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FLUID FLOW AND HEAT TRANSFER AT SUPERCRITICAL PRESSURE

J.D. Jackson
Emeritus Professor of The University of Manchester, Manchester, UK
jdjackson@manchester.ac.uk or J.DerekJackson@hotmail.com

Abstract

The feature of fluids at supercritical pressure which makes them of special interest is, continuous transition from a liquid-like to a gas-like state with increase of temperature at constant pressure accompanied by variation of properties over a particular band of temperature within which the specific heat reaches its peak value (the pseudocritical point). At pressures just above the critical value this band of temperature is very narrow, the peak is high and it is very sharp. As a consequence of the extreme dependence on temperature of fluid properties under such conditions the equations which govern fluid flow and heat transfer are very non-linear and strongly inter-linked. Thus, some of the simplifying concepts and assumptions which are widely employed in the case heat transfer to conventional fluids (such as fully developed flow and negligible influence of buoyancy and thermal expansion) no longer apply. Non-uniformity of density, can lead to important effects on the mean flow and turbulence fields and the effectiveness of heat transfer. When the author and his colleagues commenced work on supercritical pressure fluids almost fifty years ago they were very conscious of these challenges. Therefore, after carrying out a careful review of the literature they made the decision to begin with a novel experiment specifically designed to include effects of strong non-uniformity of fluid properties on heat transfer without involving the particular complications identified above. This very challenging experiment on stably-stratified turbulent flow of carbon dioxide at slightly supercritical pressure between two horizontal planes, with the upper one heated and the lower one cooled in such a way that there was no net heat transfer, yielded interesting results and some evidence of a special mechanism for enhancement of turbulent mixing. Non-dimensional representation of the governing equations and boundary conditions for flow and heat transfer in vertical tubes of fluids with highly temperature dependent properties enabled the appropriate dimensionless groups to be identified and the conditions for similarity to be rigorously specified. Experiments with uniformly heated vertical tubes using carbon dioxide at pressures very near to the critical value produced results, which in some cases exhibited very striking features. On increasing the imposed heat flux or reducing mass flow rate, severe localized non-uniformity of heat transfer developed in the case of upward flow. This was manifested by sharp, wall temperature peaks which were not observed with downward flow. These observations, provided clear evidence of gravitationally-induced motion leading to effects on heat transfer which could only be explained by postulating drastic modification of turbulence. This stimulated the development of a physically-based criterion for determining the conditions under which such effects might occur and led, eventually, to the development of a physically-based, semi-empirical model of buoyancy-influenced heat transfer with effects of thermal expansion. The aim of this lecture is to show how such early work is now providing a basis for correlating experimental data on heat transfer to supercritical pressure fluids and enabling the complicated phenomena encountered with such fluids to be better understood and properly accounted for in thermal design procedures.

Keywords: Supercritical pressure fluids, buoyancy-influenced flow and heat transfer, SCWR's

Introduction

A growing interest in heat transfer supercritical pressure fluids had been evident in the literature for some time prior to work commencing on that topic at Manchester University in the early nineteen sixties. The design of conventional thermal power plant operating at supercritical pressure had led to research and development on problems associated with the once-through steam generators used in such systems (Miropolsky and Shritsman, 1957, [1], Dickinson and Welch, 1958, [2]; Shitsman, 1959, [3]; Schmidt, 1959, [4]; Petukhov et al., 1961, [5]). Also, supercritical pressure water was under consideration as a coolant for nuclear reactors ((Goldman, 1954 and 1961, [8,9]). Hydrogen and oxygen were of interest in connection with rocket propulsion systems (Powell (1957) [6]; Hsu and Smith, 1961, [11]; Hendricks et al., 1962, [12]), and helium for the cooling of large electromagnets. Supercritical pressure carbon dioxide was mainly of interest to heat transfer experimentalists as a convenient working fluid for basic research (Bringer and Smith, 1957, [13]). The main motivation for the author's research on heat transfer to supercritical pressure fluids was to try to obtain a better understanding of the physical mechanisms by which non-uniformity of fluid properties might influence turbulent heat transfer. This interest in variable property forced convection stemmed from an earlier involvement (Barnes and Jackson, 1961, [14]) in work associated with the development of gas-cooled nuclear reactors designed to supply helium at very high temperature to gas turbines. The turbine blades needed to be maintained at acceptable temperatures by means cool helium flowing through small flow passages in them.

It is appropriate here to contrast the features of supercritical pressure fluids with those at subcritical pressure. The principal difference in the behaviour of a fluid as its temperature is raised above the critical temperature is clearly evident from the pressure-volume diagram shown on Figure 1. Below the critical temperature the variation of pressure and volume along an isotherm exhibits discontinuities where the isotherm intersects the saturation line. Phase change occurs at that line, and the horizontal, or constant pressure, segments represent the presence of two distinct phases in varying proportions. Above the critical temperature, no such discontinuous behaviour is found and from a macroscopic standpoint there is a continuous variation from a liquid-like condition to a gas-like one. However, at a molecular level, there is evidence from X-ray diffraction for the presence of a liquid-like structure even at supercritical temperatures and pressures, and the transition along a slightly supercritical isobar, such as A-A, is thought to be from liquid-like clusters of molecules in a matrix of gas, to a gas-like structure with a high degree of association between the individual molecules. At pressures which exceed the critical pressure by at least a few per cent, such inhomogenity of molecular structure is unlikely to have a significant effect on flow and heat transfer phenomena and a continuum view of fluids at supercritical pressure should suffice.

When heat is transferred between a surface and a moving fluid a thermal boundary layer is created, across which there is negligible variation of pressure. Therefore, it is appropriate, in the context of a study of convective heat transfer in a tube, to consider the manner in which the fluid properties vary with temperature at constant pressure. As mentioned above, at sub-critical pressure the thermodynamic and transport properties exhibit discontinuous changes when the saturation temperature is reached, i.e. a phase transition. At supercritical pressure this behaviour is replaced by continuous property variation concentrated in a relatively narrow band of temperature somewhat above the critical value – see Figure 2. In particular, the specific heat at constant pressure achieves a peak value in this region; the temperature at which this occurs is called the pseudocritical temperature, T_{pc}. This temperature increases as the pressure is increased, the maximum value of the specific heat falls, and the variation with temperature of all the other properties becomes less severe.

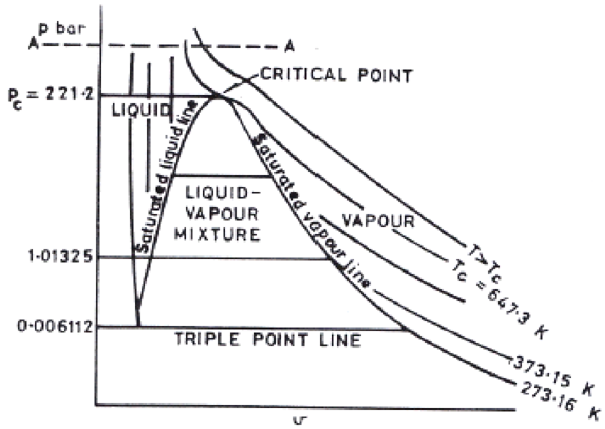


Figure 1 Pressure volume diagram for water

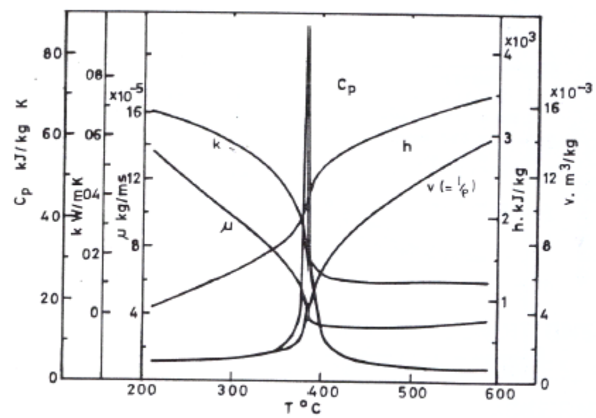


Figure 2 Property variation with temperature for water at 245 bar

1. Turbulent diffusion with very non-inform fluid properties

1.1 Experimental arrangement

Within a fluid at a pressure just above the critical value, heat transfer can be accompanied by considerable non-uniformity of properties and, as a consequence, a variety of influences can be involved. In designing an experiment to aid the understanding of variable property heat transfer, great care needs to be exercised to try to avoid complex interactions between such influences With this in mind an arrangement of the kind shown in Figure 3, below was used at Manchester by Khan, 1965, [15], and later by Woode, 1967, [16] and McFall, 1970, [17], in a fundamental study aimed at achieving fully developed fluid flow and heat transfer in a fluid having extremely temperature dependent fluid properties, carbon dioxide at a pressure of 75.8 bar ($p/p_c \approx 1.03$).

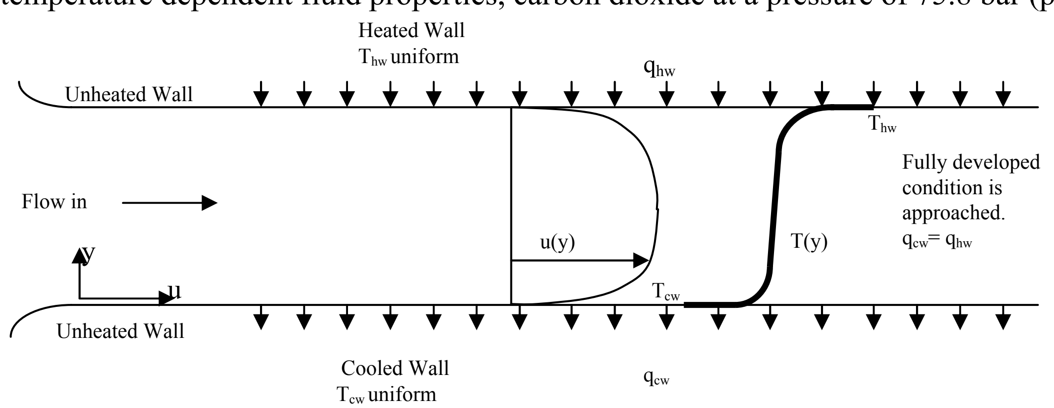


Figure 3 Arrangement for achieving fully developed flow and heat transfer with temperature dependent fluid properties

It flowed through a horizontal plane passage formed by two copper plates 0.2 m wide and 1.0 m long, spaced 10 mm apart. The upper wall could be heated and the lower one cooled. The aim was to achieve a condition of stably-stratified, fully developed turbulent flow and heat transfer in the downstream region of the passage, with the heat input from the upper surface being balanced by the heat removal from the lower one and the temperature of the fluid remaining constant in the flow direction. Thus, there would not be any advection of heat; it would simply be diffused across the flow by molecular and turbulent action. Furthermore, by adjusting the heating and cooling of the walls, the temperatures of the upper and lower surfaces could be controlled in relation to the pseudocritical value. By arranging that the hot and cold wall temperatures spanned the pseudocritical temperature and controlling their values in relation to it, the location of the region within the flow where very strong non-uniformity of fluid properties occurred could be varied in a systematic manner. This feature, together with that of achieving a condition of fully developed fluid flow and heat transfer, provided a unique opportunity of studying the effects of non-uniformity of fluid properties on heat transfer within a turbulent shear flow. It enabled attention to be focused on diffusion of heat in a situation where large fluctuations of density might occur in the course of turbulent mixing.

1.2 The flow loop

Fluid from the test section passed through an after cooler, so that near-critical conditions with attendant large density changes were avoided in the flow metering section. It then passed to a centrifugal pump, and returned to the inlet of the test section via a control valve, and a pre-heater. Pressure variations around the flow circuit were only a few centimetres of water, so that the physical properties only changed by a small amount along the test section on that account.

1.3 Experimental procedure

The procedure adopted in an experiment was to select a pressure and a flow rate, and then to cool the lower plate so as to maintain it at a fixed temperature below the pseuodocritical value. The temperature of the upper plate was increased in suitable steps from a value below the pseudo critical to a value well above it. It was necessary at each stage to adjust the pre-heaters to obtain an inlet fluid temperature that would enable the desired fully developed flow condition to be achieved in the downstream region. In all the experiments, the upper and lower wall temperatures became uniform in the direction of flow beyond of the mid point of the test section.

1.4 Experimental results

The overall heat transfer coefficient $q_w/(T_{hw}-T_{cw})$ was determined in each case and found to increase to a maximum when the pseudocritical temperature was achieved within the upper boundary layer. It then fell with further increase of T_{hw} providing evidence that a mechanism was present which enhanced turbulent heat transfer under such conditions (see Hall, Khan and Jackson, 1966, [18]). However, the experiment proved to be extremely challenging. It was not easy to decide whether the desired fully developed condition had been completely achieved. Furthermore, the interpretation of the results to quantify the enhancement of heat transfer as a result of increased turbulent mixing was difficult.

Fortunately, nearly fifty years later, developments in Computational Fluid Dynamics (CFD) using Direct Numerical Simulation (DNS) now make it possible now for such an experiment to be performed by completely numerical means, opening up the possibility of exposing the detailed fluid physics by which non-uniformity of fluid properties affects turbulence and heat transfer under such conditions and providing accurate quantitative measures of the effects. Hopefully this will be done in the near future and followed, eventually, by DNS studies of turbulent heat transfer to supercritical pressure fluids in heated tubes and also under more complicated conditions nearer to those encountered in practical applications.

2. Theoretical background on heat transfer to fluids at supercritical pressure in vertical heated tubes

2.1 Aims

The theoretical background presented in this section is aimed at identifying the dimensionless groups involved in heat transfer to fluids at supercritical pressure with a view to examining the requirements for similarity and scaling with such fluids.

2.2 The system and governing equations

The system under consideration is a vertical uniformly heated vertical tube through which fluid is drawn at a steady rate. At entry the fluid has a specified uniform velocity u_o and a specified uniform temperature T_o.

The governing equations for convective heat transfer to a fluid having temperature dependent properties are presented below in cylindrical co-ordinates with the x axis vertical. Attention is restricted to conditions which are axially symmetric.

$$\frac{D\rho}{Dt}+\rho\left(\frac{\partial u}{\partial x}+\frac{1}{r}\frac{\partial rv}{\partial r}\right)=0 \qquad \text{(Continuity)} \tag{1}$$

$$\rho\frac{Du}{Dt}=\pm\rho g-\frac{\partial p}{\partial x}+\frac{1}{r}\frac{\partial}{\partial r}\left[\mu r\left(\frac{\partial u}{\partial r}+\frac{\partial v}{\partial x}\right)\right]+\frac{\partial}{\partial x}\left[\mu\left\{2\frac{\partial u}{\partial x}-\frac{2}{3}\left(\frac{\partial u}{\partial x}+\frac{1}{r}\frac{\partial rv}{\partial r}\right)\right\}\right] \qquad \text{(Motion)} \tag{2}$$

$$\rho\frac{Dv}{Dt}=-\frac{\partial p}{\partial r}+\frac{\partial}{\partial r}\left[\mu\left\{2\frac{\partial v}{\partial r}-\frac{2}{3}\left(\frac{\partial u}{\partial x}+\frac{1}{r}\frac{\partial rv}{\partial r}\right)\right\}\right]+\frac{2\mu}{r}\left(\frac{\partial v}{\partial r}-\frac{v}{r}\right)+\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial r}\right)\right] \tag{3}$$

$$\rho c_p\frac{DT}{Dt}=\frac{\partial}{\partial x}\left[k\frac{\partial T}{\partial x}\right]+\frac{1}{r}\frac{\partial}{\partial r}\left[kr\frac{\partial T}{\partial r}\right]+\beta T\frac{Dp}{Dt}+\mu\phi \qquad \text{(Energy transfer)} \tag{4}$$

In Equation 2 the negative sign in the ± refers to an upward flow configuration and the positive sign to an inverted (downward flow) arrangement.

2.3 Non-dimensional representation of the problem

To identify the dimensionless groups which govern heat transfer under conditions where strong non-uniformities of fluid properties are present and to examine the requirements for similarity the problem is represented below in non-dimensional form. Normalised axial and radial coordinates X and R are defined as x/d and r/d, respectively and non-dimensional time η is defined as $\nu_o\, t/d^2$, in which ν_o is the kinematic viscosity at entry. Non-dimensional axial and radial components of velocity U and V are defined as ud/ν_o and vd/ν_o and normalized temperature θ is defined as T/T_o. Non-dimensional pressure P is defined as $pd^2/(\rho_o \nu_o^2)$. The governing equations represented in non-dimensional form are:-

$$\frac{D\rho^*}{D\eta}+\rho^*\left(\frac{\partial U}{\partial X}+\frac{1}{R}\frac{\partial(RV)}{\partial R}\right) = 0 \tag{5}$$

$$\rho^*\frac{DU}{D\eta} = \pm\left(\frac{gd^3}{\nu_0^2}\right)\rho^* - \frac{\partial P}{\partial X} + \left[\begin{array}{l}\frac{1}{R}\frac{\partial}{\partial R}\left\{\mu^* R\left(\frac{\partial U}{\partial R}+\frac{\partial V}{\partial X}\right)\right\} \\ +\frac{\partial}{\partial X}\left[\mu^*\left\{2\frac{\partial U}{\partial X}-\frac{2}{3}\left(\frac{\partial U}{\partial X}+\frac{1}{R}\frac{\partial(RV)}{\partial R}\right)\right\}\right]\end{array}\right] \tag{6}$$

$$\rho^*\frac{DV}{D\eta} = -\frac{\partial P}{\partial R} + \left[\begin{array}{l}\frac{\partial}{\partial R}\left\{\mu^*\left\{2\frac{\partial V}{\partial R}-\frac{2}{3}\left(\frac{\partial U}{\partial X}+\frac{1}{R}\frac{\partial(RV)}{\partial R}\right)\right\}\right\} \\ +\frac{2\mu^*}{R}\left(\frac{\partial V}{\partial R}-\frac{V}{R}\right)+\frac{\partial}{\partial X}\left\{\mu^*\left(\frac{\partial V}{\partial X}+\frac{\partial U}{\partial R}\right)\right\}\end{array}\right] \tag{7}$$

$$\begin{aligned}\rho^* c_p^*\frac{D\theta}{D\eta} = &\left(\frac{k_0}{\mu_0 c_{p_o}}\right)\left[\frac{\partial}{\partial X}\left(k^*\frac{\partial\theta}{\partial X}\right)+\frac{1}{R}\frac{\partial}{\partial R}\left(k^* R\frac{\partial\theta}{\partial R}\right)\right] \\ &+(\beta_o T_o)\left(\frac{\nu_o^2}{c_{p_o}T_o d^2}\right)\beta^*\theta\frac{DP}{D\eta}+\left(\frac{\nu_o^2}{c_{p_o}T_o d^2}\right)\mu^*\Phi\end{aligned} \tag{8}$$

Density, viscosity, specific heat, thermal conductivity and thermal expansion coefficient are all normalised using the values at entry; thus $\rho^*= \rho/\rho_o$, $\mu^*= \mu/\mu_o$, $c_p^*= c_p/c_{po}$, $k^*=k/k_o$ and $\beta^*= \beta/\beta_o$. Two dimensionless groups which only involve the properties of the fluid at entry appear in the equations, $\mu_o c_{po}/k_o$ (Prandtl number) and $\beta_o T_o$ (compressibility number). Dimensionless groups gd^3/υ_o^2 and $\upsilon_o^2/(c_{po}T_o d^2)$ which characterise gravitational and viscous dissipation effects, respectively and involve the properties of the fluid at entry and also the dimension of the system (tube diameter), also appear in the equations.

2.4 Inlet conditions and thermal boundary conditions

The condition of the fluid entering the system depends on the choice of working fluid and the pressure and temperature at inlet. These can be represented in non-dimensional form in terms of the reduced values p_o/p_c and T_o/T_c, where p_c and T_c are the values of thermodynamic critical pressure and temperature. Once the fluid condition at entry is specified the fluid property groups $\mu_o c_{po}/k_o$ and $\beta_o T_o$ are also specified.

Expressed in non-dimensional form the fluid flow and thermal boundary conditions are:-

$$At \quad X=0 \quad for \quad 0\le R\le\frac{1}{2},\ U = \left(\frac{u_o d}{\nu_o}\right) \quad and \quad \theta=1$$
$$At \quad R=\frac{1}{2} \quad for \quad 0\le X\le L,\ U=V=0 \quad and \quad k^*\frac{\partial\theta}{\partial R}=\left(\frac{q_w}{k_o}\frac{d}{T_o}\right)$$

2.5 Dimensionless groups

Two important dimensionless groups appear in this representation of the boundary conditions. These are the Reynolds number $u_o d/v_o$ (which can be viewed as a dimensionless imposed mass flux at entry) and, the thermal loading number $q_w d/(k_o T_o)$, which can either be viewed as a dimensionless imposed wall heat flux, or as the ratio of a representative reference temperature difference $q_w d/k_o$ and the reference fluid temperature T_o.

On multiplying $q_w d/(k_o T_o)$ by $\beta_o T_o$ a dimensionless thermal expansion group, $q_w \beta_o d/k_o$ is obtained.

On dividing Reynolds number squared, $(u_o d/\nu_o)^2$, by the gravitational body force group gd^3/v_o^2 a dimensionless group, the Froude number, u_o^2/gd, is obtained.

Combining $q_w d/(k_o T_o)$ with gd^3/v_o^2 and $\beta_o T_o$, gives the dimensionless group Grashof number, $g\beta_o q_w d^4/(k_o v_o^2)$.

3. Early literature on heat transfer to fluids at supercritical pressure in tubes

3.1 A critical review of the early literature

During the nineteen-sixties there was an explosion of interest in heat transfer to supercritical fluids. From 1963 to 1967 an additional ten experimental investigations were reported and two further empirical correlation equations were proposed. It was decided that the time was right to conduct a critical review of the literature on the subject. Such a review was carried out during the period 1966-67 and published a year later (Hall, Jackson and Watson, 1968, [19]). That paper included a comparison of calculations made using the four of the five empirical correlation equations available in the literature at that time ([1], [5], [6] and [7]) when they were used with water at 254 bar (p/p_c=1.15); heat transfer coefficients were determined as a function of wall temperature for a constant value of bulk temperature of 360°C using the equations. Agreement in terms of the level of the heat transfer coefficient proved to be poor.

3.2 Conclusions and recommendations

Firstly, it was concluded that the empirical correlation equations were inadequate as a means of predicting heat transfer in the critical region. Secondly, it was noted that there existed sets of data for apparently similar conditions in which radically different types of behaviour were exhibited. This strongly suggested that some relevant parameters might not have been controlled. Thirdly, it was felt that the effect of buoyancy under conditions of forced flow did not appear to have received sufficient attention; it was noted, however, that Miropolsky, Picus and Shitsman, 1966, [20] had presented results showing a marked difference between wall temperatures on the upper and lower surfaces of a horizontal heated pipe, together with further data for upward flow in a vertical pipe which showed a sharp localized peak in wall temperature.

It was recommended that a series of carefully planned experiments should be undertaken with due attention being paid to the additional mechanisms that may affect the heat transfer process at pressures just above the critical value and, in particular, that more attention be given to the effects of buoyancy influence. It was felt that upward and downward flows with heating or cooling needed to be systematically studied and that buoyancy-influenced heat transfer in horizontal pipes needed more attention. The need for turbulence measurements to be made under carefully selected conditions with fluids at supercritical pressure was emphasized. Systematic experiments aimed at resolving questions concerning thermal development under non-uniform property conditions were identified as being needed. Finally, it was recommended that selected experiments be repeated using a variety of fluids, in view of the lack of any strict similarity for different fluids of transport property variation with temperature.

3.3 Other early literature reviews

The Hall, Jackson and Watson review, coincided with another early review. Petukhov (1968) [21], which provided valuable information about work in the USSR on heat transfer to fluids at supercritical pressure at that time. Soon afterwards, a further detailed review was prepared in the USA (Hendricks et al., 1970, [22]).

4. Experimental studies of buoyancy-influenced heat transfer to carbon dioxide

4.1 Initial investigation

In response to one of the main recommendations in the review of Hall, Jackson and Watson, a comprehensive experimental study of buoyancy-influenced heat transfer in vertical tubes to carbon dioxide at pressures near the critical value was undertaken at Manchester (by Evans Lutterodt, [23], see also Weinberg, [24]). The test facility was designed to operate at pressures just above and also just below, the critical value of 73.8 bar and to accommodate test sections of various diameters. The first one had an internal diameter of 19 mm. An unheated entry length of 64 diameters was followed by a heated length of 129 diameters. This was instrumented with over 160 chromel alumel thermocouples welded to the outer surface of the tube at axial locations approximately one tube diameter apart, in some cases in sets of three around the circumference. Perforated copper discs in a compartment at the test section outlet ensured that thorough mixing of the fluid took place there before its temperature was measured. All the thermocouples were connected through a scanner/switch unit to an ice 'cold junction' and the electrical signals were measured using a digital voltmeter with a resolution of 2.5 microvolts (equivalent to about 0.06 of a degree). Before entering the test section the carbon dioxide passed through an electrical pre-heater unit and after leaving the test section it passed to a shell and tube cooler, which removed all the heat which had been supplied in the pre-heater and the test section. The fluid then entered a standard orifice plate flow-metering section before being returned by a canned centrifugal circulating pump to the pre-heater. The test section could be readily inverted with minimal modification of the pipework. Thus, heat transfer could be studied with either upward or downward flow under otherwise identical conditions.

4.2 Experimental results

Selected results, from a paper by Jackson, Evans Lutterodt, Weinberg and Fewster [25], are presented here. Figures 4 to 7 illustrate the effects on wall temperature for upward and downward flow of varying the heat flux and mass flow rate at a pressure just above the critical value (p=75.8 bar, p/p_c~1.03). Very strong effects of buoyancy on heat transfer are evident in some of these results.

With upward flow, striking axial non-uniformities of wall temperature developed when the pseudo-critical value was exceeded, indicating severe localized impairment of heat transfer (see Figure 4). Then, the fluid in the near-wall region had a much lower density than that in the core region and very considerable changes of all the properties occurred where the temperature passed through the pseudo-critical value. Under such conditions the mean flow, turbulence and thermal fields were modified in such a way that the effectiveness of heat transfer was strongly impaired. There was considerable circumferential non-uniformity of wall temperature at some axial locations (the vertical lines shown on the wall temperature distributions provide an indication of the extent of this). On reducing the flow rate, the peaks of wall temperature were smaller (Figure 5). However, the strength of the buoyancy influence, as manifested by the difference between wall temperature for upward and downward flow, was clearly greater. On reducing the flow rate further, localized wall temperature peaks developed even when the wall temperature did not exceed the pseudocritical value (see Figures 6 and 7).

With downward flow no such non-uniformity of wall temperature occurred in any of the experiments and the effectiveness of heat transfer was consistently better than with upward flow.

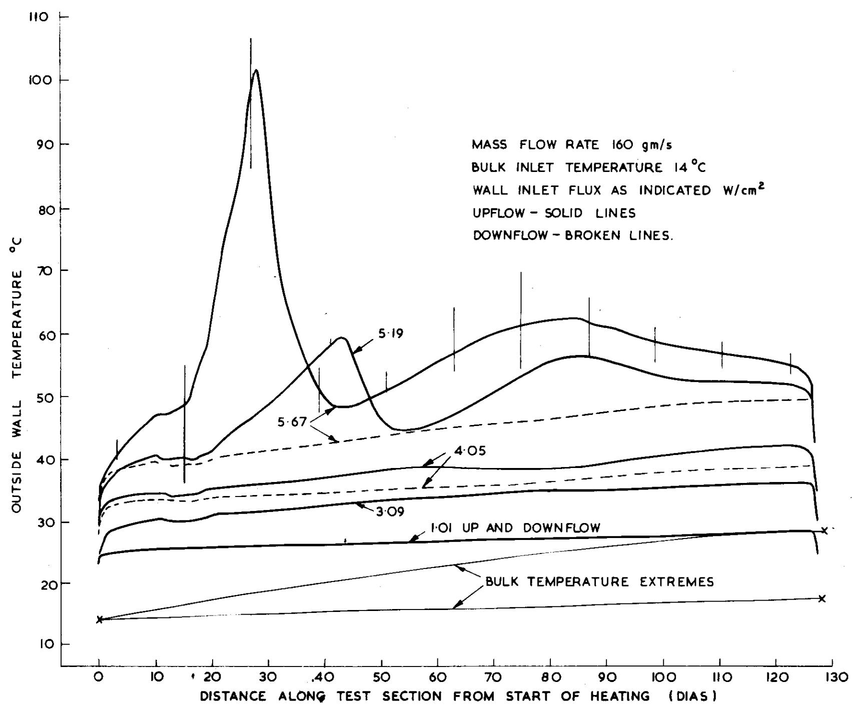


Figure 4 Effect of heat flux on wall temperature, upward and downward flow, 19 mm tube

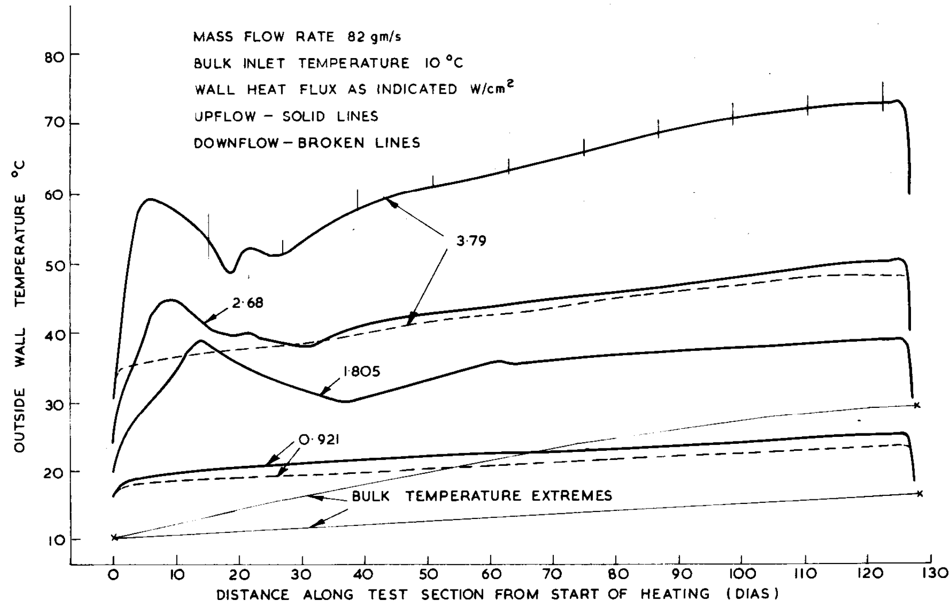


Figure 5 Effect of heat flux and on wall temperature, upward and downward flow, 19 mm tube

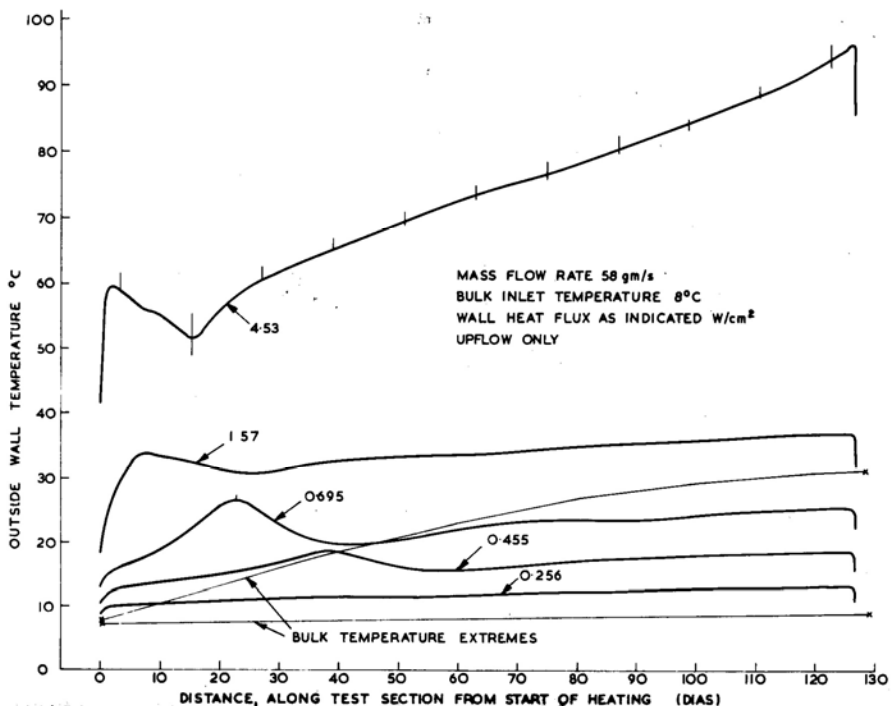


Figure 6 Effect of heat flux on wall temperature, upward and downward flow, 19 mm tube

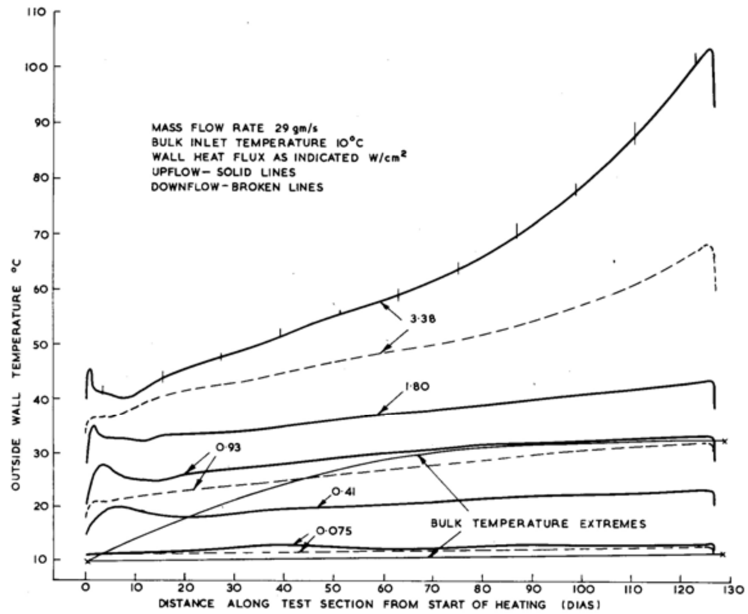


Figure 7 Effect of wall heat flux on wall temperature, upward and downward flow, 19 mm tube

4.3 An investigation of heat transfer at near-critical and high subcritical pressures

Further experimental results from Jackson, Evans Lutterodt, Weinberg and Fewster [25] are presented next. These highlight the fact that with upward flow, heat transfer to fluids at pressures just below the critical pressure can also be strongly affected by buoyancy and exhibit severe localized impairment of heat transfer very similar to that found just above the critical value. Figures 8 and 9 show the effects of heat flux on wall temperature with upward flow for two pressures just below the critical value (p/p_c 0.990 and 0.982). Three modes of heat transfer can be readily identified, namely, forced convection heat transfer to liquid carbon dioxide, subcooled nucleate boiling and gradual onset of film boiling. Striking, buoyancy-induced non-uniformities of wall temperature, which are very similar to those seen with upward flow at slightly supercritical pressures, are present in the film boiling region. It is clear that, over a limited range of pressures just below the critical value, heat transfer behaviour under conditions of buoyancy-influenced film boiling is very similar to that at pressures just above the critical value. An interesting point to note is that under such conditions the effectiveness of heat transfer is even lower just below the critical pressure than just above it.

On Figure 10, further results are presented from experiments performed with upward and downward flow at somewhat lower values of pressure, but still in the high subcritical range (70.5 bar down to 67.7 bar, p/p_c from 0.955 to 0.917). Under such conditions the transition which takes place from nucleate boiling to film boiling is quite different from that at the two higher pressures just considered. Instead of being gradual, departure from nucleate boiling is accompanied by what is almost a step increase of wall temperature to a condition of subcooled film boiling. As can be seen, the behaviour is similar for upward and downward flow but, in the latter case, for corresponding conditions of inlet temperature, flow rate and heat flux, the step increase of temperature is delayed and reduced in magnitude. Thus, the film boiling process is modified by buoyancy in such a way that the effectiveness of heat transfer in the post transition region is significantly improved for downward flow compared with upward flow.

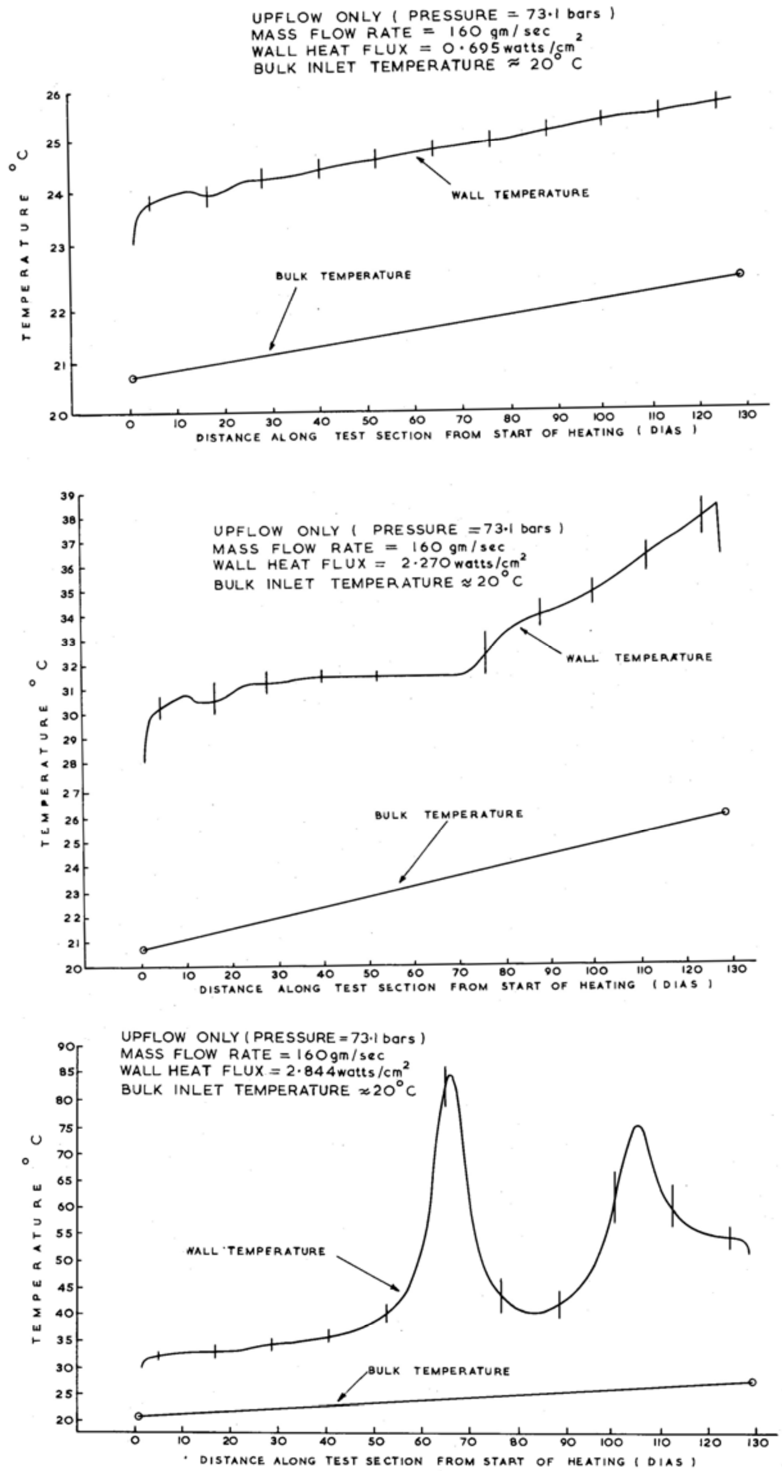


Figure 8 Effect on wall temperature of increasing heat flux
at a pressure just below the critical value (p/p_c ~0.990, upward flow)

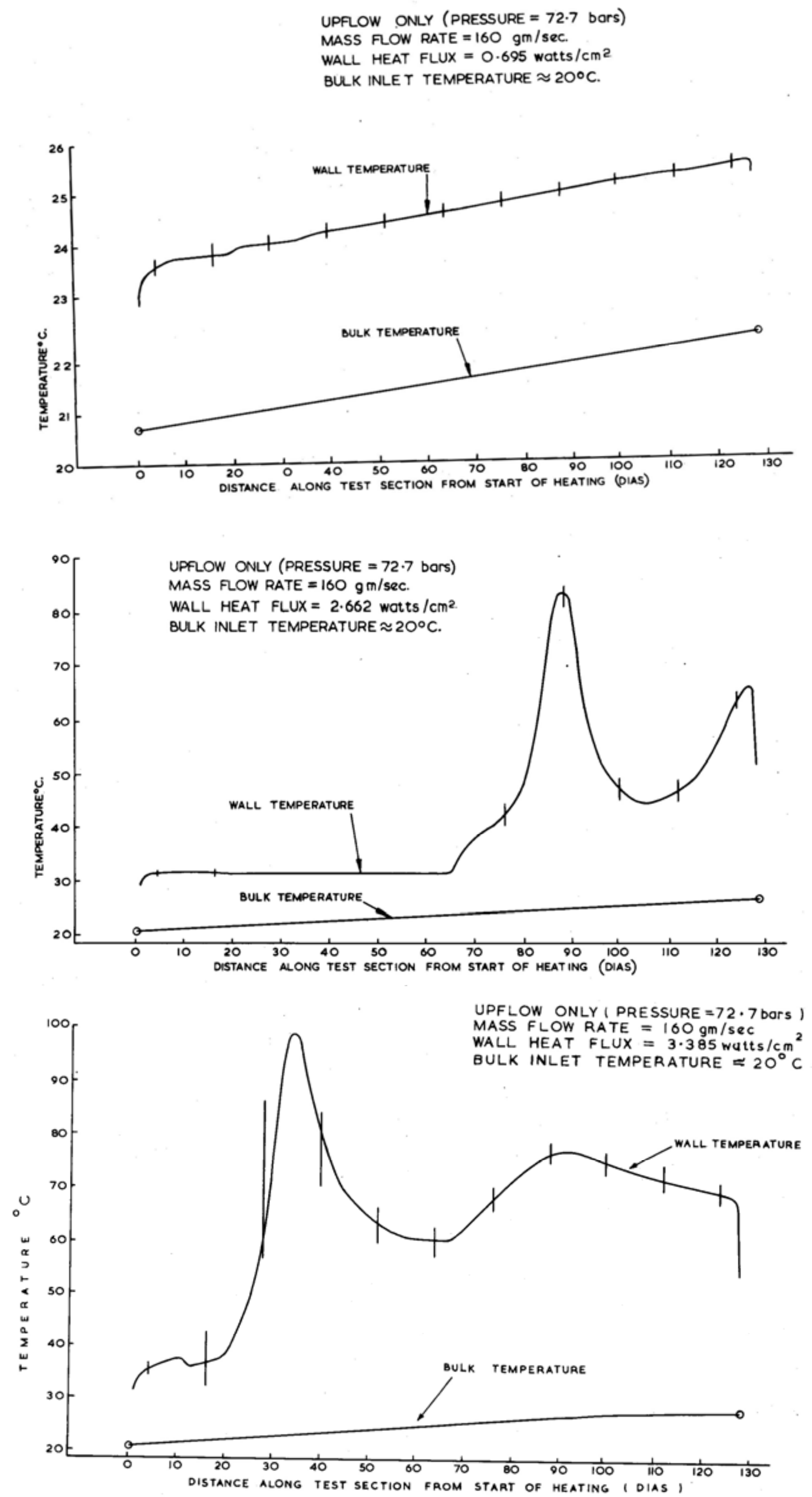
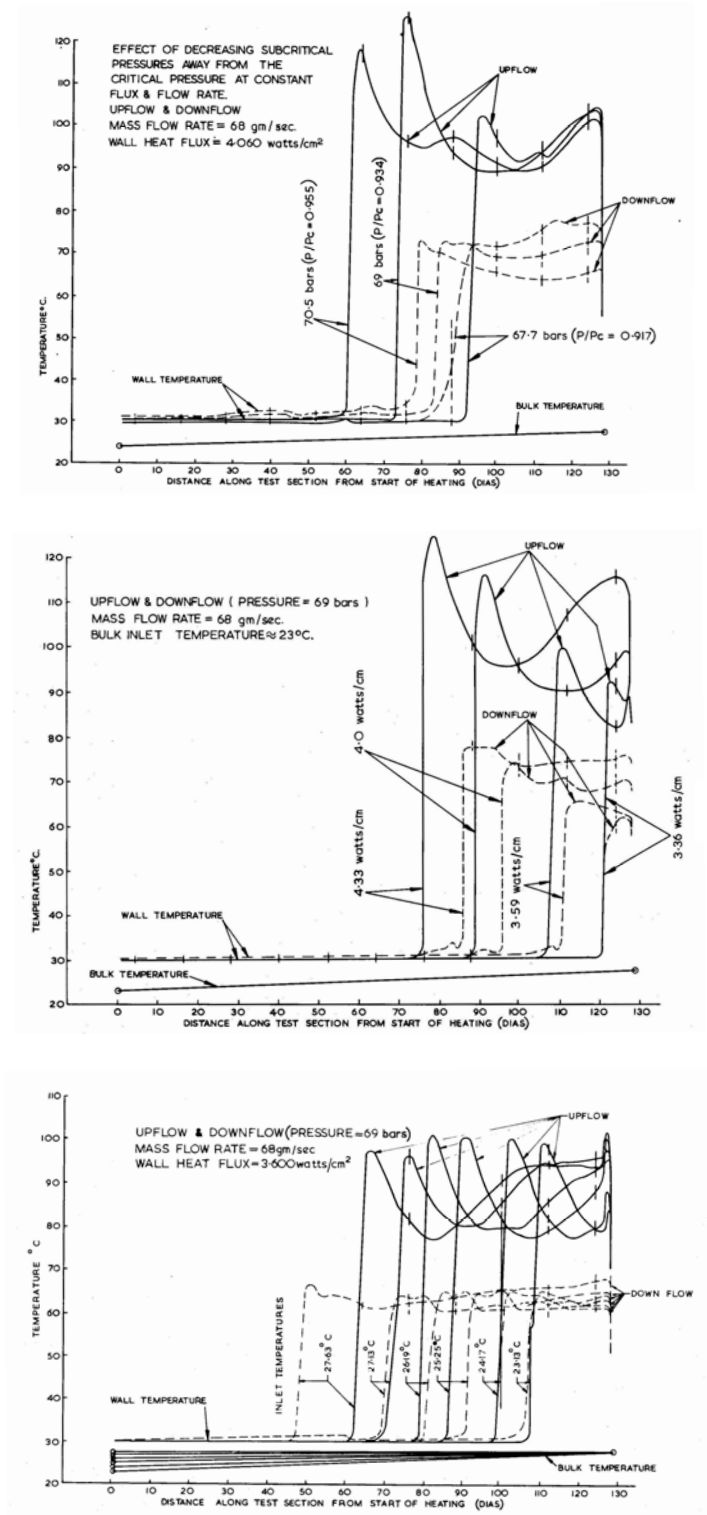


Figure 9 Effect on wall temperature of increasing heat flux at a pressure just below the critical value (p/p_c ~0.985, upward flow)



(a)

(b)

(c)

Figure 10 Effects on wall temperature of varying (a) pressure, (b) heat flux and (c) inlet fluid temperature at pressures well below the critical value (p/p_c ~0.955 to 0.917)

5. Laminarization of turbulent pipe flow by buoyancy forces

5.1 Localized deterioration of heat transfer

Even before the investigation of Evans Lutterodt at slightly supercritical pressure which has just been described, Shitsman, 1963 [26] had demonstrated the sensitivity to imposed heat flux of local heat transfer deteriorations with upward flow of a fluid at supercritical pressure in a vertical heated tube. He likened this to the deterioration found after transition from nucleate to film boiling at sub-critical pressures. However, it is difficult to accept such an analogy. The metastability of a bubble nucleus, and the consequent explosive growth of the bubble, seems to be an essential ingredient of boiling heat transfer. One may talk of drastic density changes in small 'bubbles' of supercritical pressure fluid, but in the absence of surface energy such 'bubbles' cannot grow independently of heat input as a vapour bubble can at subcritical pressure. It is also difficult to explain the rapid recovery in heat transfer downstream of the local deterioration of heat transfer on the basis of nucleate and film boiling; once 'burn out' or 'dry out' occurs in forced convection boiling it appears to dominate the whole of the downstream region.

It might be argued that the impairment of heat transfer could be caused by the reduction in thermal conductivity in the viscous sub-layer as the fluid in this region passes from a subcritical to a supercritical temperature. Calculations by Hall, Jackson and Watson, 1968, [19] suggested that this was not the case; it should be noted that such a reduction in conductivity will be accompanied by a similar reduction in viscosity, so that the thickness of the viscous layer decreases, thus offsetting the reduction in conductivity. It is also difficult to imagine how such an effect could account for the striking difference between the results for upward and for downward flow.

Explanations involving buoyancy were proposed by Shitsman (1968) [27], and Hall, Jackson and Watson (1968) [19]. Shitsman remarked, 'In upward water flow at moderate velocities in intensively heated tubes, the correlation and direction of the forces might be such as to damp down transverse turbulent velocity fluctuations in some part of the flow, resulting in the boundary layer becoming laminar'. Hall, Jackson, 1969a, [28] presented a model (to be described below) which depends on a reduction in the rate of production of turbulence, following a modification of the distribution of shear stress across the flow due to the action of buoyancy.

5.2 Effect of buoyancy in modifying the distribution of shear stress

The sharpness of the density change in passing from a temperature below the pseudocritical value to a temperature above it at a pressure slightly above the critical value allows one to use, with a fair degree of accuracy, a two region pattern of density variation in calculating the distribution of shear stress across a pipe. Let us assume that there is a layer of fluid at a uniform low density ρ_w adjacent to the wall, and that the core is at a uniform higher density ρ_b. If we neglect inertia effects in the wall layer region the modification to the shear stress distribution will be as shown diagrammatically in Figure 11 for three different thicknesses of wall layer. The particular thickness, δ, for which the shear stress at the edge of the layer (and throughout the core) is zero, is calculated below.

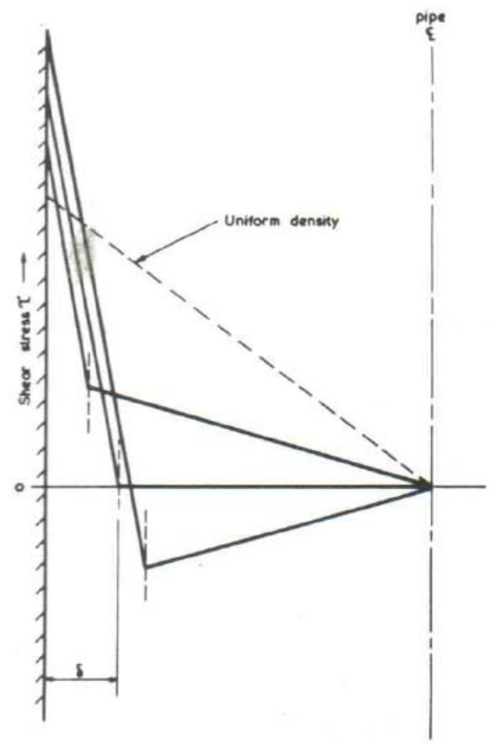


Figure 11 Shear stress distributions for three thicknesses of buoyant layer with upward flow

Typical values of ρ_w and ρ_b corresponding to the experiments with carbon dioxide presented earlier are 300 kg/cm^3 and 700 kg/cm^3 respectively (these values are appropriate to wall and bulk temperatures of about 35 and 29 degrees respectively). If the shear stress is to be reduced to zero at $y=\delta$, then the buoyancy force per unit length of pipe, must balance the wall shear force per unit length of pipe. The latter force increases slightly as the wall layer thickness increases, but the value for a uniform density equal to that in the core is still a good approximation when the shear stress in the core is reduced to zero. Equating an estimate of the wall shear force per unit length, in a pipe of diameter 19 mm at a Reynolds number of about 10^5, to the buoyancy force per unit length acting on the wall layer gives a value for δ of 6.4 μm in the CO_2 experiment under consideration. Thus, a very thin layer of low density fluid at the wall is capable of reducing the shear stress in the core to zero. This could have a profound effect on turbulent diffusion, as shown below.

5.3 Effect of the shear stress distribution on turbulent diffusion

The turbulence in the flow is maintained by an energy input which arises from the shearing of the turbulent fluid by the mean velocity gradient; this input is equal to the turbulent shear stress τ_T multiplied by this velocity gradient and has its greatest value close to the wall (not at the wall, because the flow there is non-turbulent, and $\tau_t = 0$). One might expect, therefore, that the modification to the shear stress shown in Figure 11 would reduce the level of turbulence production, and thus the turbulent diffusivity, ε. In the wall layer, the net effect will depend upon the relative changes in turbulent shear stress and velocity gradient and can be estimated by using a mixing length model of turbulent diffusion. This asserts that $\tau_t = \rho\overline{u'v'} = \rho\ell^2 \left|\frac{\partial \overline{u}}{\partial y}\right| \frac{\partial \overline{u}}{\partial y}$.

The turbulent diffusivity is given by $\varepsilon = \sqrt{\rho\ell^2 |\tau_t|}$. (9)

Thus, provided that the mixing length ℓ is not caused to increase significantly by the density variation, a reduction in total shear stress will involve a reduction in τ_t and an associated reduction in the turbulent diffusivity. As illustrated in Figure 11, the shear stress very close to the wall is increased by the presence of the low density layer. However, this increase will usually be in a region in which turbulence is damped out by the wall, and in which the turbulent shear stress is small. The increase will not, therefore, give rise to an increase in turbulence production.

It could be argued that the above analysis is open to objection on several grounds; firstly, it is by no means certain that the mixing length will be unaffected by the presence of the low density layer; secondly, and perhaps more fundamentally, the use of a mixing length model may be unjustified in these circumstances. Townsend [29] has shown that a description of the distribution of turbulence based on the turbulent kinetic energy equation (which balances the production, dissipation, convection and diffusion of the kinetic energy of turbulence) reduces to the mixing length model for an equilibrium boundary layer when the convection and diffusion terms in the equation are omitted; this implies local equilibrium between the production and dissipation of turbulence. In the present application it is likely that the turbulence level will change quite rapidly in the direction of flow, and that convection of turbulent kinetic energy will become important. Nevertheless, one can feel fairly sure that the basic mechanism proposed above is at least qualitatively correct.

5.4 A mechanism for local deterioration of heat transfer

The consequence of the events described in sub-sections 5.2 and 5.3 is that the turbulent diffusivity will be reduced in the case of upward flow when the low density wall layer becomes thick enough to reduce materially the shear stress in the region where energy is normally fed into the turbulence. This will, of course, reduce the diffusivity for heat, and therefore the heat transfer coefficient. As the process develops, along the tube, the wall temperature rises, the density difference becomes greater, and so does the buoyant layer thickness; both these effects accentuate the laminarisation. It is possible that the low density layer eventually becomes sufficiently thick for the sign of the shear stress to be changed in the central region, as shown in Figure 11. The wall layer will then exert an upward force on the core fluid; the production of turbulence will be restored, and the turbulent diffusivity will increase. The model therefore accounts not only for the observed deterioration of the heat transfer coefficient, but also for its subsequent improvement. It is interesting to note that when the shear stress falls to zero at the interface, the core is completely de-coupled from the wall, and is moving upwards as a true ‘plug flow’. The wall layer, on the other hand, is entirely motivated by buoyancy forces, which are acting in precisely the region where they are required to overcome viscous shear (rather than being transmitted by a shear process from the core – a process which normally maintains the production of turbulence). If the wall layer is so thin that turbulence in it is suppressed by the wall, then the whole flow could, if the situation persisted long enough, become non-turbulent.

An approximate criterion for the impairment of heat transfer by the proposed mechanism may be established in the following manner. Firstly it should be noted that for the effect to be significant the buoyancy forces must be capable of reducing the shear stress in the region where the turbulence production is a maximum; in other words, it must be possible to achieve the reduction with a thin layer of the low density fluid. The turbulence production is normally greatest at values of y^+ in the range from about 10 to about 30, so we might adopt as a criterion the existence of a wall layer of a thickness corresponding to y^+=30 which at the same time reduces the shear stress at the interface between it and the core to zero. For this situation to occur the buoyancy force per unit length must equal the wall shear force per unit length at $\delta\sqrt{\tau_w/\rho_w}/\nu_w = 30$. Thus,

$$30\pi g d \nu_w (\rho_b - \rho_w)(\tau_w/\rho_w)^{1/2} = \pi d \tau_w \tag{10}$$

If we use an empirical equation of the form f_b=0.046Re$^{-0.2}$ to describe the relationship between friction factor and Reynolds number than $\sqrt{\tau_w/\rho_u} = \bar{u}_b\sqrt{f_b/2} = \bar{u}_b\sqrt{0.023\,\mathrm{Re}^{-0.2}}$ (noting that because the wall layer is thin, the average mass velocity will be very close to $\rho_b\bar{u}_b$), then the following criterion for buoyancy-induced laminarisation of a turbulent flow is obtained

$$(Gr_b/Re_b^{2.7})(\mu_w/\mu_b)(\rho_w/\rho_b)^{-1/2} \approx 1.2\mathrm{x}10^{-4} \tag{11}$$

in which $Gr_b = gd^3(\rho_b - \rho_w)/(\rho_b \nu_w^2)$ and $Re = \bar{u}_b d/\nu_b$. If we insert the values ρ_b=700 kg/m^3, ρ_w=300 kg/m^3, d=19 mm, ν_w=ν_b=7.5x10^{-8} m^2/s and Re_b=10^5, then the buoyancy parameter Bo_b is about 7x10^{-5}, compared with the above value of 1.2x10^{-4}. Bearing in mind the rather approximate nature of the analysis and the various assumptions made, these two values are suitably close. Thus, the foregoing analysis enables the following conclusions to be drawn:-

The observed localised deterioration in heat transfer to supercritical pressure fluids flowing upwards in vertical pipes is the result of fluid buoyancy which causes the level of shear stress to be significantly reduced in the wall layer region leading to a diminution of the turbulence production component $(-\rho\overline{u'v'})\frac{\partial \bar{u}}{\partial y}$ and to impaired turbulent diffusion. As a result of this process a build up of wall temperature occurs along the tube and this accentuates the laminarisation process by increasing the buoyancy force acting on the low density wall layer. The process continues until at some stage turbulence production suddenly builds up again, probably due to the velocity profile being inverted and so distorted that the flow becomes unstable in the region where the fluid is moving faster than that in the core. The laminarisation process depends on the value of a buoyancy parameter $(Gr_b/Re_b^{2.7})(\mu_w/\mu_b)(\rho_w/\rho_b)^{-1/2}$ and will occur when this parameter has a value of about 10^{-4}.

6. An experimental investigation using smaller diameter tubes

A further experimental investigation using carbon dioxide at a pressure of 75.8 bar ($p/p_c \approx 1.03$), using test sections of 8 mm and 5 mm internal diameter, was carried out by Fewster (and reported in 1975, [30]). In each case an unheated entry length was followed by a long uniformly heated section which was instrumented with numerous chromel alumel thermocouples welded to the outer surface at axial locations approximately one tube diameter apart. Some sample experimental results from that study, taken from Fewster and Jackson [31], are presented here. Figures 12 to 14 show selected results obtained using the 8 mm bore test section. Effects of buoyancy still developed readily with this smaller diameter tube and are apparent (Figure 12) even at the highest flow covered. Clear differences can be seen between the wall temperature distributions for upward and downward flows over the entire length of the tube. On reducing the flow rate by a factor of five, very striking localized deterioration of heat transfer was found with upward flow but not with downward flow (see Figure 13). Note that the development of the peak coincided with the wall temperature exceeding the pseudocritical value and thus the presence of gas-like fluid in the near-wall region. The location of the wall temperature peak for upward flow could be readily varied by adjusting the inlet fluid temperature, (see Figure 14).

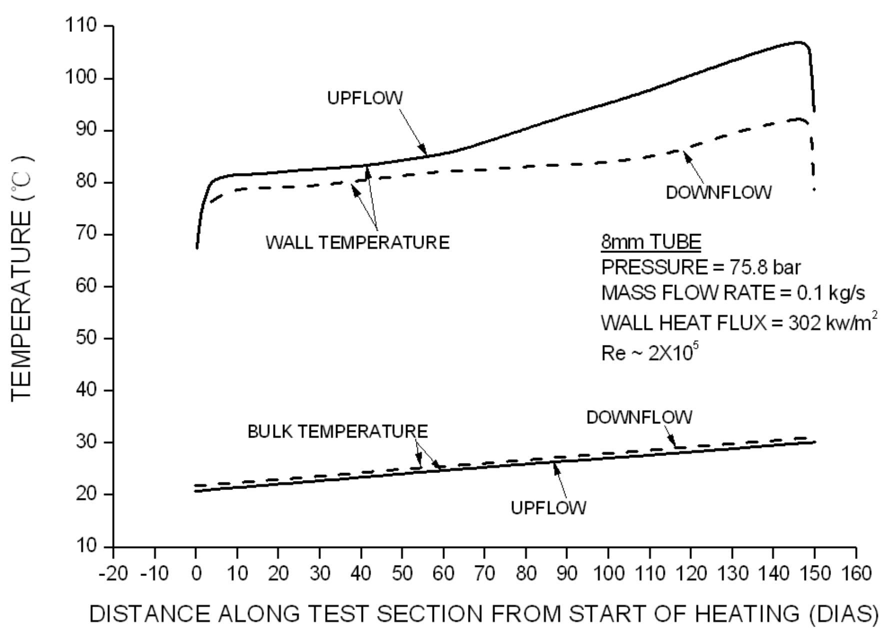


Figure 12 Highest mass flow rate and a high heat flux, 8 mm tube

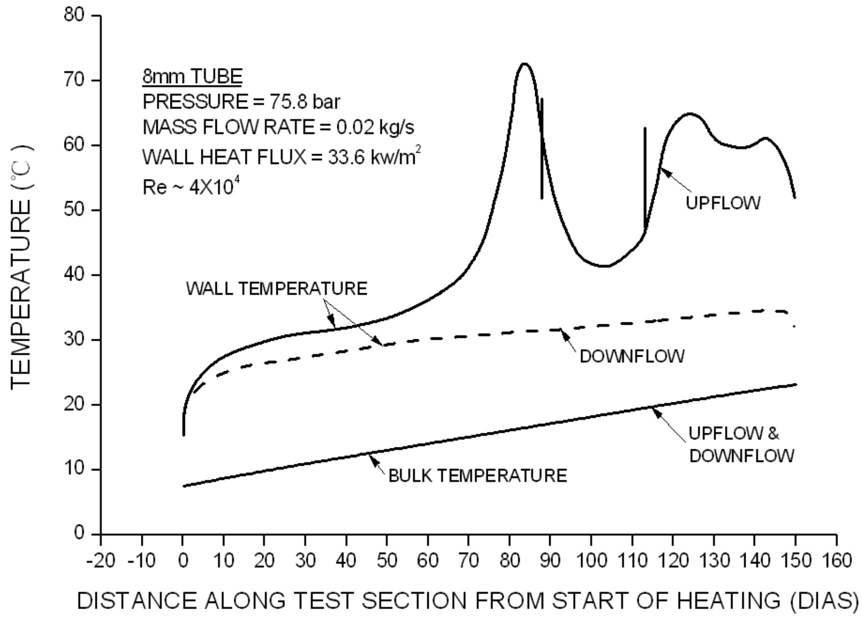


Figure 13 Low mass flow rate and high heat flux, 8 mm tube

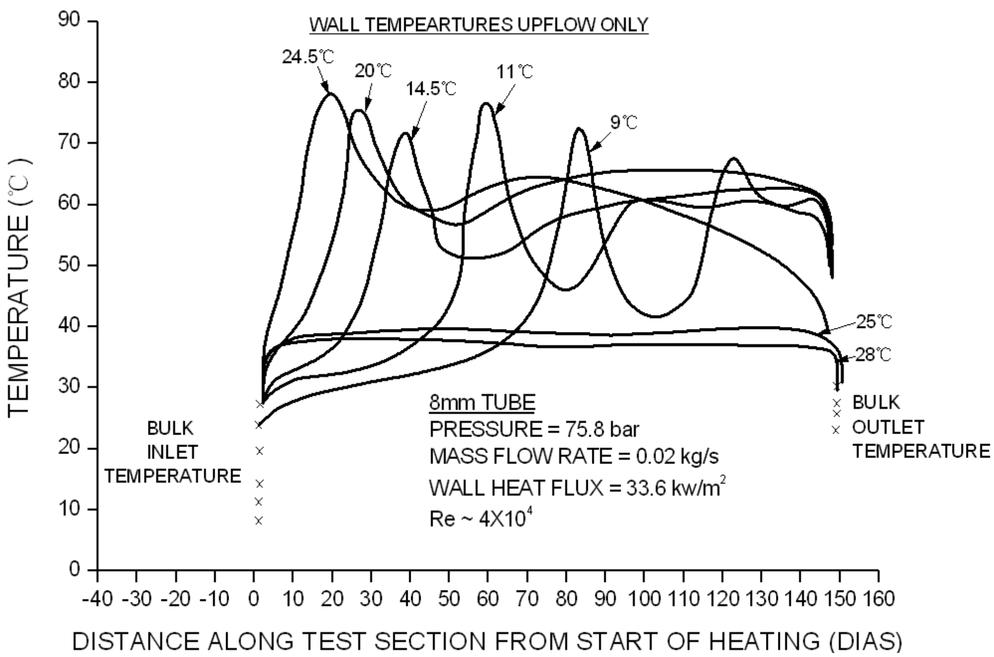


Figure 14 Effect of reducing inlet fluid temperature, 8 mm tube

Figures 15 to 19 show selected results obtained using the 5 mm bore test section. At the highest value of mass flow rate covered, the results are identical for upward and downward flow, even though the wall heat flux was high and caused the tube temperature to exceed 80^0C (Figure 15). Thus, in spite of the large variations of density within the flow there was no effect of buoyancy.

However, when the mass flow rate was halved some difference between the wall temperature distributions for upward and downward flow developed in the downstream region (Figure 16).

Further reduction in flow rate, caused a very sharp localized peak in wall temperature to develop in the upward flow experiment (Figure 17). This did not occur with downward flow. The onset and development of impaired heat transfer with increase of heat flux for upward flow is illustrated in Figure 18.

Interestingly when the heat flux was increased even further peaking of wall temperature no longer occurred and a maximum wall temperature of only about 80^0C was achieved (Figure 19).

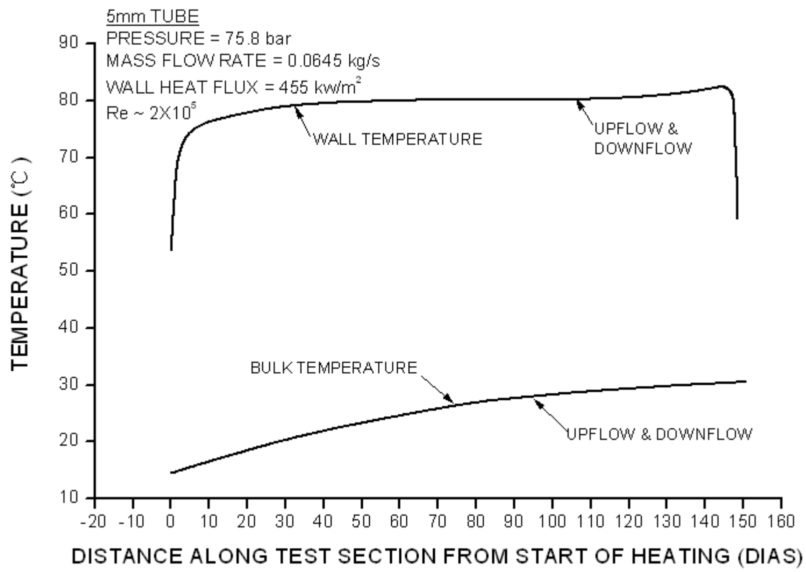


Figure 15 Highest mass flow rate, 5 mm tube

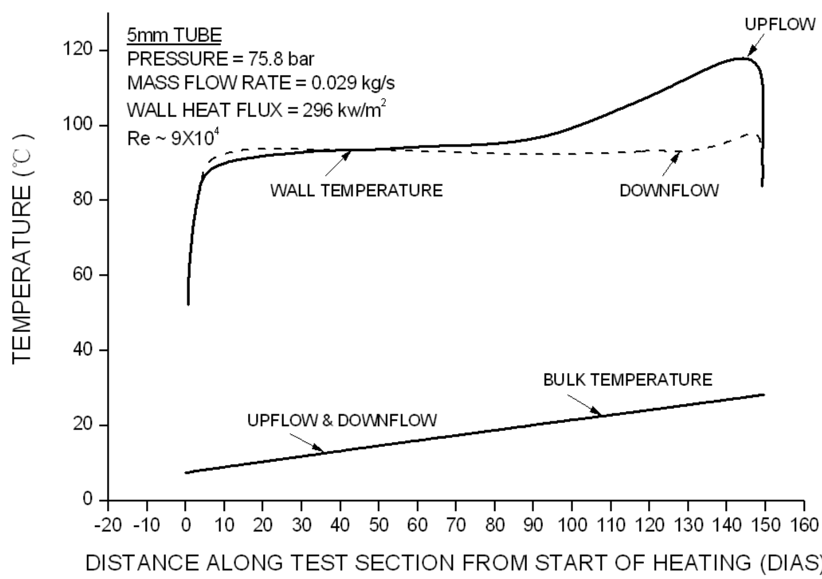


Figure 16 Reduced flow rate, 5 mm tube

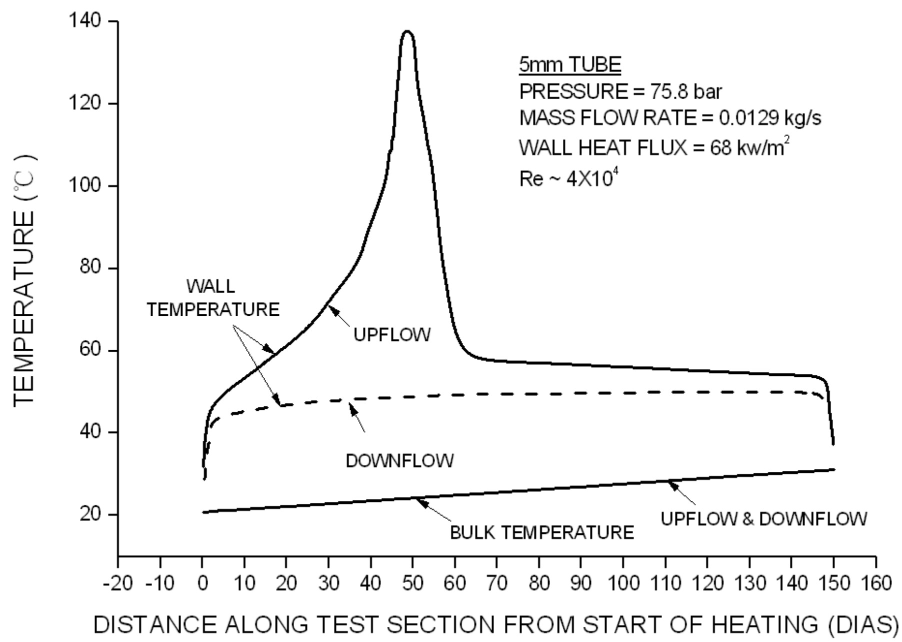


Figure 17 Further reduction of flow rate, 5 mm tube

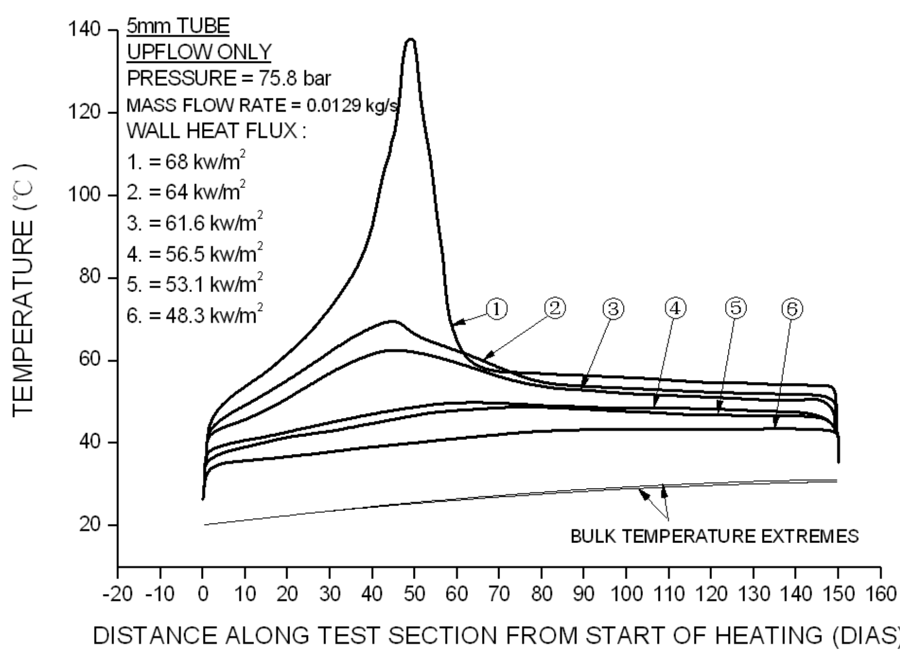


Figure 18 Development of impaired of heat transfer with increase of heat flux

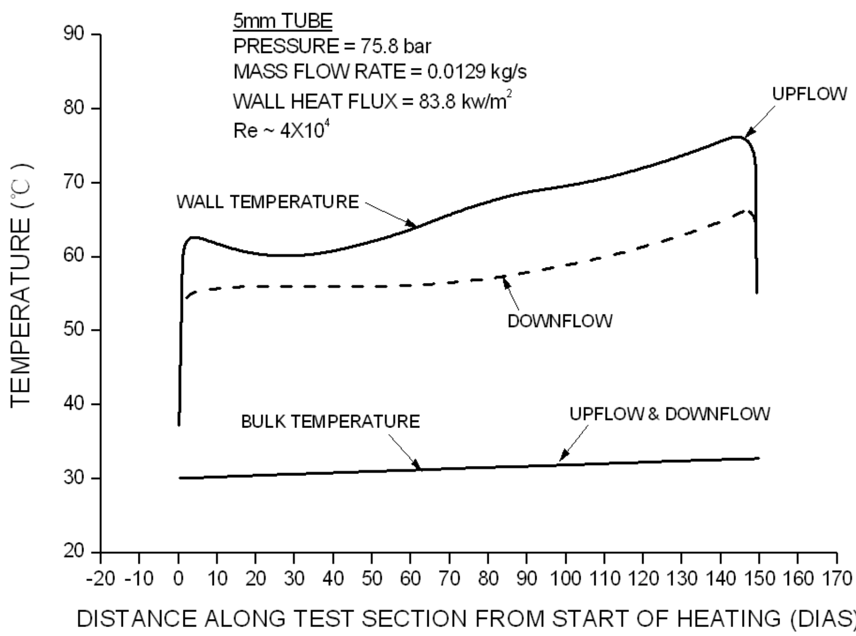


Figure 19 Further increase of heat flux, 5 mm tube

7. A comprehensive design report on heat transfer to supercritical pressure fluids

7.1 Background

The striking results obtained in the experiments reported by Evans Lutterodt, 1970, [21] (and later by Weinberg [24], Fewster [30]) and Adebiyi, 1973, [32] led in 1973 to the commissioning at Manchester by the United Kingdom Atomic Energy Authority of a detailed design report on 'Heat Transfer to Supercritical Pressure Fluids'. This lengthy document was completed two years later (Jackson et al., 1975, [33]). In it, the main emphasis was on data and design procedures for water, under the conditions of pressure and temperature encountered in supercritical pressure steam generators.

7.2 Coverage of the report and availability of the material

The report began with a description of the physical properties of fluids in the critical region and a review of the effects which the strong temperature dependence of fluid properties exerts on heat transfer. Buoyancy influence was shown to be of particular importance and criteria were proposed for determining whether the mode of heat transfer in a pipe for specified conditions of fluid inlet temperature, flow rate and wall heat flux was forced or mixed convection. Data and design procedures for forced convection were then considered. A critical evaluation of published empirical correlation equations available at that time was reported. A detailed review of data and correlations for mixed convection with upward flow in vertical tubes was then presented. This was followed by corresponding reviews of downward flow mixed convection and mixed convection in horizontal tubes. Brief surveys of work on the influences on heat transfer of tube curvature, internal ribbing, swirlers, non-uniform heating and pressure pulsations were then reported to complete the picture as far as available research data was concerned. The developments leading to the application of the supercritical pressure cycle for thermal power generation were described next and a review of the practical problems associated with supercritical pressure steam power plant was reported. The remainder of the document dealt with aspects of the thermal design of supercritical pressure steam generators.

Much of the basic research material and analysis contained in the report can be found in three papers published in the open literature soon after the completion of the design report (Hall and Jackson, 1978, [34] and Jackson and Hall, 1979a,b, [35,36]). Some of this will be summarized here.

7.3 An early attempt to correlate buoyancy-influenced heat transfer data

The buoyancy parameter $Gr_b / Re_b^{2.7}$ which came from the analysis of Hall and Jackson [28], was used by Fewster [30] to try to correlate hid data on heat transfer to carbon dioxide at supercritical pressure in vertical tubes and those of Evans Lutterodt. However, it is clear from Figure 20, that for upward flow such data do not correlate in terms of the purely local parameters Nu_b / Nu_{b_o} and $Gr_b / Re_b^{2.7}$. This is not really surprising because the analysis Hall and Jackson which yielded the parameter $Gr_b / Re_b^{2.7}$ involved a number of simplifying assumptions, one being that the net shear force could be equated to the buoyancy force as the velocity profile shape changed shape under the action of buoyancy, i.e. that inertia effects could be neglected. With upward flow, where buoyancy leads to reduced shear stress, this is a questionable assumption.

However, when the same approach was used in the case of downward flow, with attention being restricted to data for axial locations sufficiently downstream that a fully developed mixed convection condition was achieved (as can the readily deduced from the heat transfer behavior), an excellent correlation of the data was obtained, (Figure 21). In the downward flow case buoyancy increases the shear stress in the boundary layer and this leads to an increase of turbulence production with a fully developed condition being approached much more readily.

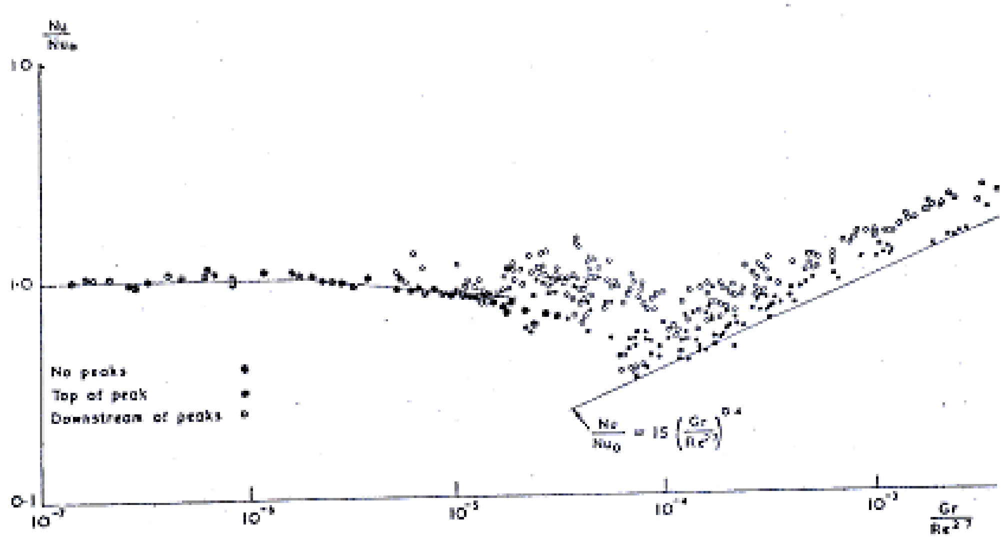


Figure 20 Mixed convection heat transfer to supercritical pressure CO_2 with upward flow

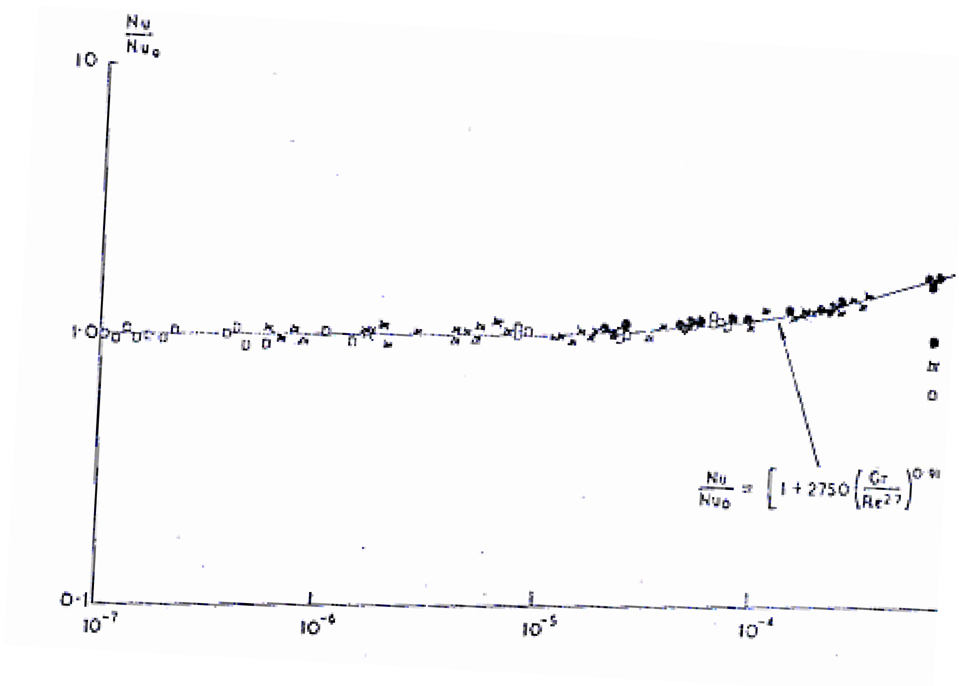


Figure 21 Correlation of data for fully developed downward flow of supercritical pressure CO_2

With downward flow wall temperature distributions are free from the localised effects which arise in the case of upward flow, the condition is inherently stable and enhancement of heat transfer develops systematically with increase in $Gr_b / \mathrm{Re}_b^{2.7}$. The correlated data for downward flow were fitted by the following simple empirical equation, the form of which was chosen so that the effectiveness of heat transfer became independent of flow rate under conditions of very strong influence of buoyancy.

$$\frac{Nu_b}{Nu_{b_o}} = \left[1 + 2750\left(\frac{Gr_b}{\mathrm{Re}_b^{2.7}}\right)^{0.91}\right]^{1/3} \tag{12}$$

7.4 Criteria for the onset of buoyancy effects

As described earlier in Section 6.3 an explanation of the mechanism by which buoyancy causes impairment of heat transfer with upward flow in a vertical pipe, was first given by Hall and Jackson in 1969 (later the matter was also addressed by Labuntsov, 1972, [37] and Tanaka et al., 1973, [38]). Hall and Jackson argued that buoyancy could modify the shear stress distribution in a turbulent boundary layer to the extent that turbulence production would be 'switched off' in a fluid at supercritical pressure when the wall temperature exceeded the pseudocritical value and a gas-like condition was achieved in a thin layer of fluid near the heated wall with rest being liquid like. Further analysis which led to more general criteria enabling the conditions for onset of significant effects of buoyancy to be determined in experiments with fluids at supercritical pressure was presented in the design report (Jackson et al. [33]) and later appeared in the open literature (Jackson and Hall [35]). The results are summarized below.

For conditions where the wall temperature does not exceed the pseudocritical value the analysis gave the following two (equivalent) criteria for the effect of buoyancy to be less 5%.

$$\frac{Gr_b}{\mathrm{Re}_b^{2.7}\,\mathrm{Pr}_b^{0.5}} < 10^{-5} \quad \text{or} \quad \frac{Gr_b}{\mathrm{Re}_b^{2.625}\,\mathrm{Pr}_b^{0.5}} < 2\times10^{-5} \tag{13}$$

For conditions where the wall temperature exceeds the pseudocritical value it gave

$$\frac{\mathrm{Gr_b}}{\mathrm{Re_b^{2.7}}} < 10^{-5} \quad \text{or} \quad \frac{\mathrm{Gr_b}}{\mathrm{Re_b^{2.625}}} < 2\mathrm{x}10^{-5} \tag{14}$$

Yamagata et al. [39,40], Lokshin et al. [41], Ackerman [42] and Belyakov et al. [43], have all reported attempts to relate the wall heat flux q_W at onset of impaired heat transfer, to mass velocity ω. Whilst there might be some basis for such a relationship, for conditions where T_w is less than T_{pc}, one would not expect the problem to be uniquely described by these two parameters where T_w and T_b span T_{pc}. For the latter case the density difference is only weakly dependent on T_w and T_b but the buoyant layer thickness is critically dependent on their levels in relation to T_{pc}. Thus in addition to varying with wall heat flux and mass velocity the magnitude of the buoyancy force can depend strongly on T_b. It is not surprising, therefore, that the various attempts to obtain relationships between wall heat flux q_w and mass velocity ω for onset of temperature peaks, have yielded rather inconsistent results. As indicted earlier, the onset of buoyancy induced impairment of heat transfer in upward flow is essentially an unstable process in the case of tubes which have an imposed heat flux. If the operating conditions are such that the onset criterion is approached with T_W just below T_{pc}, a small increase in wall heat flux may result in a considerable deterioration of heat transfer and locally high wall temperature.

7.5 Evaluation of empirical equations for forced convection heat transfer

The task of finding simple empirical equations capable of reliably describing the strong influences of property variation over the entire forced convection region was a formidable one. However, at the time that the design report was being produced some success had already been achieved in doing so using relatively simple extensions of established empirical correlation equations for forced convection heat transfer in tubes in the absence of significant non-uniformity of fluid properties. As part of the work on producing the design report sixteen empirical equations for fluids at supercritical pressure which had been published in the open literature were used to calculate heat transfer coefficients for approximately 2000 experimental conditions. About 75% of the data points were from experiments using water and the remainder were for carbon dioxide. An important feature of this exercise was that all the data were carefully screened to exclude any for which the influences of buoyancy might have been significant. This was done using the criteria presented earlier (Equation 5 and 6). The discrepancies between predicted and experimental heat transfer coefficients were tabulated for each case and performance statistics for the correlation equations were produced. Some of these are shown in Table 1. It can be seen that the correlation of Krasnoschekov and Protopopov, 1966, [44] is easily the most effective. A modified form of that correlation was also tried in which the constant properties part Nu_o (due to Petukhov and Kirillov, 1958, [45] was replaced by a simpler Dittus Boelter form similar to that recommended in Kays [46]. This version gave equally good results. It has the following form

$$\mathrm{Nu_b} = 0.022\ \mathrm{Re_b^{0.8}}\ \mathrm{Pr_b^{0.5}} \left(\frac{\rho_w}{\rho_b}\right)^{0.3} \left(\frac{\bar{c}_p}{c_{p_b}}\right)^{n_2} \tag{15}$$

The index n_2 is not constant. It depends on the values of T_w and T_b, in relation to T_{pc}, and is in the range 0.4 to 0.6 for most of the conditions for which data are available. The equations governing its variation are:-

$n_2 = 0.4$ for $T_b < T_w < T_{pc}$ and for $1.2\ T_{pc} < T_b < T_w$

$n_2 = 0.4 + 0.2\ ((T_w/T_{pc}) - 1)$ for $T_b < T_{pc} < T_w$

$n_2 = 0.4 + 0.2\ ((T_w/T_{pc}) - 1)\ (1 - 5\ ((T_b/T_{pc}) - 1))$ for $T_{pc} < T_b < 1.2\ T_{pc}$ and $T_b < T_w$

The equations of Bishop et al., 1965, [6] and Swenson et al., 1965, [7] also performed quite well for water but less well for carbon dioxide. A better compromise in terms of all round performance was obtained by Jackson and Fewster using a simplification of Equation 7 in which a mean value of n_2 of 0.5 was used giving

$$Nu_b = 0.022\ Re_b^{0.8}\ \overline{Pr}_b^{0.5} \left(\frac{\rho_w}{\rho_b}\right)^{0.3} \tag{16}$$

in which $\overline{Pr}_b = \mu_b \bar{c}_p / k_b$.

Table 1
Table showing the performance of some correlation equations for forced convection heat transfer to supercritical pressure water and carbon dioxide

Names of investigators	Limits	Water date %	Carbon dioxide data %
Krasnoschekov and Protopopov, 1966, [44]	± 15%	79	84
	± 20%	90	93
	± 25%	97	98
Version of above correlation with a Dittus Boelter form for Nu_o	± 15%	75	83
	± 20%	90	94
	± 25%	95	98
Jackson and Fewster (Equation 8)	± 15%	63	63
	± 20%	78	79
	± 25%	91	91
Bishop, Sandberg and Tong, 1965, [6]	± 15%	74	41
	± 20%	83	56
	± 25%	92	66
Swenson, Carver and Kakarala, 1965, [7]	± 15%	58	49
	± 20%	75	61
	± 25%	86	72
Miropolsky and Shitsman, 1957, [1]	± 15%	55	32
	± 20%	69	46
	± 25%	77	57
Miropolsky and Pikus, 1968, [47]	± 15%	38	33
	± 20%	53	43
	± 25%	67	51
Yamagata, Nishikawa, Hasegawa, Fujii and Yoshida, 1972, [40]	± 15%	34	12
	± 20%	44	17
	± 25%	54	21

A particularly satisfactory aspect of the performance of the Krasnoschekov Protopopov correlation equation and also Equations 15 and 16 is that they were equally good for both fluids, whereas most of the other equations were much less effective for carbon dioxide than for water.

Figure 22 illustrates the variation of wall temperature with bulk enthalpy for water at a pressure of 235 bar calculated using Equation 15. The parameter defining each curve is $q_w d/(\omega d)^{0.8}$. The deterioration of heat transfer which occurs as this parameter is increased is clearly apparent, however, there is no tendency for a peak in wall temperature to develop. In the latter respect the equation does not appear to be consistent with some experimental results for strongly heated flow in tubes of relatively small bore (e.g., Domin, 1963, [48] and Shiralkar and Griffith, 1969 and 1970, [49,50]) in which additional impairment of heat transfer was evident from peaking of the wall temperature over a wide range of bulk enthalpy somewhat below the pseudocritical value. This condition has been referred to as impaired forced convection because it is encountered with both upward and downward flow and thus is clearly not caused by buoyancy.

It is not really surprising that Equation 15 fails to reproduce the behaviour seen in the above-mentioned experiments with strongly heated tubes of small diameter because such data was not included in the forced convection dataset used in the evaluation exercise describe above.

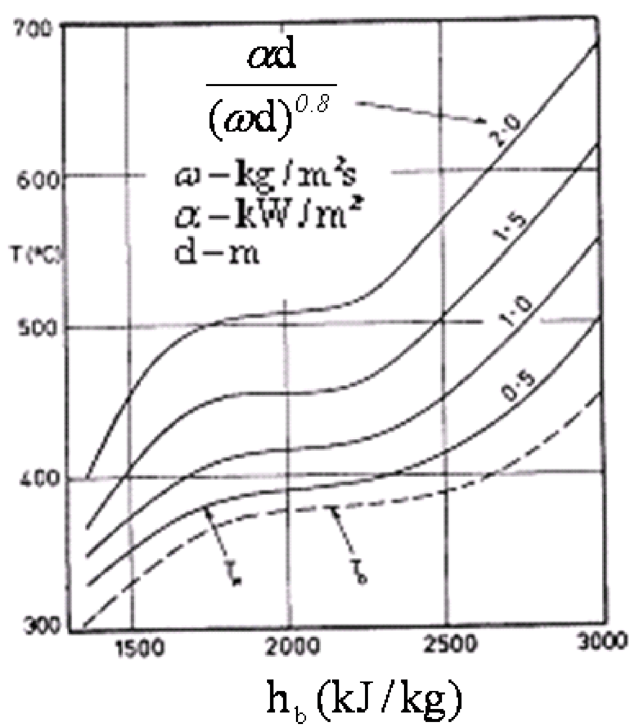


Figure 22 Wall temperature distributions calculated using Equation 7

7.6 Requirements for limited similarity

From the non-dimensional representation of the problem of heat transfer to fluids at supercritical pressure presented in Section 2 (which was originally published in Jackson and Hall in 1979 [35] and then later in an expanded form [51]), several statements can be made concerning the requirements for similarity between two geometrically similar thermal convection systems A and B, where the tube diameters, wall heat fluxes and inlet velocities may be different.

Firstly, if the same fluid is involved and both the reduced pressure p_o/p_c at inlet and reduced temperature T_o/T_c at inlet are the same for each system the two fluid property groups $\mu_o c_{po}/k_o$ and $\beta_o T_o$ will also be the same for each system.

Secondly, for the property ratios ρ^*, μ^*, k^*, c_p^*, and β^* to vary in a similar manner for each system, the pressure should not change significantly along the tube (then these ratios will each simply depend only on θ and the relationship with that variable will be the same for each system.

Thirdly, if the two conditions specified below are satisfied

$$(u_o d/\nu_o)_A = (u_o d/\nu_o)_B \qquad \text{(Reynolds number)} \qquad (17)$$

$$(q_w d/(k_o T_o))_A = (q_w d/(k_o T_o))_B \qquad \text{(Thermal loading number)} \qquad (18)$$

and the characteristic dimension of the system is small enough that no significant influence of buoyancy on the flow field is experienced but large enough that effects of viscous dissipation have a negligible influence on the thermal field, then strict similarity can be achieved in the case of forced convection to fluids at supercritical pressure. This limited possibility for strict similarity in the case of such fluids was first pointed out by Hall and Jackson, 1969b, [52] who indicated that measurements of wall temperature from experiments using tubes of different diameter should, coincide in cases where the above requirements are met, if plotted on a base of bulk temperature or enthalpy. It was demonstrated experimentally at Manchester using water at supercritical pressure in tubes of relatively small diameter (1.64 mm and 3.10 mm), see Harrison and Watson, 1976, [52].

7.7. The contrasting behaviour of buoyancy-free and buoyancy-influenced data

Where temperature distributions are compared on the above basis and found to differ, this points to the presence of significant influences of buoyancy, compressibility or dissipation. The comparison made by Hall and Jackson [53] between results which satisfy Equations 17 and 18, but which in some cases were clearly influenced by buoyancy (Figure 23), highlights the above point.

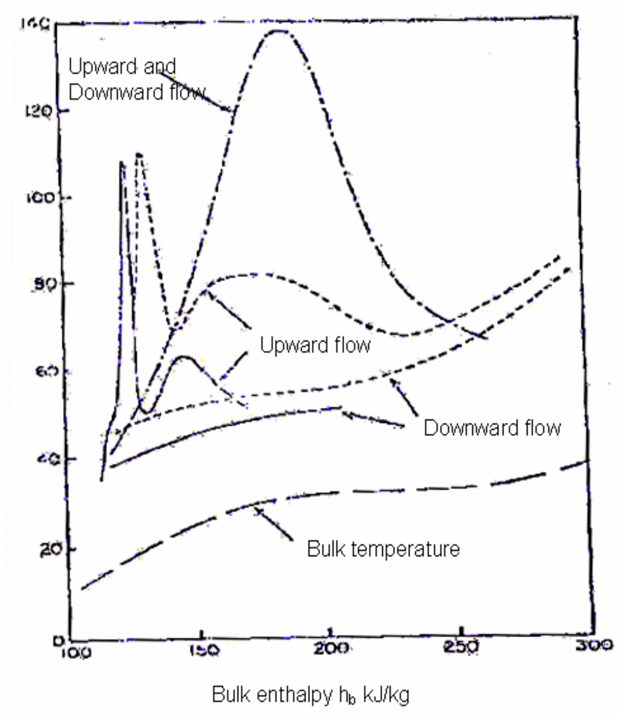


Figure 23 Experimental results compared on a forced convection basis

Note, the contrast between the broad temperature peak for the case where upward and downward flow are the same (data of Shiralkar and Griffith [50]) and the sharp localised peaks which are present in the buoyancy influenced data (upward and downward flow different – data obtained by Evans Lutterodt [21], see Hall and Jackson [53] and Bourke et al. [54]). Note that, downstream of the buoyancy-induced peaks the effectiveness of heat transfer is better than for the buoyancy-free case. Note also that for downward flow with significant influence of buoyancy there is substantially enhanced heat transfer and a monotonic variation of wall temperature.

We have seen that for 'fully developed' forced convection (buoyancy-free) heat transfer to a fluid at a specified value of pressure above the critical value the functional relationship between local Nusselt number, Reynolds number, Prandtl number and heat loading parameter is

$$\left(\frac{q_w d}{k_b(T_w - T_o)}\right) = \psi\left[\left(\frac{u_b d}{v_b}\right),\left(\frac{\mu_b c_{p_b}}{k_b}\right),\left(\frac{q_w d}{k_b T_b}\right)\right] \qquad (19)$$

The heat loading parameter $\left(q_w d / k_b T_b\right)$ determines the magnitude of the temperature variations within the fluid and, therefore, the extent to which non-uniformities of fluid properties are present. However, this parameter has not usually been used directly to account for property variations in the various approaches which have been developed to correlate forced convection heat transfer data for supercritical pressure fluids and fit such data using empirical equations. As we have seen, earlier variable property effects have usually been accounted for by adding carefully chosen functions of the ratios certain fluid properties evaluated at the wall and bulk temperatures to well-established empirical equations for constant property forced convection. However, such correction terms are themselves functions of wall to bulk temperature ratio and thus can be expressed in terms of heat loading parameter and Nusselt number. Thus, the empirical heat transfer equations formed using them are, in effect, expressions which do relate Nu_b to Re_b, Pr_b and $q_w d/(k_b T_b)$.

As we have seen, the requirements for strict similarity are limited to conditions of forced convection. However, the possibility of achieving approximate similarity in the case of experiments with two different fluids at supercritical pressure, where the variations of fluid properties with reduced pressure and temperature exhibit broadly similar behaviour, cannot be discounted but remains to be properly demonstrated experimentally. Similar features of heat transfer behaviour under conditions of forced convection and even mixed convection, to fluids at supercritical pressure are exhibited by different fluids. However, the general problem of similarity and scaling remains to rigorously addressed.

8. An experimental investigation of buoyancy-influenced heat transfer to supercritical pressure water in vertical tubes

The preparation of the review papers, Hall and Jackson, 1978, [34] and Jackson and Hall, 1979a,b, [35,36], produced after the completion of the design report, coincided with the construction of a large-scale test facility designed to operate at a pressure of 250 bar for studies of buoyancy-influenced heat transfer to water in vertical circular tube test sections of diameter typical of those used in the once-through steam generators (Watts, 1980, [55]). This test facility incorporated certain novel features, designed to keep its cost within the limits of available funding. The flow was driven round the loop by natural circulation, thus avoiding the need to purchase a very expensive pump, and was controlled by a magnetically activated value which was designed and manufactured 'in house'. The test facility yielded valuable heat transfer data, which was reported in Watts and Chou, 1984, [56]. A sample of such data is shown on Figure 24. The results from the study were in some respects complementary to those in the experimental studies using carbon dioxide of Evans Lutterodt, Weinberg and Fewster which were reviewed earlier but the pressure was significantly higher in relation to the critical value and representative of that in practical power plant.

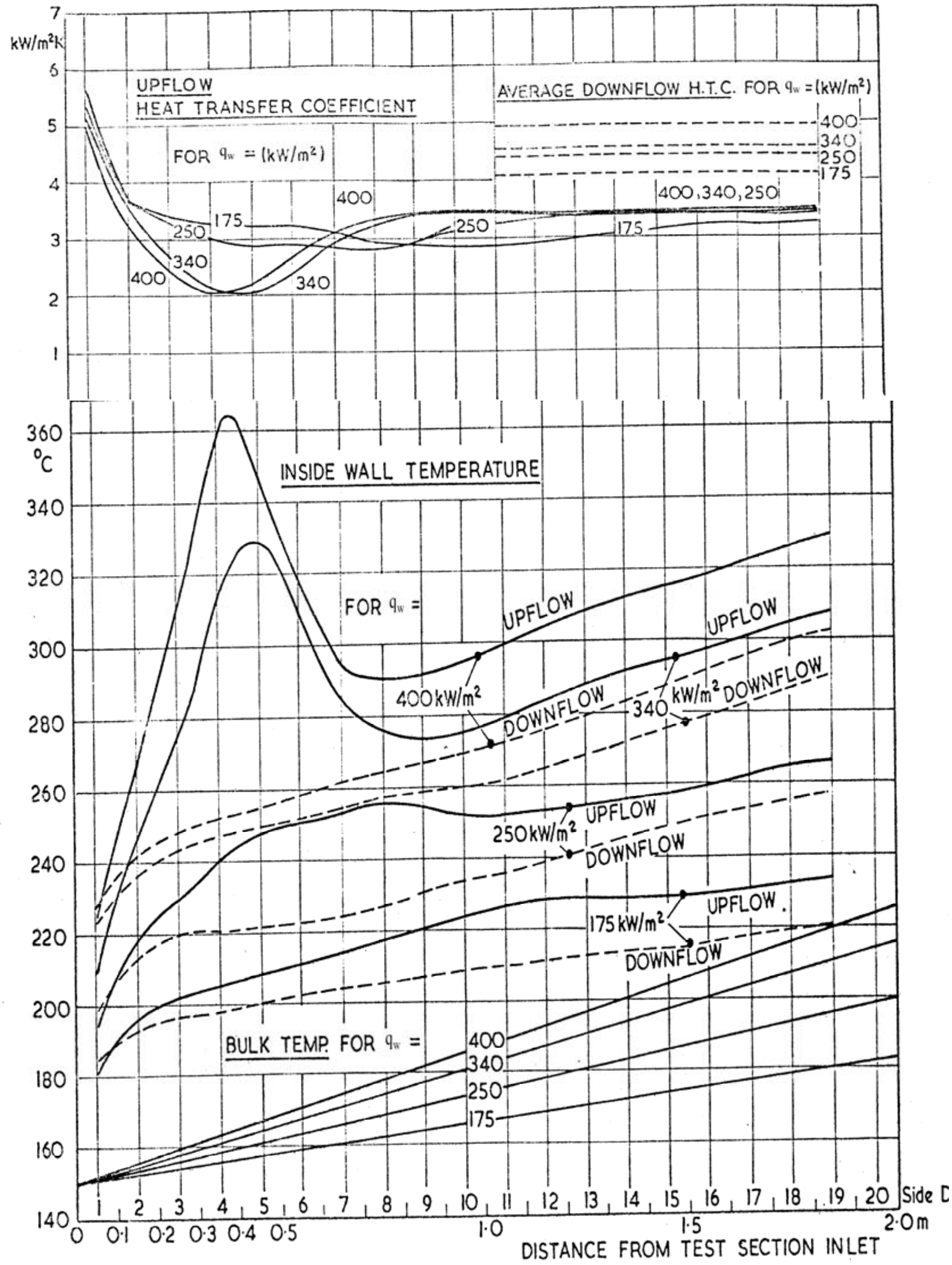


Figure 24 Developing mixed convection heat transfer to water at 250 bar with upward and downward flow; tube bore 22.5mm and mass velocity 380 kg/m^{2}s, Watts [55]

9. Further developments

The investigation of Watts marked the end of the experimental work at Manchester University on heat transfer to fluids at supercritical pressure. At that time the author's research interests shifted to buoyancy-influenced heat transfer with conventional fluids such as water and air at atmospheric pressure and also to heat transfer with liquid sodium. A comprehensive review of that work can be found in Jackson, 2006, [57]. Thus, the article on Supercritical Pressure Fluids published in the Encyclopeadia of Heat Transfer, Jackson, 1997, [58], might have signalled the end of the author's contributions to that topic. However this was not to be so because a renewed interest developed in heat transfer to fluids at supercritical pressure driven by the prospect of an important new application, advanced water-cooled nuclear reactors designed to operate at pressures above the critical value (SCWR's). The possibility of making use of his earlier experience with fluids at supercritical pressure stimulated him to become involved in this new venture. In two invited lectures (Jackson 2001, 2002), [59,60] he reviewed earlier work and then collaborated on experimental research which had been initiated at Tsinghua University, Beijing on heat transfer to carbon dioxide at supercritical pressure. Over the next few years the work there produced a steady stream of experimental results (see Jiang et al. [61-65]). Also, in conjunction with researchers elsewhere, he collaborated on CFD modeling studies of heat transfer to fluids at supercritical pressure (see He et al. [66], He et al. [67], He, Kim, Jiang and Jackson [68], He et al. [69], He, Kim and Jackson [70], He, Kim, Fewster and Jackson [71], He, Kim and Jackson [72], Sharabi et al. [73] and De Rosa et al. [74]).

In 2007 the author was invited to participate in the newly established Coordinated Research Programme on Heat Transfer Behaviour and Thermo-hydraulics Codes Testing for SCWRs under the auspices of the IAEA in Vienna. Since then, he has concentrated his attention on the development of physically-based, semi-empirical models of heat transfer to fluids at supercritical pressure (see Jackson [75-78] and Jackson, Jiang, Liu and Zhao [79]), extending his earlier work on mixed convection heat transfer to conventional fluids (Jackson [57]). The approach adopted to develop such models is outlined below in the final section of this paper.

10. Semi-empirical model of heat transfer to supercritical pressure fluids accounting for non-uniformity of fluid properties thermal expansion and buoyancy

10.1 A physically-based semi-empirical model of the impairment of forced convection heat transfer due to thermally-induced bulk flow acceleration

When fluid flows through a heated tube its bulk enthalpy increases, its bulk temperature rises and its density falls. Since the mass flow rate at all axial locations is the same, the bulk velocity increases and, therefore, the fluid accelerates. An additional applied pressure difference (over and above that required to overcome friction) is needed to maintain the flow rate. The velocity of the fluid in the boundary layer is lower than in the core region and so the extra pressure gradient applied is greater there than is needed to accelerate the fluid. Thus, the gradient of shear stress across the flow in the wall region has to undergo an adjustment. Consequently, in the buffer layer region the shear stress is lower than in a flow without acceleration at the same mass flow rate and less turbulence is produced. Hence, the heated flow takes on some of the characteristics of pipeflow at a reduced value of flow rate. The viscous sub-layer is thicker and the effectiveness of heat transfer is reduced. This mode of heat transfer has been referred earlier to as impaired forced convection. A semi-empirical model of such behavior is outlined below.

Using a simplified equation of motion and an idealised velocity profile, integration enables an expression for the reduction of shear stress across the wall layer to be related to bulk flow acceleration. Using the continuity equation this can be expressed in terms of the axial gradient of bulk density. On the assumption that this is mainly a consequence of the axial variation of enthalpy, an equation is obtained relating the reduction of shear stress across the wall layer to bulk velocity, wall heat flux and the wall layer thickness. Using empirical correlation equations for friction and heat transfer in pipeflow a relationship between heat transfer coefficient and shear stress is then obtained. Combining the latter with the equation for the drop of shear stress across the wall layer and expressing the resulting equation in terms of dimensionless groups lends to the following non-linear algebraic equation which describes the effect on heat transfer of thermally-induced bulk flow acceleration

$$\frac{Nu_b}{Nu_{b_o}} = \left[\left|1 - C_A Ac_b F_{VP_1}\left(\frac{Nu_b}{Nu_{b_o}}\right)^{-1.1}\right|\right]^{0.46} \tag{20}$$

Nu_b is the Nusselt number with bulk flow acceleration, Nu_{b_o} is corresponding value in the absence of acceleration, $Ac_b = Q_b/(Re_b^{1.625}\,Pr_b)$, is an acceleration parameter where $Q_b = q_w \beta_b d / k_b$. F_{VP_1} is a property ratio function, $(\mu_w/\mu_b)(\rho_w/\rho_b)^{-1/2}$. The coefficient C_A is proportional to the dimensionless buffer layer thickness δ^+ and assigning this the value 30 gives an estimate for C_A of 10^4.

Figure 25 shows the manner in which Nu_b/Nu_{b_o} is predicted to vary with the parameter $C_A Ac_B F_{VP_1}$. As can be seen, starting from a value near to unity, Nusselt number ratio, Nu_b/Nu_{b_o} falls systematically with increase of $C_A Ac_B F_{VP_1}$ due to turbulence production in the flow being reduced as a result of thermally-induced bulk flow acceleration. It reaches a value of just over 0.6 when $C_B Ac_b F_{VP_2} = 0.385$. With stronger heating complete laminarisation of the flow is likely to occur.

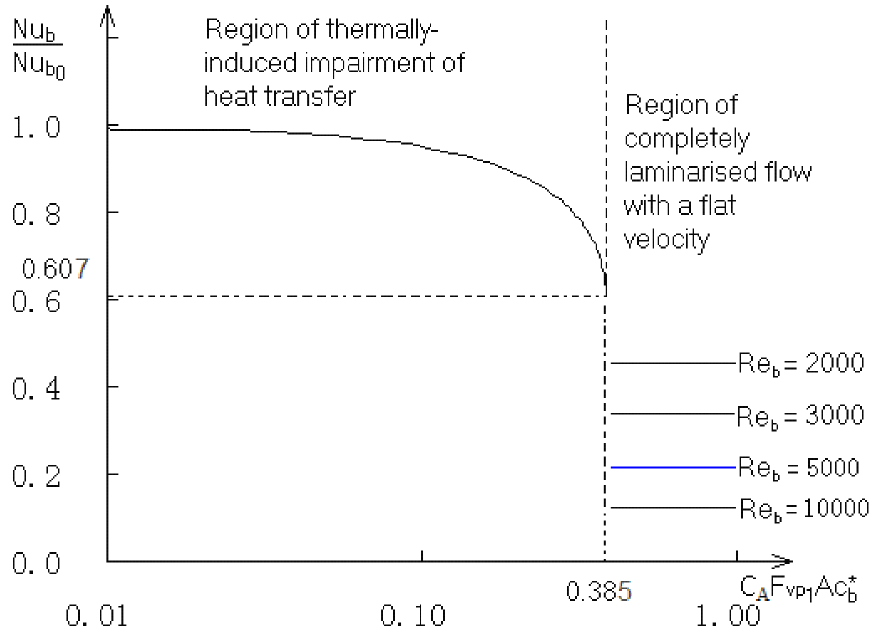


Figure 25 Predicted effect on heat transfer of bulk flow acceleration

10.2 A physically-based, semi-empirical model of axially developing variable property buoyancy-influenced heat transfer in a vertical tube

In a vertical heated tube the fluid within the thermal layer experiences an upward force due to buoyancy because of its reduced density. Thus, there will be an additional net shear force acting on this layer to balance the buoyancy force. The gradient of shear stress across the wall layer region will, therefore, be different from that in the absence of buoyancy.

For upward flow, shear stress will be lower at the outer edge. Thus the production of turbulence will tend to be reduced, the diffusion of heat through the action of turbulence will be less effective and some impairment of heat transfer is to be expected.

When fluid flows downwards in a heated tube the shear stress at the outer of the wall layer will be bigger than it would be in the absence of buoyancy, the production of turbulence will be increased and some enhancement of heat transfer is to be expected. Of course, there will be an additional effect of buoyancy aiding the motion in the case of upward flow and thus improving the advection of heat, or opposing the motion in the case of downward flow and the reducing the advection of heat. However, the effect of buoyancy on heat transfer due to modification of turbulence production will dominate the effect of buoyancy on advection. A physically-based, semi-empirical model has been developed to describe the effects on turbulent heat transfer to fluids at supercritical pressure, (see Jackson [75,76,77,78 and 79] and Jackson et al. [80]).

The approach is very similar to that used earlier to account for the effect of bulk flow acceleration. A simplified equation of motion is used in which the inertia terms are neglected. Account is taken of buoyancy by means of specifying an idealised distribution of density across the thermal layer, the thickness of which is related to the turbulent buffer layer thickness by an empirical factor involving the Prandtl number in that region. Integration enables an expression to be obtained for the change of shear stress across the thermal layer due to the influence of buoyancy. Using standard empirical correlation equations for friction and heat transfer in pipeflow a relationship between heat transfer coefficient and shear stress is then obtained. Combining the latter with the equation for the change of shear stress across the buoyant layer and expressing the resulting equation in terms of dimensionless groups gives the following simple non-linear algebraic equation to describe the effect of buoyancy on heat transfer in vertical tubes.

$$\frac{Nu_b}{Nu_{b_o}} = \left[\left|1 \mp C_B Bo_b F_{VP_1} F_{VP_3} \left(\frac{Nu_b}{Nu_{b_o}}\right)^{-1.1}\right|\right]^{0.46} \qquad (21)$$

Nu_b is the buoyancy-influenced Nusselt number, Nu_{bo} is the corresponding value in the absence of buoyancy. Bo_b is a buoyancy parameter $(= Gr_o/(\mathrm{Re}_b^{2.625}\,\mathrm{Pr}_b^{0.4})$, where $Gr_b = gd^3(\rho_b - \rho_w)/(\rho_b \nu_b^2))$. F_{VP_1} is the property ratio function defined earlier under Equation 20 and F_{VP_3} is a Prandtl number function $(\overline{\mathrm{Pr}}/\mathrm{Pr}_b)^{n_1}$ where n_1 is an index having a specified constant value or one calculated from a specified function of Prandtl number. The coefficient C_B is proportional to the dimensionless buffer layer thickness δ^+ and assigning this the value 30 gives an estimated value of C_B of about 2000. Figure 26 shows the manner in which Nu_b / Nu_{b_o} is predicted to vary with the buoyancy function, $C_B Bo_b F_{VP_1} F_{VP_3}$.

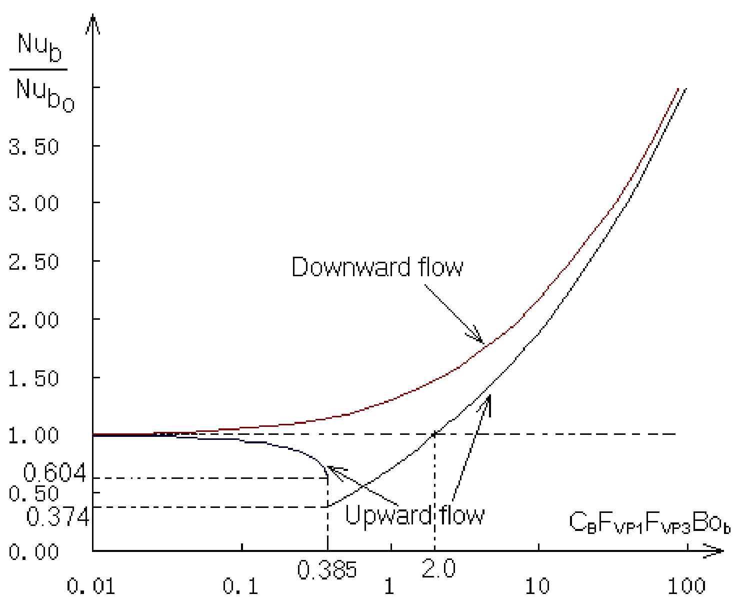


Figure 26 Effect of buoyancy on heat transfer

As can be seen, with upward flow, starting from a value very near to unity, the ratio Nu_b / Nu_{b_o} falls systematically with increase of the parameter $C_B Bo_b F_{VP_1} F_{VP_3}$ due to turbulence production in the flow being reduced as a result of the influence of buoyancy on the distribution of shear stress. Nu_b / Nu_{b_o} falls to a value of just over 0.6 when $C_B Bo_b F_{VP_1} F_{VP_3} = 0.375$. At this point the model predicts a discontinuous change to some other operating point on the lower curve shown. This curve rises systematically with increase of $C_B Bo_b F_{VP_1} F_{VP_3}$, indicating recovery of heat transfer as a result of enhanced turbulence production due to increasingly negative shear stress in the region where velocity profile is inverted. This recovery of heat transfer is eventually followed by enhancement of heat transfer in relation to the value for forced convection. With downward flow the model predicts systematic enhancement of heat transfer as $C_B Bo_b F_{VP_1} F_{VP_3}$ increases.

10.3 Combined effect of the influences of thermally induced bulk flow acceleration and buoyancy

When buoyancy is the dominant influence on heat transfer the model of forced convection with thermally-induced bulk acceleration can be combined with that for buoyancy-influenced heat transfer to take account of both effects. The following equation, describes the resulting effect on heat transfer due to both effects.

$$\frac{Nu_b}{Nu_{b_o}} = \left[\left|1-(C_A Ac_b \pm C_B Bo_b F_{VP_3})F_{VP_1}\left(\frac{Nu_b}{Nu_{b_o}}\right)^{-1.1}\right|\right]^{0.46} \tag{22}$$

Note that here the positive sign in the $\pm$ applies for buoyancy-aided flow and the negative one for buoyancy-opposed flow. Equation 22 can be re-written as

$$\frac{Nu_b}{Nu_{b_o}} = \left[\left|1-C_B F_{A/B}\left(\frac{Nu_b}{Nu_{b_o}}\right)^{-1.1}\right|\right]^{0.46} \tag{23}$$

in which $F_{A/B} = ((C_A/C_B)Ac_b \pm Bo_b F_{VP_3})F_{VP_1}$. As mentioned earlier the buoyancy coefficient C_B has an estimated value of 2000 (based of δ^+=30) but this might also need to be increased to about 3000, because of the simplified nature of the assumptions made in the analysis. Note that the ratio C_A/C_B takes a fixed value of about 5. The model indicates that, in order to try to correlate experimental data the Nusselt number ratio, $Nu_{b_{exp}}/Nu_{b_o}$ should be plotted against $F_{A/B}$.

10.4 Extended model to account for inertia effects and axial variation of buoyancy effect

In order to take account of inertia effects and axial variation of the buoyancy effect an extended version of the model is needed. An empirical function of the form shown below is introduced to account for such effects.

$$F(X)=[1-\exp(-AX)] \tag{24}$$

In Equation 24, X=x/d and A is a decay coefficient, the value of which is chosen to optimise the performance of the model in the upstream region where the flow is developing.

Firstly, F(X) is used to scale $F_{A/B}$ to give modified values of Nusselt number ratio $(Nu_b(i)/Nu_{bo}(i))_{mod}$ found using the equation

$$\left(\frac{Nu_b}{Nu_{b_o}}\right)_{\text{mod}} = \left[\left|1 - C_B F(X) F_{A/B} \left(\frac{Nu_b}{Nu_{b_o}}\right)_{\text{mod}}^{-1.1}\right|\right]^{0.46} \quad (25)$$

The model is used to calculate the curves of Nusselt number ratio for developing flow $(Nu_b/Nu_{bo})_{dev}$ versus F(X) by applying the scaling factor F(X) to the normalised relative Nusselt number $(Nu_b\text{-}Nu_{bo})/Nu_{bo}$ as in Equation 26, below.

$$(Nu_b/Nu_{bo})_{dev}=1+FX[(Nu_b/Nu_{bo})_{mod}-1] \quad (26)$$

Equations 25 and 26 can be used to fit the developing flow model to the correlated data by adjusting the coefficient C_B.

The extended model will give values of Nusselt number ratio which tend to those given by the fully developed model at downstream axial locations which depend on the value of the decay coefficient A. The optimum values of A for upward and downward flow remain to be established by comparing the calculated curves of $(Nu_b/Nu_{bo})_{dev}$ versus $F_{A/B}$ with the correlated data in the development regions. Approximate values for the decay coefficient A for upflow and downflow are 0.1 and 0.2, respectively. Work is still in progress to validate the extended model. This is currently being done in collaboration with researchers at Tsinghua University, Beijing, the University of Wisconsin, Madison and the University of Pisa, Italy.

11. General discussion and conclusions

There is scope for building on the experience gained in the experimental investigation described in Section 1 of the effect of strong non-uniformity of fluid properties on turbulent mixing and diffusion of heat. An experiment using a test section in the form of a long passage of annular cross section having a heated outer surface and a cooled inner one could enable a fully-developed condition to be achieved more reliably, thus eliminating the limitations encountered in the earlier experiment. If such an investigation were to be backed up by Direct Numerical Simulations (DNS), and possibly also Large Eddy Simulations (LES), then valuable insight into heat transfer with strong non-uniformity of fluid properties could be obtained.

The non-dimensional representation in Section 2 of the problem of heat transfer to fluids at supercritical in vertical tubes enabled the dimensionless groups for correlating experimental data to be identified and a clear statement to be made in Section 7 of the requirements for similarity and scaling. The wider problem of comparing experimental data on heat transfer from different fluids and establishing a more general basis for similarity and scaling remain to be addressed in a detailed and rigorous manner.

The early review of the literature on heat transfer to fluids at supercritical pressure reported in Section 3 enabled the limitations of the data and empirical equations available at that time to be clearly identified and useful recommendations made for further work, some of which were addressed in Section 4 where the samples of experimental work on supercritical pressure carbon dioxide were presented.

The similarity between buoyancy-influenced heat transfer behaviour fluid at pressures just above the critical value, under conditions where the tube temperature exceeded the pseudocritical value, and that for conditions of film boiling at pressures just below the critical value seen in Section 4 provides an interesting postscript to the description in the introductory section this paper of the molecular characteristics of near-critical pressure fluids. There it was pointed out that whereas such fluids appear from the microscopic point of view to change from a liquid-like state to a gas-like one in a continuous manner, there is evidence from x-ray diffraction observations of transition from liquid-like clusters of molecules in a matrix of gas to a gas-like structure with significant association between individual molecules.

The model of laminarisation of turbulent flow of fluids at supercritical pressure presented in Section 5 provides a simple and clear interpretation of the buoyancy-influenced heat transfer behaviour seen with carbon dioxide in Sections 4. A particularly useful feature of the results presented is that they were obtained for upward and downward flow under otherwise identical conditions. The further samples of experimental data for carbon dioxide and also was presented in Sections 6 and 8, respectively, provide valuable additional snapshots of buoyancy-influenced heat transfer to fluids at supercritical pressure.

The description in Section 7 of the comprehensive design report on ‘Heat Transfer to Supercritical Pressure Fluids’ commissioned by the UKAEA provides a useful picture of the coverage in that report, including the early attempt to correlate data on fully developed mixed convection heat transfer to carbon dioxide at supercritical pressure, the criteria developed for specifying the conditions for onset of buoyancy effects and the procedures used in the evaluation of the empirical equations available at that time (procedures which should still be followed in similar present-day exercises). The illustration in Section 7 of how comparisons can be made using the idea of limited similarity between experimental data for conditions of impaired forced convection (due to thermally-inducted bulk flow acceleration) and buoyancy-influenced experimental data should be useful as an aid to understanding heat transfer to fluids at supercritical pressure.

Finally, the outline in Section 10 of the approach adopted to develop physically-based, semi-empirical models of heat transfer to fluids at supercritical pressure provides a useful picture of the present ‘state of the art’. When successfully evaluated, the approach described could lead to significant progress in correlating data on heat transfer to fluids at supercritical pressure and describing such data by simple equations.

Reliable procedures are available for deciding whether or not influences of buoyancy are significant in experiments on heat transfer to fluids at supercritical pressure. Thus, the performance of empirical forced convection correlation equations for such fluids can be properly assessed so as to provide a sound basis for using them in the thermal design.

An important matter which remains to be confirmed experimentally is whether acceleration of the flow due to strong heating really can have a significant effect on turbulence under the conditions encountered in practical applications with fluids at supercritical pressure and cause impairment of heat transfer. The validity of a particular assumption made in modeling that phenomenon, namely, that the increase of bulk temperature is the dominant factor in reducing the fluid density, needs to be confirmed.

Precise and detailed measurements of velocity, temperature and turbulence in heat transfer experiments with fluids at supercritical pressure are needed, especially for conditions where heat transfer is impaired as a result of the influence of buoyancy or thermal expansion. There are practical difficulties and the cost and effort involved may limit such work, even though with present day instrumentation experimental techniques such measurements are certainly possible.

Investigations of heat transfer with thermal boundary conditions other than that of imposed heat flux would help to advance our understanding of some fundamental aspects of heat transfer to fluids at supercritical pressure and extend available thermal design procedures. Further experiments with passages of non-circular cross-section (triangular and square) would be useful to provide additional data for comparison with computational simulations, especially for buoyancy-influenced conditions. Studies of the influence of wall conduction and resistivity changes on the stability and non-uniformity of heat transfer are needed. Experiments on systems which involve cooling rather than heating and experiments to extend the range of fluids for which correlations have been tested would be of value.

By causing heat to be diffused across a flow in which the temperature remains constant in the flow direction, the influence on turbulent diffusion of non-uniformity of density can be retained whilst effects of advection are removed. The experiment reported here using a test section formed by two parallel plans, one heated and the other cooled did not involve heat fluxes high enough to cover the range of temperature which is of real interest in this connection. A test section in the form of an annulus with a heated core and a cooled outer surface might prove to be a more appropriate arrangement for such experiments.

Direct Numerical Simulation (DNS) is potentially a valuable way in which to study and understand the basic fluid physics involved in heat transfer to fluids at supercritical pressure. Large Eddy Simulation (LES) is a further approach which could be used for this purpose. Models of turbulence and turbulent heat transfer in fluids at supercritical pressure for use with the RANS equations have not yet been developed to the stage where such simulations can be used with confidence for the purposes of heat transfer prediction. A feature of results produced in a number of recent studies is that the low Reynolds number turbulence models used to try to describe the effect of buoyancy on heat transfer all seem to over-respond to the strong non-uniformity of fluid properties which is encountered within the flow when the conditions are such that the tube wall temperature exceeds the pseudocritical value. Better turbulence models able to reliably describe influences of fluid property non-uniformity on turbulent flow and heat transfer are needed.

Nomenclature

A	Decay constant in Equation 15
Ac_b	Acceleration parameter $(= Q_b/(Re_b^{1.625} Pr_b))$
c_p	Specific heat at constant pressure, kJ/kgK
Bo_b	Buoyancy parameter $Gr_b/(Re_b^{2.625} Pr_b^{n_1})$

C_A	Acceleration coefficient in Equation 20
C_B	Buoyancy coefficient in Equation 21
d	Tube diameter, m
g	Acceleration due to gravity, m/s^2
h	Enthalpy, kJ/kg
f_b	Friction factor $(\tau_w/(\frac{1}{2}\rho_b u_b^2))$
F_g	Dimensionless gravitational group $(= gd^3/\nu_o^2)$
F_{VP_1}	Variable property function 1 $((\bar{\mu}/\mu_b)(\bar{\rho}/\rho_b)^{-1/2})$
F_{VP_2}	Variable property function 2 $(=(\rho_w/\rho_b)^{0.3}(\bar{c}_p/c_{p_b})^{n_2})$
F_{VP_3}	Variable property function 3 $(=(\overline{Pr}/Pr_b)^{-n_1}$
Gr_b	Grashof number based on density difference $(gd^3(\rho_b-\rho_w)/(\rho_b\nu_b^2))$
k	Thermal conductivity, kW/mK
ℓ	Turbulent mixing length, m
n_1	Prandtl number index
n_2	Index defined under Equation 7
Nu_b	Nusselt number $(= q_w d/(k_b(T_w-T_b)))$
Nu_{b_o}	Nusselt number for variable property forced convection $(= q_w d/(k(T_{w_o}-T_b)))$
p	Pressure, MPa or bar
P	Dimensionless pressure $(= pd^2/(\rho_o\nu_o^2))$
Pr	Prandtl number $(= \mu c_p/k)$
$\overline{Pr}$	Integrated value, wrt T over the range T_b to T_w
Pr_b	Bulk value of Prandtl number $(=\mu_b c_{p_b}/k_b)$
$\overline{Pr}_b$	Prandtl number defined using $\bar{c}_p$ $(= \mu_b\bar{c}_{p_b}/k_b)$
q_w	Wall heat flux, kW/m^2
Q_b	Thermal expansion parameter $(= q_w\beta_b d/k_b)$
r	Radial coordinate, m
Re_b	Reynolds number $(= \rho_b u_b d/\mu_b)$
t	Time, s
T	Temperature, ^oC or K
T_b	Local bulk temperature, ^oC
T_w	Local wall temperature, ^oC
T_{w_o}	Local wall temperature under conditions of variable property forced convection
u	Local axial component of velocity, m/s
v	Transverse component of velocity, m/s, or specific volume, m^3/kg
u_b	Local bulk velocity, m/s
U	Dimensionless axial velocity component, $(= \rho_o ud/\mu_o)$
V	Dimensionless transverse velocity component, $(= \rho_o vd/\mu_o)$
x	Axial coordinate, m
X	Dimensionless axial coordinate (=x/d)
R	Dimensionless radial coordinate, (=r/d)

y Transverse coordinate (measured inwards from wall), m

Greek symbols

α Heat transfer coefficient ($=q_w/(T_w-T_b)$), kW/m^2K

β Thermal expansion coefficient $(=-(1/\rho)(\partial\rho/\partial T)_p)$

μ Viscosity, kg/ms

ν Kinematic viscosity, m^2/s

ε Turbulent diffusivity of momentum, m^2/s

η Dimensionless time coordinate $(=\nu_o t/d^2)$

ρ Density, kg/m^3

δ Turbulent buffer layer thickness, m

δ_T Thermal layer thickness, m

τ Shear stress, N/m^2

τ_w Wall shear stress, N/m^2

τ_t Local turbulent shear stress

ω Mass velocity, kg/m^2s

θ Dimensionless temperature ($=T/T_o$)

Subscripts

b Denotes properties evaluated at the bulk temperature

o Denotes properties evaluated at the inlet fluid temperature

pc Denotes pseudocritical temperature

w Denotes properties evaluated at the wall temperature

c Denotes critical point values

hw Refers to the hot wall

cw Refers to the cold wall

Superscripts

\- Bar over a property symbol denotes that it is integrated over the range between the wall and the bulk temperature, or bar over a velocity component denotes that it is a time mean value in turbulent flow

\+ Denotes the thickness of buffer layer in the universal wall coordinate form, $\delta^+ = \rho_b \tau_w^{1/2} \delta / \mu_b$

′ Denotes fluctuating component of velocity component in turbulent flow

* Denotes normalized fluid property ratio made dimensionless using the value at the inlet fluid temperature and pressure

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