PERGAMON

International Journal of Heat and Mass Transfer 43 (2000) 4267–4274

International Journal of
HEAT and MASS
TRANSFER

www.elsevier.com/locate/ijhmt

Buoyancy effects on heat transfer and temperature profiles in horizontal pipe flow of drag-reducing fluids

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Received 12 November 1999; received in revised form 7 February 2000

Abstract

We have studied the extent to which buoyancy effects in horizontal pipe flows of drag-reducing viscoelastic fluids cause distortions to both laminar and turbulent temperature profiles. In the case of laminar flows, these distortions may lead to variations in Nusselt numbers that are larger than those seen for Newtonian pipe flows under similar conditions. In the case of turbulent drag-reducing flows, the effects of buoyancy can also be large and may in turn result in large errors in estimated Nusselt numbers if not properly accounted for. These errors are quantified and recommendations are made on how to reduce them.

1. Introduction

In heated flows in horizontal pipes, gravity-induced body forces may result from density variations within the fluid. Mori et al. [1,2] have shown flow visualization results for mixed convection in laminar flows of *Newtonian* fluids in horizontal pipes, demonstrating that the local variations in the fluid density lead to counteracting transverse vortices (or secondary flow patterns), which are superimposed to the main axial flow.

They also showed that the differences between their local measurements of temperature and velocity, and the theoretical laminar profiles with no buoyancy effects, could be as high as 50%, and that — for heating of the fluid — the point of lowest temperature was displaced from the center towards the lower portion of the pipe. All these experiments were conducted in the range of $Re \times Ra = 2 \times 10^4$–$1.6 \times 10^5$. On the other hand, their measurements of turbulent temperature and velocity profiles in the range of $Re \times Ra$ between 3.87×10^5 and 4.7×10^5 showed a negligible difference with respect to those of pure forced convection. Based on experimental data, Morcos and Bergles [3] provided averaged Nusselt number (Nu_{avg}) correlations which take into account the effect of circumferential heat flux variations for the problem of mixed convection on laminar flow of Newtonian fluids in horizontal tubes. Metais and Eckert [4] proposed practical charts in which the regions of forced and mixed convection for Newtonian laminar and turbulent flows in horizontal pipes can be clearly identified in terms of the *Re*, *Gr*, and *Pr*.

More recently, *viscoelastic* drag-reducing fluids, and particularly surfactant solutions, have attracted the attention of researchers because of their potential for energy savings applications. The buoyancy effects on laminar and turbulent flows for these fluids is not only interesting from a theoretical point of view, but also for the proper design of experimental procedures. Shenoy and Ulbrecht [5] studied the effect of natural convection on a laminar flow next to a vertical flat plate for various solutions of a viscoelastic fluid, and they found that the local convective heat transfer coefficients (h) were systematically higher for elastic fluids than for

0017-9310/00/$ - see front matter
PII: S0017-9310(00)00065-X

Nomenclature

$C_f = 2\tau_w/\rho V^2$	friction coefficient
$C_{f,\,water} = [1.58 \times \ln(Re) - 3.28]^{-2}$	friction coefficient for turbulent newtonian flow (Filomenko)
D	pipe diameter (m)
$DR = [1 - C_f/C_{f,\,water}] \times 100$	drag reduction level (%)
$Gr = g\beta D^3 \Delta T_{w,\,avg\text{-}b}/\nu^2$	Grashof number based on D
$h = q''/\Delta T_{w\text{-}b}$	convective heat transfer coefficient (W/m^2 K)
k_f	thermal heat conductivity (W/m K)
$Nu = q''D/\Delta T_{w\text{-}b} k_f$	Nusselt number based on D
$Nu_{avg} = q''D/\Delta T_{w,\,avg\text{-}b} k_f$	average Nusselt number (based on the average temperature of the three wall sensors)
$Pr = \nu/\alpha$	Prandtl number
$Pr_t = \varepsilon_M/\varepsilon_H$	turbulent Prandtl number
$Ra = g\beta\Delta T^4_{w,\,avg\text{-}b}/\nu\alpha$	Rayleigh number
$Re = VD/\nu$	Reynolds number based on D
q''	heat flux at the wall (W/m^2)
$T_b = T_i + q''/\rho C_p V$	bulk temperature (°C)
$T^+ = (T_w - T)u^*\rho C_p/q''$	dimensionless wall-to-fluid temperature difference
$u^* = (\tau_w/\rho)^{0.5}$	friction or shear velocity
V	bulk velocity (m/s)
$y^+ = yu^*/\nu$	dimensionless distance from the wall

Greek symbols

α	thermal diffusivity (m^2/s)
β	thermal expansion coefficient (1/K)
$\Delta T_{w\text{-}b}$	inner wall-bulk temperature difference (K)
$\Delta T_{w,\,avg\text{-}b}$	average inner wall-bulk temperature difference for the three wall locations (K)
ε_M	momentum eddy diffusivity (m^2/s)
ε_H	heat eddy diffusivity (m^2/s)
ν	kinematic viscosity (m^2/s)
ρ	fluid density (kg/m^3)
τ_w	wall shear stress (N/m^2)

Subscripts

up	refers to top portion of the pipe
dn	refers to bottom portion of the pipe

inelastic ones at similar heat and flow rate conditions, being in some cases higher by 40%. Other researchers have studied, for viscoelastic fluids, the effect of buoyancy on laminar flows over vertical and horizontal plates, as well as around the stagnant region of a heated cylinder [6–8]. In all cases, there is an increase in the convective heat transfer coefficient with increasing fluid viscoelasticity. Regarding internal flow, Ref. [9] appears to be the only study that has addressed the problem of mixed convection of viscoelastic drag-reducing fluids on vertical pipe flow under turbulent conditions. For this particular case, the turbulence generation is affected by the redistribution of the shear stress across the pipe, which is in turn affected by the buoyancy-driven flow moving in the flow direction.

Little work has been done on the mixed convection of drag-reducing fluids in horizontal pipes, however, although one might guess that buoyancy can induce secondary flows perpendicular to the main free-stream direction, presumably similar to those seen in laminar flows. The only reference to the effects of buoyancy in channel flows of drag-reducing surfactant solutions that we know is the one by Kawaguchi et al. [10]. They measured temperature profiles in the cross section of a square channel heated at the bottom wall, and found a region of very high diffusivity in the viscous sublayer. They also measured an increase in Nusselt number (Nu) of 20% due to a twofold increase in the heat flux, with all other conditions remaining constant. The increased diffusivity and Nu were attributed to buoyancy effects, although they also considered the possibility of thermal destruction of the micelles. In our recent studies on the heat transfer and temperature profile measurements of various drag-reducing polymer and surfactant solutions, we have found that buoyancy effects are present in laminar as well as in turbulent flows. The aim of this paper is to draw attention to the effect of buoyancy in drag-reducing turbulent pipe flows, which may affect experimental results if there is a lack of awareness of its presence.

2. Experimental installation

The experimental setup consists of a stainless steel tube of 19.95 mm (20 mm nominal) inner diameter and 680 diameters length, a centrifugal pump or a pressurized tank used for fluid circulation, and various pressure taps installed along the pipe for drag reduction measurements (DR). A more detailed description of the setup is given elsewhere [11].

Two types of heat transfer measurements were conducted: overall heat transfer coefficients (h), and local temperature profiles across the pipe. A DC Joule heating source provided a good approximation of a constant and uniform heat flux condition. We used four temperature sensors: three sensors (miniature RTDs 10 $\times$ 2 mm, 100 Ω), which were cemented with RTD epoxy adhesive (Omega OB-101-2) on the outer wall at the top, side, and bottom of the pipe at 675 diameters downstream of the entrance to detect the possibility of circumferential asymmetry of h; and one temperature sensor, a shielded type RTD, which was inserted across the pipe at the inlet in order to measure the fluid inlet temperature (T_i). The local fluid bulk temperature (T_b) at the location of the three RTD temperature sensors is calculated through the inlet temperature, the flow rate, and the heat flux measurements. These four temperature sensors were connected to a precision multimeter (Keithley DMM/Scanner) for data acquisition.

Altogether, the uncertainty of our data for circumferentially-averaged Nu_{avg} for water is around $\pm$12–15%, which is indeed about the usual uncertainty for most of the Nu correlations available for Newtonian turbulent flows. For the case of the drag-reducing fluids, the uncertainty is reduced given the increase in the temperature differences (details of this analysis can be found in [12]). On the other hand, there is an additional error of up to 10%, due to variations in the radial heat flux, which is another consequence of buoyancy effects (as explained below).

The measurement of the temperature profiles across the main flow direction is a challenging task, but it provided us with much new information. For this purpose, we have built a temperature sensor [13] which is moved perpendicularly to the main flow stream. The sensor is a home-made type E thermocouple (Chromel-Constantan). Each lead is 0.003 in. (0.08 mm) thick, and the welded bead is of an approximately spherical shape with a mean diameter of about 0.007–0.008 in. (0.18 mm). This sensor is displaced across the pipe by an external mechanism that allows it to move in increments of 0.001 in. (0.025 mm). This profile temperature sensor was located at approximately the same axial location as the RTD temperature sensors, so that the values of h measured by the RTDs, and those calculated by integration of the temperature profile should be about the same for symmetric profiles.

3. Results and discussion

3.1. Laminar profiles

Fig. 1 shows the results of temperature profile measurements for a 1500 ppm polyacrylamide (Separan AP-273) solution in deionized water. The span of our device covers only approximately 8 mm, and the two halves of the profile had to be measured by turning the pipe 180°, leaving a small gap in data at the center. A theoretical no-buoyancy laminar profile is shown for comparison, and there are two effects of buoyancy which one can readily see: a distortion of the temperature profile (or circumferential variation of local convective heat transfer), and a change in the average Nu compared to the flow without buoyancy. The profile looks similar to that measured for a Newtonian fluid [1], and shows also the coldest point shifted towards the bottom wall, presumably due to the action of secondary flows generated by buoyancy, as in the case of laminar flow. However, the effect is larger than for Newtonian fluids at the same Re and Pr numbers. The Nu_{avg} calculated by averaging the three wall-to-bulk temperature differences ($\Delta T_{w,\ avg\text{-}b}$) corresponding to each of the RTD sensors, is increased by buoyancy by 53% over its theoretical value with no buoyancy effects ($Nu = 4.36$), whereas only a 12% difference was expected based on the correlations proposed by Morcos and Bergles [3] for horizontal laminar pipe flows of Newtonian fluids. This is also qualitatively consistent with previous observations for vertical flat plates [5]; and indirectly with the results for flows around various geometries [6–8], for which fluid elasticity enhances the buoyancy-generated heat transfer. Note also that because of the effects of buoyancy, the Nusselt number calculated with the top wall temperature measurement ($Nu_{up} = 8.2$) and that with the bottom one ($Nu_{dn} = 3.9$), differ by a factor of more than 2.

Circumferential variations of the convective heat transfer may also induce a circumferential heat flux in the wall, which in turn may cause variations in the outer wall temperature measurements around the pipe. In this case, the assumption of the constant radial heat flux may not be exactly valid, and the Nu also loses its

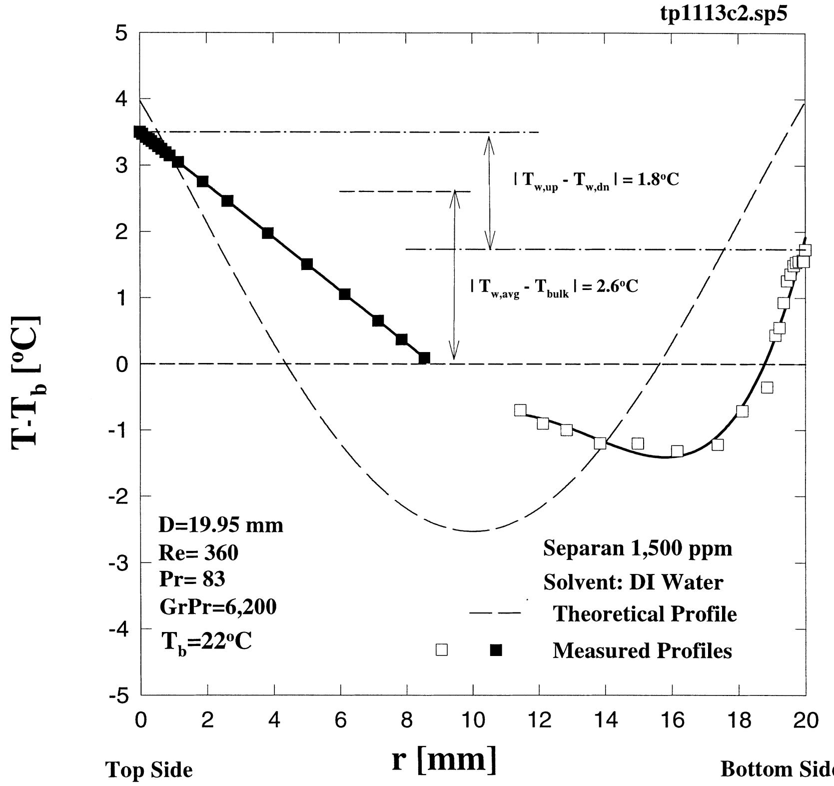

Fig. 1. Temperature profiles measured in the vertical plane of a laminar pipe flow of a drag-reducing fluid (water-based 1500 ppm polyacrylamide — Separan AP273 — solution). The top/bottom average temperature is 2.6°C higher than the bulk temperature.

strict meaning since the $\Delta T_{\text{w-b}}$ is no longer unique. However, the local *Nu* parameter is a good indicator of how how the local heat transfer varies around the pipe. According to our analysis based on the temperature differences between the top and bottom wall sensors (1.8°C for the case illustrated in Fig. 1), and on the pipe heat conductivity (k_p), it was estimated that the radial heat flux could vary by up to 10% between the top and bottom for average test conditions. Considering that under turbulent flow conditions (the regime of greatest interest for these fluids, see below) the buoyancy effects are relatively weaker because of increased forced convection mixing, it was concluded that it is acceptable to use the constant heat flux approximation in our analyses.

4. Turbulent profiles

During our measurements of temperature profiles in turbulent flow of *drag-reducing* fluids, we noticed that these profiles, as well as the outer wall temperatures used for the overall heat transfer coefficient calculations, were also significantly distorted by buoyancy effects, an effect that is generally considered to be negligible for turbulent flows of *Newtonian* fluids. Consequently, we had to determine under which conditions the buoyancy could affect significantly our results, and then determine under which range it was possible to conduct the experiments to avoid these effects. For that purpose, we carried out tests for various fluid concentrations, heat fluxes, and bulk velocities. Fig. 2 shows the results of wall-to-bulk temperature difference measurements for the three locations around the pipe: top, side, and bottom. These measurements were conducted for a 1500 ppm cationic surfactant solution (Ethoquad T13/27) plus 1300 ppm of NaSal as counterion, diluted in tap water. On the ordinate are shown each of the three $\Delta T_{\text{w-b}}$, normalized by $\Delta T_{\text{w, avg-b}}$; and on the abscissa, $Gr\ Pr/Re$, a combination of parameters which compares the effects of buoyancy ($Gr\ Pr$) to the flow intensity (Re). The Reynolds number is used in this correlation under the assumption that the flow is essentially turbulent, despite the fact that for this particular fluid the turbulence is highly damped since it showed asymptotic DR over the whole range of Re.

One can see at high $Gr\ Pr/Re$ (e.g. around 10) that if the *Nu* were calculated using the top wall temperature measurement (which is larger than the average wall temperature), it would be as much as 50% smaller than the *Nu* calculated from the average wall temperature, whereas the one calculated from the bottom wall temperature would be about 50% higher than the averaged one. On the other hand, and very usefully so, the sensor placed at mid-elevation shows approximately the average value between top and bottom throughout the whole range of $Gr\ Pr/Re$. As mentioned before, in the case of turbulent flows, the uniformity of the radial heat flux is less affected than it is for laminar ones (i.e. below 10%), and therefore, the ratio between the wall-to-bulk temperature differences measured at the top and bottom sections of the pipe, i.e. $\Delta T_{\mathrm{w,\ up\text{-}b}}/\Delta T_{\mathrm{w,\ dn\text{-}b}}$, can be considered, in first approximation inverse to the ratio of the h (or local *Nu*). Interestingly, the difference between $\Delta T_{\mathrm{w,\ up\text{-}b}}$ and $\Delta T_{\mathrm{w,\ dn\text{-}b}}$ at higher values of $Gr\ Pr/Re$ in turbulent flow is comparable to that measured in laminar flow (Fig. 1) which correspond to a $\Delta T_{\mathrm{w,\ up\text{-}b}}/\Delta T_{\mathrm{w,\ avg\text{-}b}} \approx 1.4$, and, $\Delta T_{\mathrm{w,\ dn\text{-}b}}/\Delta T_{\mathrm{w,\ avg\text{-}b}} \approx 0.7$, respectively, indicating a strong effect of buoyancy. For values of $Gr\ Pr/Re$ less than about 3, the effects of buoyancy do not seem to be important, and the scatter of the data around Nu_{avg} looks to be about the normal uncertainty for h measurements ($\pm 10\%$). Although not seen in this figure, for $Gr\ Pr/Re$ values less than 3 we have $\Delta T_{\mathrm{w,\ avg\text{-}b}} \leq 1.8°C$, a value one should, therefore, strive not to exceed in this case if the effects of buoyancy are to be avoided. One should keep in mind, however, that in the relationship shown in Fig. 2, the viscoelastic properties are not considered, and only one single fluid is used, which exhibit asymptotic drag and heat transfer reductions. Consequently, this criterion may have some limitations but it may well be valid for all asymptotic fluids.

To verify the effect of buoyancy on the temperature profiles, we measured a couple of temperature profiles under the conditions where we expected the effects of buoyancy to be negligible, and also where they would be more significant, i.e. at low and high heat flux conditions, respectively. Fig. 3 shows the effect of an increase in the *Gr* for a 400 ppm cationic surfactant solution (Ethoquad T13/27 by Akzo) diluted in tap water, which provides slightly less than asymptotic DR. Two temperature profiles measured at the bottom half of the pipe (where the effects of buoyancy are

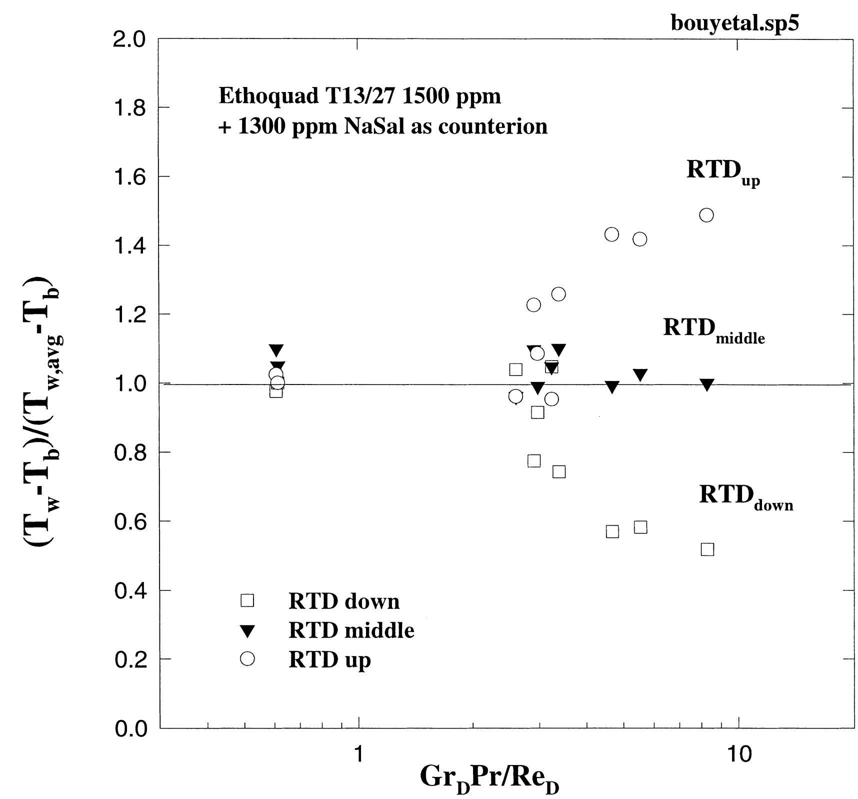

Fig. 2. Effect of buoyancy on the wall temperature measurements located at the top, side, and bottom of a circular pipe for turbulent pipe flow of a drag-reducing fluid (1500 ppm cationic surfactant solution — Ethoquad T13/27).

believed to be highest), are plotted in the usual dimensionless coordinates T^+ vs. y^+. For profile T10 (low heat flux: $q'' = 755$ W/m^2), the $\Delta T_{\mathrm{w,\,avg\text{-}b}}$ is calculated by integration of the profile to be about 1.6°C, and the corresponding $Gr\,Pr/Re$ to be about 3.1; for profile T11 (high heat flux: $q'' = 2100$ W/m^2), the $\Delta T_{\mathrm{w,\,avg\text{-}b}}$ is 5.2°C and the corresponding $Gr\,Pr/Re$ is about 8.1. For temperature profile T10, the difference between $Nu_{\mathrm{up}} = 15.1$ and $Nu_{\mathrm{dn}} = 15.5$ is less than 2%, whereas for T11, the difference between $Nu_{\mathrm{up}} = 12.0$ and $Nu_{\mathrm{dn}} = 13.2$ is already about 10%. For the lower heat flux, the temperature profile does not show a shift of the coldest point, which is usually a good indication of the top/bottom asymmetry of the temperature profile due to buoyancy. However, a slight shift of the coldest point towards the bottom wall is apparent at the higher heat flux. More importantly, even though the $\Delta T_{\mathrm{w,\,avg\text{-}b}}$ for the profile T11 is larger, when normalized by the wall heat flux, the region closer to the pipe center shows a somewhat smaller T^+, indication of the enhanced heat transfer due to stronger buoyancy (note that both profiles were measured along the bottom half of the pipe). Also, for the lower heat flux, the top and bottom wall RTD sensors did not show a significant temperature difference, whereas in the case of higher heat flux this difference was about 10% (T11 was intentionally measured at much larger heat flux than for the average tests).

The shift of the coldest point in Fig. 3 is not dramatic, and in order to see it more clearly, Fig. 4 shows a temperature profile measured at the bottom half of the pipe for the same fluid as in Fig. 2, with even higher heat flux than the one imposed for the experiments shown in Fig. 3. Although the level of DR is similar in both cases, the profile presented in Fig. 4 reflects a stronger distortion of the profile (shift of the coldest point towards the bottom half of the pipe) than the one shown in Fig. 3 (open symbols), as well as a larger difference between the $\Delta T_{\mathrm{w,\,up\text{-}b}}$ and $\Delta T_{\mathrm{w,\,dn\text{-}b}}$ measured. In this case the values of $Nu_{\mathrm{up}} = 10.5$ and $Nu_{\mathrm{dn}} = 26.5$ vary by a factor of 2.5, which corresponds to a wall temperature difference between the top and the bottom of almost 3°C. The fluid used in the tests presented in Fig. 4 (Ethoquad 1500 ppm) exhibits, however, significantly higher elasticity (normal stress differences) than the fluid in Fig. 3. It is

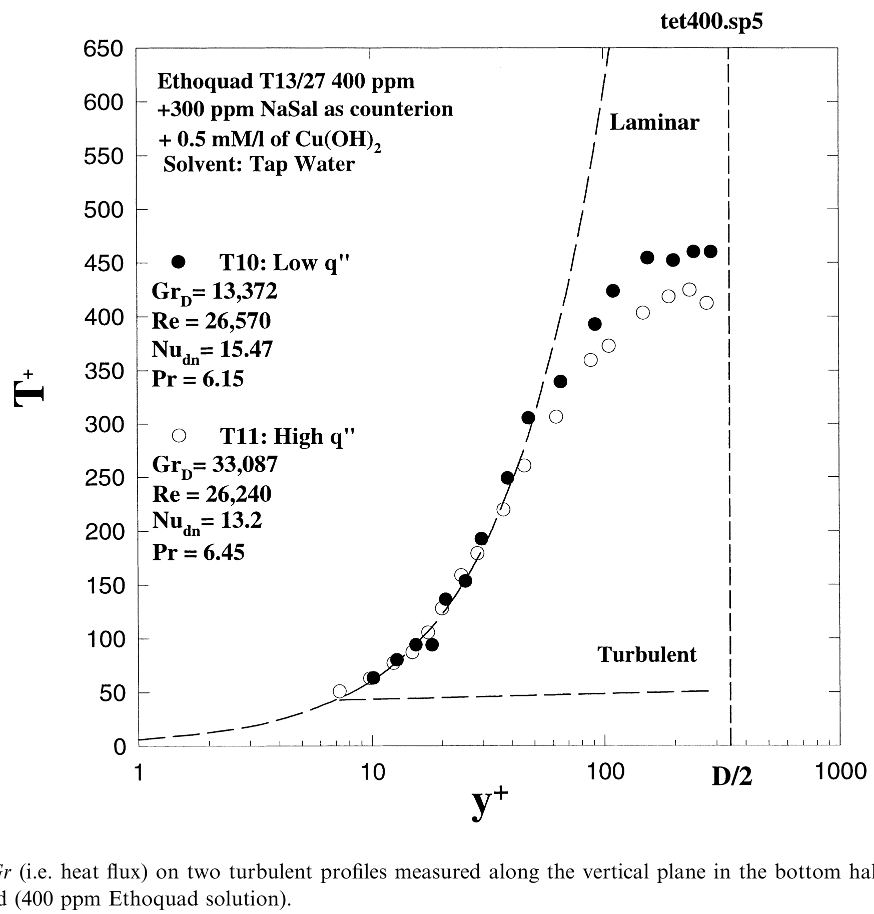

Fig. 3. Effect of Gr (i.e. heat flux) on two turbulent profiles measured along the vertical plane in the bottom half of the pipe for a drag-reducing fluid (400 ppm Ethoquad solution).

possible that higher elastic properties affect local turbulence, as is the case for turbulent flows in vertical pipes [9]. If so, this should be seen through direct measurements of the turbulent *Pr*, for which we have recently estimated an average value of about 5 for various drag-reducing fluids without buoyancy effects [13]. The reason for this particularly high value of the turbulent Prandtl number (Pr_t), may be related to an unexplained effect of elasticity on the correlation between temperature and velocity fluctuations. In this respect, buoyancy in turbulent flow of drag-reducing fluids deserves a deeper analysis that we cannot present in this short communication.

Note that in Fig. 4 there is a very strong shift of the point of coldest temperature towards the bottom wall, comparable to the one showed in Fig. 1 for the laminar flow. This similarity of the temperature profiles affected by buoyancy in the laminar and drag-reducing turbulent flows, may suggest that in drag-reducing flow we also have secondary flows, just as in laminar flow. This would not be surprising, because in the cases shown in Figs. 3 and 4, the Nu_{avg} is only three to four times higher than in the laminar flow despite the high *Re* associated to those flow conditions.

An important practical issue is worth noting here. Although the details of other experimental setups used for heat transfer measurements in turbulent flows of drag-reducing fluids are not known to us, it is likely that in many cases the sensors were located for convenience on the upper surface of the pipes (as we did ourselves initially), thus providing possibly greatly underestimated measurements of the *Nu*. To the best of our knowledge, very few researchers in this field have reported being aware of the problems that buoyancy effects could have on their heat transfer measurements and analyses. It is indeed very easy to take it for granted that the effects of buoyancy may be neglected for turbulent flows of drag-reducing solutions as well, as is indeed often done for Newtonian fluids. This would clearly be a serious mistake in some cases for drag-reducing fluids.

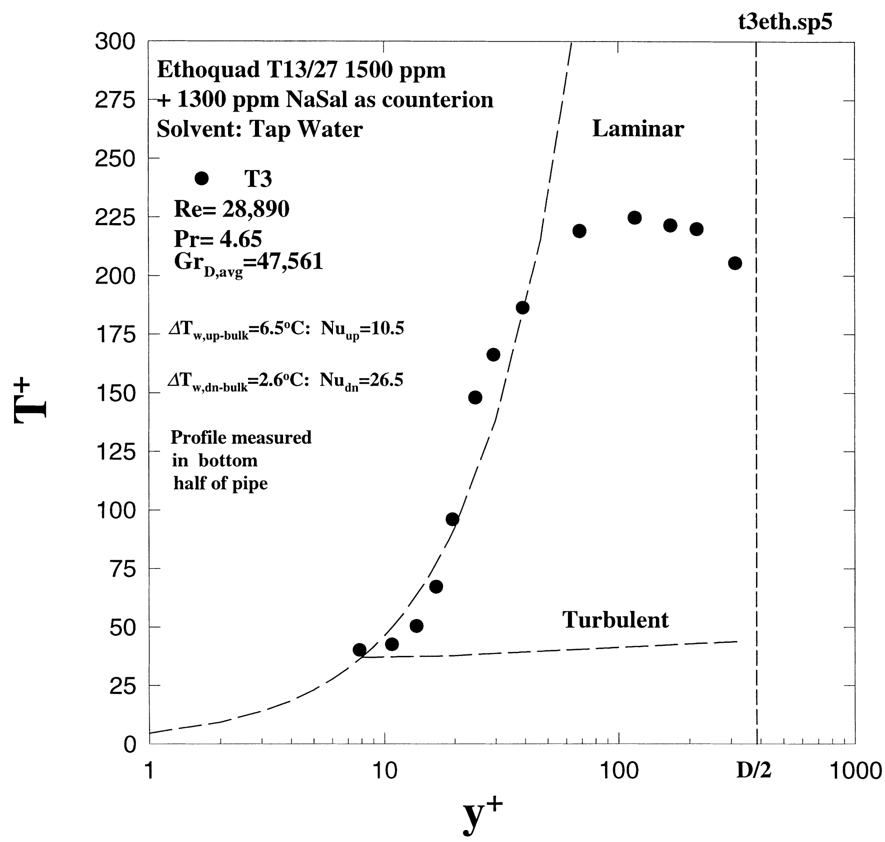

Fig. 4. Temperature profile profiles measured under high heat flux along the vertical plane in the bottom half of the pipe for a drag-reducing fluid (1500 ppm Ethoquad solution).

5. Summary and conclusions

The temperature profiles of drag-reducing fluids are significantly affected by the action of buoyancy under laminar flow conditions. These buoyancy effects are larger at the same *Gr Pr* than is generally observed with Newtonian fluids. In terms of *Nu*, buoyancy may increase the *Nu* for drag-reducing viscoelastic fluids to a level about 50% greater than that of the expected theoretical value with no buoyancy effects, whereas only a 12% difference is expected under the same conditions for Newtonian fluids.

Even in turbulent flows, the effects of buoyancy are still very noticeable for DR fluids. Given the distortion of the temperature profiles, it appears likely that the effect of buoyancy in turbulent flows of DR fluids is of the same nature as in laminar flows, i.e. caused by secondary flows. This is not surprising because even at relatively high *Re* (say up to 30,000), the Nu_{avg} are only three to four times higher than for laminar flows in the case of asymptotic DR, indicating very low turbulence. As a result of the asymmetry of the profiles, the usual heat transfer measurements in drag-reducing flows may show a large difference between the *Nu* calculated with the top and the bottom wall temperatures, in our case amounting to a ratio of over 2 between the two. It is therefore very important that experimentalists be aware of this issue, and use relatively low heat fluxes (in our case corresponding to $Gr\,Pr/Re$ of less than 3) to reduce the effects of buoyancy on the measurements. If that is not convenient or feasible, a temperature sensor placed on the side of the tube at mid-elevation should be used, since it will give a good approximation of the average *Nu* between top and bottom measurements.

Acknowledgements

We acknowledge gratefully the financial support of the California Energy Commission (contract No. 490260 to EFM). GA also wishes to acknowledge the Universidad Nacional Autonoma de Mexico, and especially the DGAPA and the IIM for support granted through their scholarship program.

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