefficient of volume expansion [1/K]

ρ : Density [kg/m^3]

ν : Kinematic viscosity [m^2/s]

τ : Shear stress [N/m^2]

Subscripts:

c : pipe centre

f : fluid

in : pipe inlet

w : wall

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