PERGAMON

International Journal of Heat and Mass Transfer 41 (1998) 4037–4054

International Journal of
HEAT and MASS TRANSFER

Upflow turbulent mixed convection heat transfer in vertical pipes

Gian Piero Celata[a,]*, Francesco D'Annibale[a], Andrea Chiaradia[b], Maurizio Cumo[b]

[a] *National Institute of Thermal-Fluid Dynamics, ENEA, Energy Department, C.R. Casaccia, Via Anguillarese, 301 00060 S.M. Galeria, Rome, Italy*

[b] *Nuclear Engineering and Energy Conversion Department, University of Rome "La Sapienza", C.so Vittorio Emanuale II, 244 00100 Rome, Italy*

Received 20 February 1998; in final form 29 April 1998

Abstract

The present work deals with the results of an experimental investigation on heat transfer in water cooled vertical pipes, for thermal–hydraulic conditions ranging from forced convective flow to mixed convective flow. The flow of water in the pipe is upwards.

Experimental data confirm the reduction in the heat transfer rate for mixed convection in upward heat flow, mainly due to the laminarization effect in the near-wall region (buoyancy effect). They are in a very good agreement with numerical methods, such as the k–ε-model.

A new method for the calculation of the heat transfer coefficient in upward mixed convection heated flow is proposed. It is based on the well-known superposition method (heated downflow) modified accounting for the phenomenology of the upward heated flow in comparison with downflow heated conditions. © 1998 Elsevier Science Ltd. All rights reserved.

Nomenclature

a parameter defined in equation (10)
b parameter defined in equation (10)
Bo buoyancy parameter, defined in equation (3)
c_p specific heat at constant pressure [J kg^{-1} K^{-1}]
D channel diameter [mm]
G mass flux [kg m^{-2} s^{-1}]
g gravitational acceleration [m s^{-2}]
Gr_h Grashof number based on the wall heat flux, equation (3)
Gr_t Grashof number based on the temperature difference, equation (2)
h heat transfer coefficient [W m^{-2} K^{-1}]
k liquid thermal conductivity [W m^{-1} K^{-1}]
L channel length [m]
Nu Nusselt number, hD/k
Nu_{FL} parameter described in equation (8), nondimensional
P parameter defined in equation (7), nondimensional
p pressure [Pa]
Pr Prandtl number, $c_p\mu/k$
q'' heat flux [W m^{-2}]
Re Reynolds number, GD/μ
T temperature [°C]
u velocity [m s^{-1}].

Greek symbols
β coefficient of thermal expansion [K^{-1}]
μ dynamic viscosity [kg m^{-1} s^{-1}]
ρ density [kg m^{-3}]
ψ corrective factor for aided mixed convective flow, equation (9).

Subscripts
b bulk conditions
cal calculated
df downflow

* Corresponding author. Fax: 00 39 6 3048 3026; e-mail: celata@casaccia.enea.it

0017–9310/98 $19.00 © 1998 Elsevier Science Ltd. All rights reserved
PII: S0017–9310(98)00177–X

exp experimental
f pertains to film conditions (average between wall and bulk conditions)
for pertains to forced convective flow conditions
in inlet
l pertains to the liquid phase
m pertains to mixed convection
nat pertains to natural convective flow conditions
up upflow
w wall, at wall temperature.

1. Introduction

Heat transfer in heated water mixed convective pipe flow (where natural convection has significant impact on forced convective flow) strongly depends on physical and geometric parameters. In other words, heat transfer in horizontal pipe flow is much different from the vertical flow behaviour, and, in the latter case, we have to distinguish between upward and downward flow.

In vertical upward heated flow the buoyancy driving force (natural convection contribution) is in the same direction as the pressure gradient driving force (forced convection contribution) and we call it aiding (or assisting) flow, while in vertical heated downflow driving forces act in opposite directions, and we call it opposing flow. For each of the above conditions we have to distinguish further between uniform wall temperature and uniform heat flux as thermal boundary conditions.

Although many researchers have dealt with heat transfer in pipe flow mixed convection since the late 1930s, if we look at a single specific case, with given thermal and geometric conditions, we realize that a lacking of experimental data still exists. This is especially true for the case of heat transfer in upward mixed convective heated flow of water in vertical channels, where, apart from a successful numerical approach pursued by some researchers, no satisfactory correlation to predict the heat transfer coefficient has been proposed. The first attempt to give a reliable method for the prediction of the heat transfer coefficient in such conditions has been done by Aicher and Martin [1]. Authors also present an interesting and exhaustive review of existing works in this field, distinguishing between theoretical works, [2–11], and experimental works [12–18].

Among the experimental works, Aicher and Martin [1] quote the following: Watzinger and Johnson [12], who measured the heat transfer coefficient in turbulent water flows under both aiding and opposing flow conditions, with $L/D = 20$; Norris and Sims [13] who investigated the heat transfer in oil flows under conditions where forced convection was the dominant driving force; Metais [14] who performed experiments of heat transfer to water flowing in a tube with $L/D = 29$; Herbert and Sterns [15] who presented experimental data of aiding and opposing convection heat transfer in a $L/D = 80$ tube; Axcell and Hall [16] who presented data for air in downward flow with $L/D = 6$; Saylor and Joye [17] who carried out experiments with water in aiding and opposing flow in a tube with $L/D = 50$; Temu et al. [18] who performed experiments of mixed convection heat transfer to turbulent water flow in a $L/D = 74$ tube.

Most of the findings described in the above mentioned papers shows a heat transfer reduction under aiding heated flow conditions with reference to pure natural and pure forced convection. For opposing heated flow, the heat transfer is instead generally higher than the two limiting cases. A possible interpretation of heat transfer behaviour in turbulent mixed convective flow, especially for the heat transfer decrease in aiding flow, mainly due to a flow laminarization in the near wall viscous layer, can be found in the literature [19–30], and will be discussed afterwards.

The present paper will present the results of a research aimed at obtaining new data on heat transfer in aided heated mixed convection with different values of L/D, together with a new method for the prediction of the heat transfer coefficient.

2. Physical background

Just to fix our mind, we may have natural convection flow when buoyancy forces (represented by the Rayleigh number) are much greater than pressure gradient forces (represented by the Reynolds number), and vice versa for forced convection flow. In between, natural convection aids forced convection and we talk of mixed convection.

The most significant feature of heat transfer in turbulent upward mixed convective heated flow in a vertical channel is the reduction in heat transfer, with regard to either natural or forced convection, for conditions where natural and forced convection act together.

In downward vertical heated flow the situation is quite different, and the heat transfer is generally larger than the corresponding forced convection value. Such a situation is reported in Fig. 1, where typical trends of heat transfer for aiding and opposing mixed convection are shown, in terms of heat transfer related to heat transfer for pure forced convection, versus buoyancy parameter, *Bo* (given in equation (3)).

Such behaviour may be explained by considering the turbulence production between the viscous layer and the bulk of the flow, which the diffusive energy transport (main responsibility for heat transfer to turbulent flows) is linked to, as argued in the two-layer model [31]. In this model, the liquid of the layer adjacent to the heated wall, due to a lower density than in the bulk liquid, is subjected to a buoyancy force which acts in the same flow direction. Therefore, buoyancy force tends to reduce the shear stress

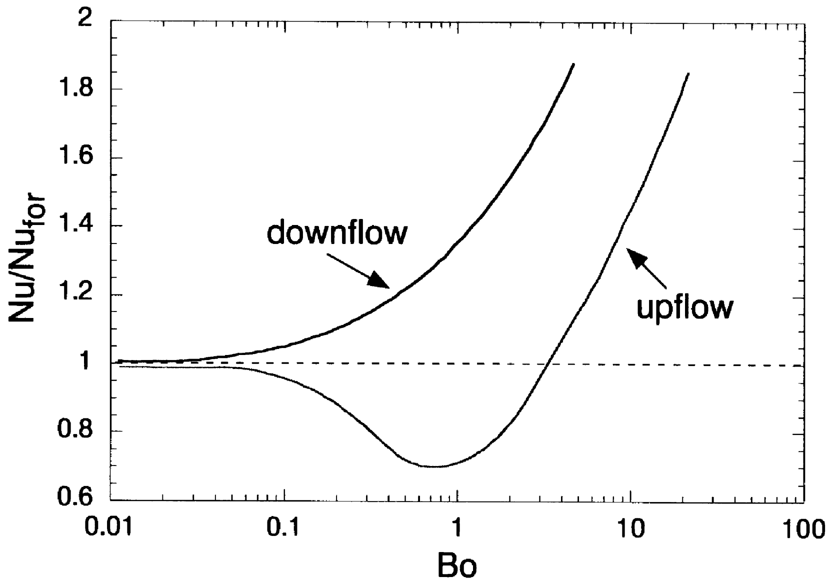


Fig. 1. Schematic view of heat transfer for aiding and opposing mixed convection.

in the layer, reducing turbulence production and causing flow laminarization. Heat transfer reduction, with respect to pure forced convection, depends on both local velocity and temperature profiles, and is therefore affected by thermal and hydraulic entrance effect, and by thermal boundary conditions.

Typical examples of velocity profiles and shear-stress distributions due to different temperature profiles, and therefore different Grashof numbers, are plotted in Fig. 2 [28]. As the heat flux increases, we have reduction in the shear stress and a laminarization effect which results in a reduction of the heat transfer. For further increase in the heat flux, buoyancy causes a change in the shear-stress slope, and then a further increase in turbulence. This eventually results in a sharp change in the heat transfer slope, which starts increasing with respect to that expected from pure natural convection.

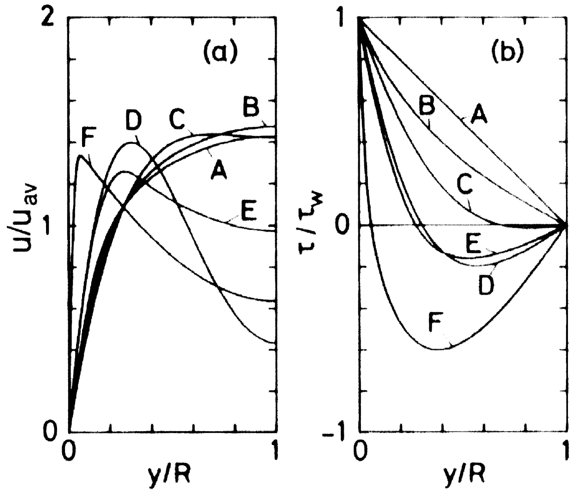


Fig. 2. Velocity (a) and shear-stress (b) distribution at various Grashof number at a constant $Re = 3000$; A, $Gr = 2.1 \times 10^3$, turbulent; B, $Gr = 6.1 \times 10^4$, turbulent; C, $Gr = 8.8 \times 10^4$ laminar; D, $Gr = 2.7 \times 10^5$, laminar; E, $Gr = 3.3 \times 10^5$, turbulent; F, $Gr = 9.2 \times 10^6$, turbulent.

3. The experimental apparatus

The facility used to perform the experiment described in the present work is shown in Fig. 3. The water flow is elaborated by a three-head piston pump (maximum flow rate 1800 kg h^{-1}, with a residual pulsation below 2.5%, owing to the dumper), filtered, pre-heated and delivered to the bottom end of the vertical test channel, after which it is directed to the tank-heat exchanger. The test section is a 316-type stainless steel tube, placed vertically, thermally insulated and electrically heated by Joule effect (DC heating). The water flow is upward. The bottom copper clamp is mobile, in order to have different heated lengths. Wall temperature are measured using two K-type 0.5 mm thermocouples, the hot junction of which is placed 0.5 mm inside the tube wall thickness. Inner wall temperature (for the evaluation of the heat transfer coefficient) is calculated by correcting the thermocouple reading with the Fourier equation. Water temperature is measured at the inlet, the exit and in the middle of the

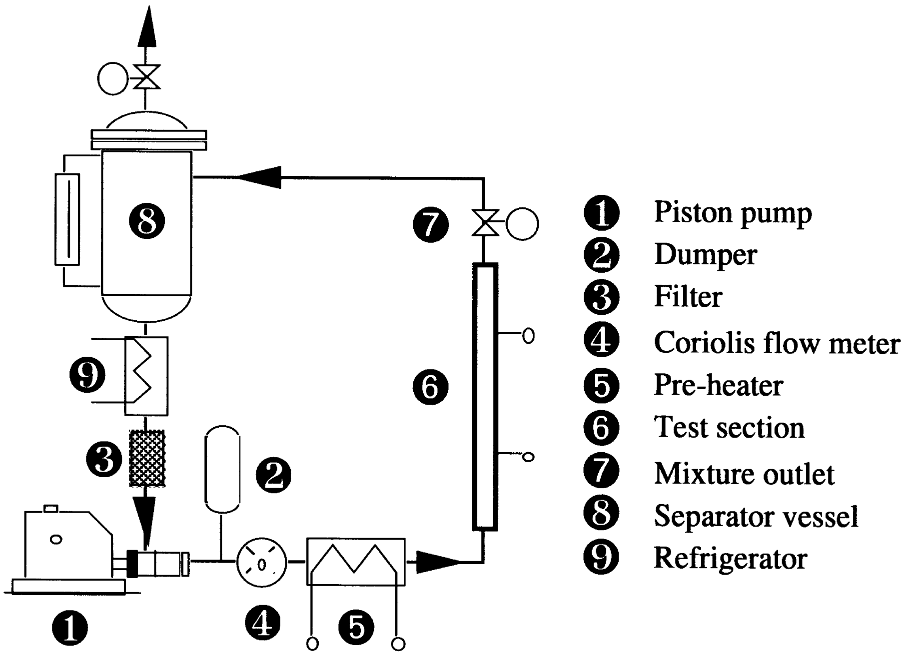


Fig. 3. Schematic of the experimental facility.

test section using the same kind of thermocouple as for the wall temperature measurements. Inlet and outlet pressure are measured using sealed strain-gauge pressure transducers, while the mass flow rate is measured using a Coriolis-meter (up to 250 kg h^{-1}) and a turbine flow meter (above 250 kg h^{-1}).

Test section characteristics are given in Table 1, while test conditions are reported in Table 2. Some derived parameters are shown in Table 3.

4. Experimental results

About 2633 data points have been collected considering the geometrical and thermal hydraulic parameters tested as described in Tables 1 and 2. Flow regimes tested ranged from turbulent natural to forced convection, through mixed convection regime, where we experienced most of data points, about 75%, according to the map of Metais and Eckert [20]. These latters proposed a map, which in a diagram of Re vs $GrPrD/L$ allows us to distinguish the laminar-turbulent transition, and different turbulent convection regimes — free, forced and mixed — for flow in vertical circular tubes, for $10^{-2} < PrD/L < 1$. The results are valid for both upflow

Table 1
Test section characteristics

Inner diameter [mm]	26			
Thickness [mm]	1			
Heated length [m]	0.5	0.72	0.92	1.14
Max thermal power, kW (1400 A)	7.8	11.3	14.9	17.5
L/D at the thermocouple position (nominal values) (measured from the bottom copper clamp)	10, 16	18, 24	27, 33	35, 40

Table 2
Test matrix (2633 tests)

Pressure [p]	from 0.1 to 0.55 MPA
Inlet wall temperature [$T_{l,in}$]	from 5 to 99°C
Heat flux [q'']	from 10 to 243 kW m^{-2}
Mass flux [G]	from 34 to 460 kg m^{-2} s^{-1}

Table 3
Derived test parameters

Water velocity [u]	from 0.034 to 0.46 m s^{-1}
Inner wall temperature [T_w]	from 25 to 153°C
Bulk water temperature [T_b]	from 12 to 113°C
Reynolds number [Re]	from 800 to 23 000
Buoyancy parameter [Bo]	from 0.038 to 12 500
Heat transfer coefficient [h]	from 370 to 4370 W m^{-2} K^{-1}

and downflow, and for uniform wall temperature and uniform heat flux boundary conditions.

Present experimental data have been plotted using the Metais and Eckert map, and the result is shown in the three graphs of Fig. 4. In the graph of Fig. 4(a) data points are grouped as a function of the buoyancy parameter, Bo, as given in equation (3). Most of the data points, characterized by $0.1 < Bo < 10$, are typically in the turbulent mixed region of the Metais and Eckert map. As per graphs plotted in Fig. 4(b) and 4(c), different symbols refer to the relative error between the experimental heat transfer coefficient, h, and that calculated for forced convective flow using the well-known Dittus–Boelter correlation, h_{for}, corrected with the viscosity ratio to account for thermal-properties change due to temperature gradients:

$$h_{\mathrm{for}} = 0.023\frac{k}{D}Re_{\mathrm{b}}^{0.8}Pr_{\mathrm{b}}^{0.4}\left(\frac{\mu_{\mathrm{b}}}{\mu_{\mathrm{w}}}\right)^{0.11} \quad (1)$$

as shown in Fig. 4(b), or between h and that calculated for natural convection using the Churchill correlation (see [9]), h_{nat}:

$$h_{\mathrm{nat}} = \frac{0.15\frac{k}{D}(Gr_{\mathrm{t}}Pr_{\mathrm{w}})^{1/3}}{(1+(0.437/Pr_{\mathrm{w}})^{9/16})^{16/27}} \quad (2)$$

with

$$Gr_{\mathrm{t}} = \frac{g\beta(T_{\mathrm{w}}-T_{\mathrm{b}})D^3}{(\mu_{\mathrm{w}}/\rho_{\mathrm{w}})^2}$$

as shown in Fig. 4(c). The Metais and Eckert map and available correlations for forced and natural convection are in close agreement in identifying mixed convective data. Where indicated by the subscript w, physical properties are calculated at wall temperature. We can realize how the relative error in the heat transfer coefficient evaluation increased as we move toward the mixed convection zone, increasing more and more if we go further into the convective regime, which is the opposite to that recommended for the used correlations. Data points plotted in the three graphs of Fig. 4 are a sample of about 300 data out of 2633. Using all the data would have produced only a black spot, preventing the reader from understanding the behaviour.

Typical data obtained with different L/D values are plotted in Fig. 5, where the ratio between the experimental heat transfer coefficient and the value obtained using the Dittus–Boelter correlation is reported versus the buoyancy parameter, Bo, given by:

$$Bo = 8\times10^4\frac{Gr_{\mathrm{h}}}{(Re_{\mathrm{f}}^{3.425}Pr_{\mathrm{f}}^{0.8})} \quad (3)$$

being

$$Gr_{\mathrm{h}} = \frac{g\beta q''D^4}{\mu_{\mathrm{f}}^2k_{\mathrm{f}}}$$

Such an expression of the buoyancy parameter was first developed by Hall and Jackson [32] from consideration of modified shear production of turbulent kinetic energy under mixed convection conditions, and is the most recent form of the parameter to be developed (see [8]). The results plotted in Fig. 5 show the reduction of the heat transfer in aided mixed convection, wherever the buoyancy parameter is between 0.03 and 3. Heat transfer reduction is at a maximum when the buoyancy parameter is around 1. Under these conditions, buoyancy forces are of the same order of magnitude as pressure gradient forces, and the laminarization effect is maximum. Continuous lines in the graphs of Fig. 5 refer to the prediction of the heat transfer coefficient obtained using the k–ε model proposed by Cotton and Jackson [30]. It is also evident from Fig. 5 the influence of the L/D on the heat transfer reduction (as referred to pure forced convective heat transfer), which would seem to be an increasing function of L/D itself, as shown in Fig. 6, where the heat transfer reduction, in terms of the ratio between the experimental and the calculated heat transfer coefficient in forced convection at $Bo = 1$ (i.e., at the conditions for which the heat transfer reduction is maximum) is plotted vs L/D.

5. Data analysis

5.1. *Existing methods*

As already said at the beginning of this paper, very few tools for the prediction of the heat transfer coefficient in turbulent aided mixed convection exist. Among them, we have already reported about the numerical approach proposed by Cotton and Jackson [30], who used the original Launder and Sharma [33] low-Reynolds k–ε turbulence model for developing flow in a tube. The low-Reynolds k–ε model has been used for comparison with present data, which has been already shown in Fig. 5. Curves plotted in the figure are directly extracted from Cotton and Jackson [30] paper, to avoid re-calculation of constitutive equations. Available calculations from the paper are for $L/D = 10$ and 20 only, obtained for air flow. The k–ε model shows close agreement with present experimental data, evidencing also a good capability in accounting for the L/D effect, which has been very little investigated so far in mixed convection heat transfer, see Aicher and Martin [1]. It is worthwhile saying here that the k–ε model is also able to predict data in opposing mixed convection flow, as reported by authors, although showing major discrepancies with experimental data in strongly buoyancy-influenced descending flow.

In spite of its good modelling of the turbulence production from the viscous layer to the bulk and its influence on the heat transfer in mixed convection (both aided and opposed, and for different fluids, such as air and

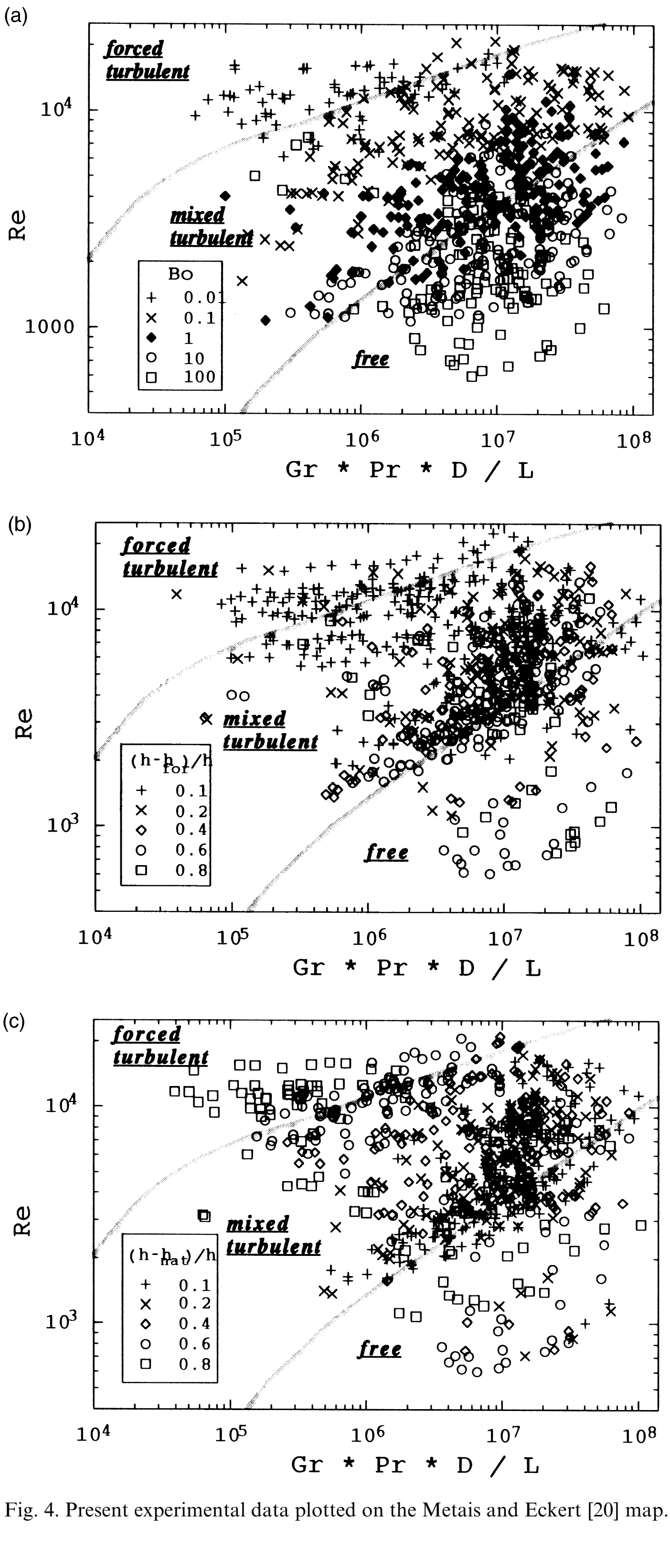
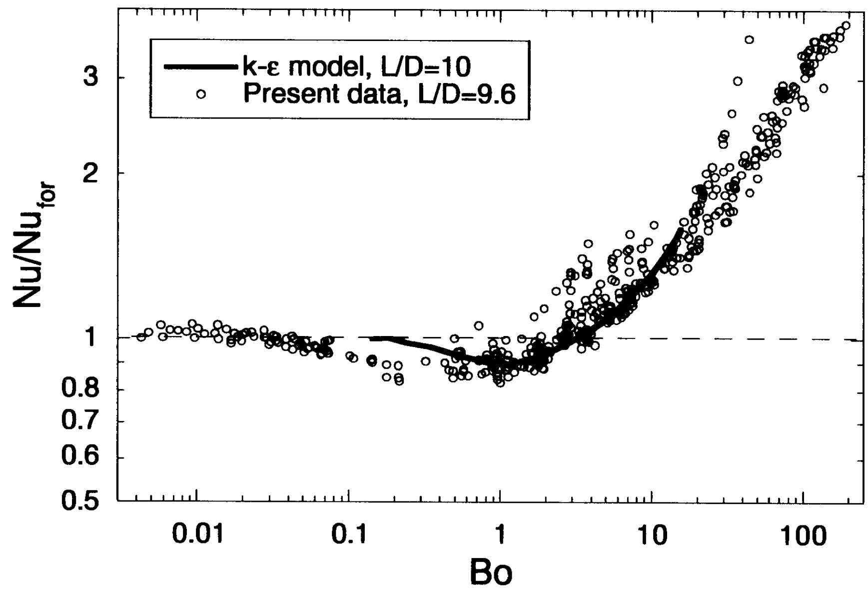


Fig. 4. Present experimental data plotted on the Metais and Eckert [20] map.



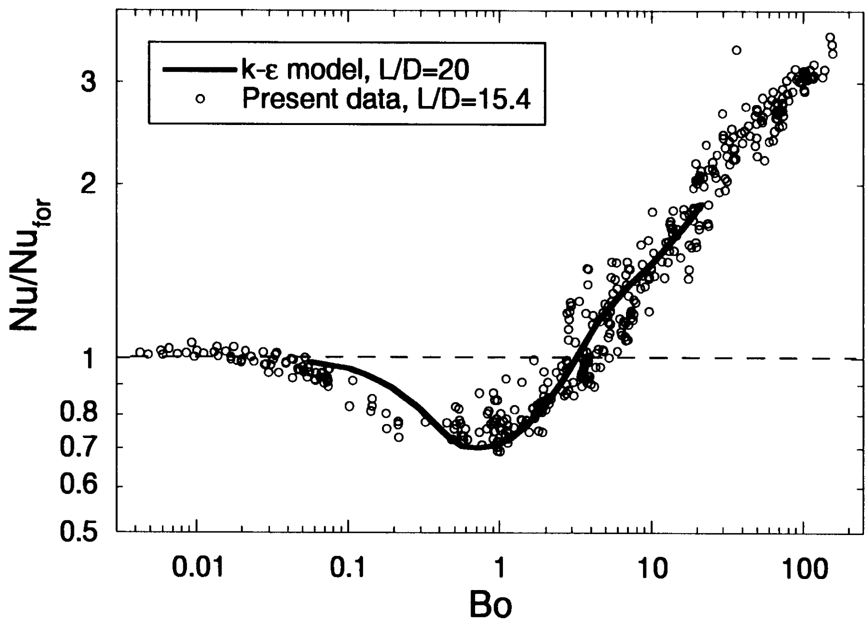


Fig. 5. Experimental-to-calculated heat transfer coefficient for forced convection (Dittus–Boelter equation) vs buoyancy parameter, together with k–ε model predictions, for different L/D.

water), the k–ε model has the main drawback in its complex utilisation, especially in the numerical solution of the constitutive equations, which lowers its possible interest for practical engineering purposes. It would therefore seem to be reasonable to look for simpler engineering methods which, although in a restricted range of validity, provide us with fast predictions of the heat transfer coefficient in aided mixed convection with an acceptable accuracy. The first method has been proposed by Martinelli and Boelter [34], and is a simple numerical regression using Nusselt number for forced and natural convection, originally developed for laminar opposing convection:

$$Nu^3_{\mathrm{m,up}} = |Nu^3_{\mathrm{for}} - Nu^3_{\mathrm{nat}}| \tag{4}$$

Typically, the behaviour of equation (4) correlation is shown in Fig. 7, where the ratio between the mixed and

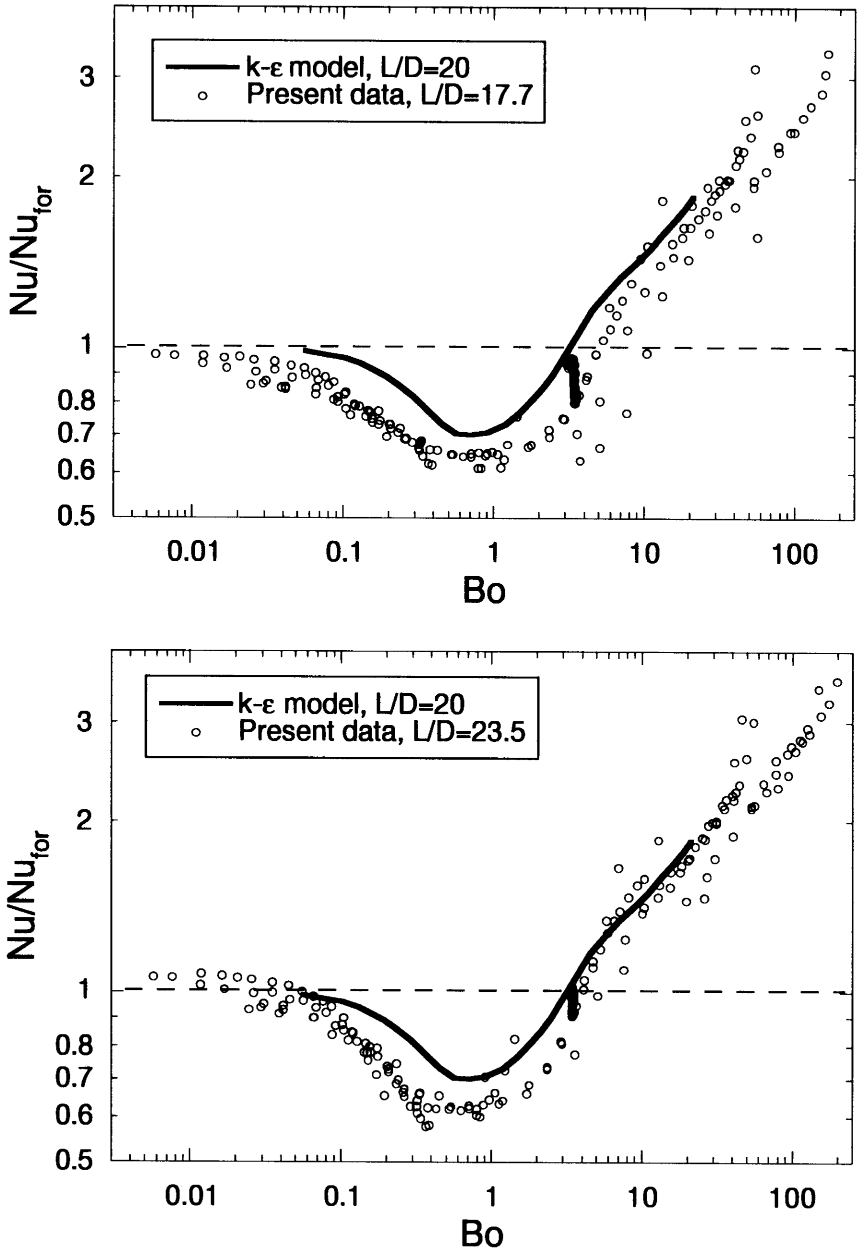


Fig. 5—*continued.*

the natural convection Nusselt number is plotted versus the ratio between the forced and the natural convection Nusselt number for the data of Herbert and Sterns [15], as reported by Churchill [9]. Such a method shows a discontinuity when $Nu_{\mathrm{for}} = Nu_{\mathrm{nat}}$, providing $Nu = 0$, which does not actually happen, and is therefore not completely satisfactory. Figure 7 also shows data for opposing flow, well correlated by the Churchill equation [9], always given in the numerical regression form, as:

$$Nu^3_{\mathrm{m,df}} = Nu^3_{\mathrm{for}} + Nu^3_{\mathrm{nat}} \tag{5}$$

This correlation will be used hereafter. In equations (4) and (5), Nu_{for} and Nu_{nat} can be obtained from equations (1) and (2), respectively.

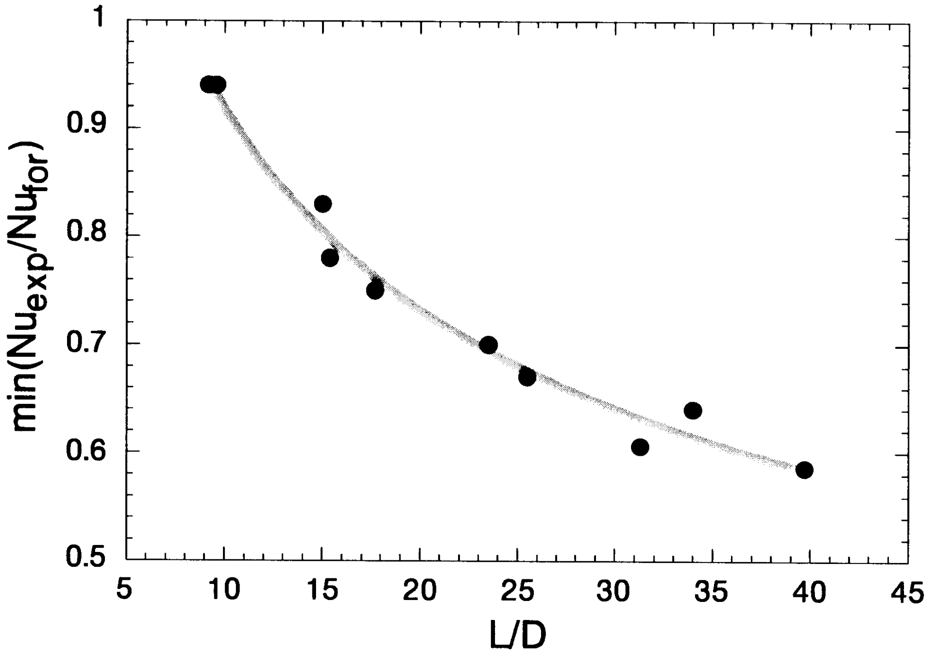


Fig. 6. Heat transfer impairment with reference to forced convection, at $Bo = 1$, for present data.

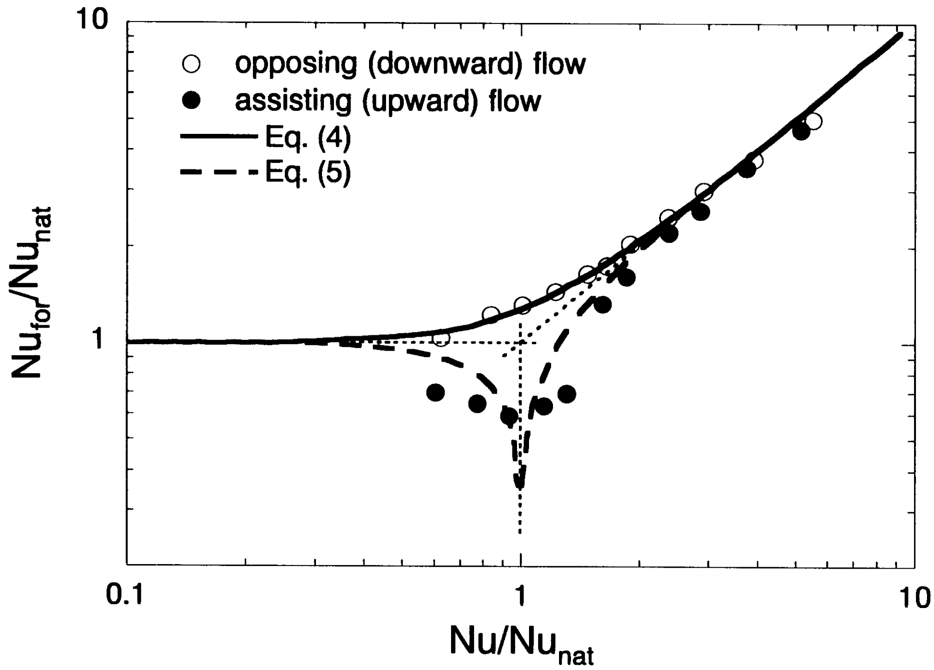


Fig. 7. Martinelli and Boelter, and Churchill correlations for upflow and downflow mixed convection respectively, Churchill [9].

To overcome this deficiency, Aicher and Martin [1] propose a new method for aided mixed convection. They start from an equation similar to equation (5), which was proved to be successful to predict their own downflow data:

$$Nu_{\mathrm{m,df}}^2 = Nu_{\mathrm{for}}^2 + Nu_{\mathrm{nat}}^2 \quad (6)$$

where Nu_{nat} is given by equation (2), slightly modified to better fit their data, and Nu_{for} is obtained using the equation of Gnielinksi [35]. Authors correct the value obtained using equation (6), valid for downflow, with a function which accounts for the decrease in the heat transfer in aided mixed convection. They define a parameter P as:

$$P = \frac{Nu_{\mathrm{nat}} - Nu_{\mathrm{for}}}{Nu_{\mathrm{m,df}}} \quad (7)$$

which varies from $P = -1$ for pure forced convection to $P = 1$ for pure natural convection, and obtain

$$\frac{Nu_{\mathrm{m,up}}}{Nu_{\mathrm{m,df}}} = \left(1 - \left(1 - 2\frac{Nu_{\mathrm{FL}}}{Nu_{\mathrm{m,df}}}\right) f(P)\right) \quad (8)$$

where

$$f(P) = \exp\left(-1.3\left(\frac{P}{1-|P|} + 0.5\right)^2\right)$$

and Nu_{FL} is a correlation for pure forced laminar convection, given by VDI Heat Atlas [36]. As suggested by Aicher and Martin [1], physical properties are calculated at film temperature. The comparison between present experimental data and calculations obtained using the Aicher and Martin method is reported in Fig. 8, where the ratio between the experimental and the calculated Nusselt number is plotted versus the buoyancy parameter, Bo. The agreement is not so good, as a general overprediction of experimental data appears, even for pure forced convection (Bo around 0.01), which means the inadequacy of the Gnielinski correlation in predicting pure forced convection data. These latters are well predicted using the well-known Dittus–Boelter correlation. Possible causes of the systematic overprediction provided by the Aicher and Martin method may be thought as follows:

(a) Nusselt number for forced laminar convection Nu_{FL}, used in equation (8), is generally used out of range, as most, if not all, of present data are in the turbulent flow regime. The presence of this number in equation (8) may be therefore questionable.
(b) The effect of L/D, which has been tested of significant importance, is not directly accounted for in equation (8), and this may be a limit, even though the overprediction is almost general.
(c) The correction function for the aided mixed convection effect with respect to downflow (right-hand side of equation (8)) is a function also of correlations used for pure natural and forced convection. The latter might be unable to predict data outside the mixed convection region (as shown in Fig. 8). Of course, this affects the accounting for the heat transfer reduction in upflow vs downflow. On the other hand, changing these correlations (Nu_{for} and Nu_{nat}) would result in a complete change of the aided correction function, upsetting the original formulation. In turn, it would be a non-sense changing such correlation only in the left-hand side of equation (8), without any modification in the correction function.

5.2. New method

On the basis of the present experimental data, and because of the drawbacks presented by existing methods for the prediction of the heat transfer coefficient in upward mixed convective flow in a vertical channel, we propose a new method which shows some similarities with that proposed by Aicher and Martin[1], although developed independently of it [37]. We start from equation (5), proposed by Churchill [9], valid for downflow. In equation (5) $Nu_{\mathrm{m,df}}$ is calculated using equations (1) and (2) for pure forced and natural convection, respectively. If we plot present data in terms of the ratio between the experimental Nusselt number, Nu_{exp}, and that given by equation (5), $Nu_{\mathrm{m,df}}$, versus the buoyancy parameter, Bo, we get the result shown in Fig. 9, where, in order to avoid the formation of a black spot, we selected only data for $L/D = 9.6$ and 15.4. The ratio $Nu_{\mathrm{exp}}/Nu_{\mathrm{m,df}}$ is equal to 1 for $Bo << 1$ (pure forced convection), and for $Bo >> 1$ (pure natural convection), while in between, as expected, we observe a lower heat transfer coefficient in upflow with respect to downflow. In this way, we obtain the result to anchor the calculation to what is known in terms of pure natural and forced convection, where we are able to well predict the relative heat transfer coefficients. Figure 9(a) reports a logarithmic abscissa scale, while Fig. 9(b) has a linear abscissa scale, to show more clearly the tendency to unity of $Nu_{\mathrm{exp}}/Nu_{\mathrm{m,df}}$ for $Bo >> 1$, and $Bo << 1$.

The deviation of experimental data (upflow) from what predicted using equation (5) (downflow) is at a maximum around $Bo = 1$, where the laminarization effect has been observed to be the largest. Such a deviation may be thought as following a Gaussian-type curve, depending only on buoyancy and geometric effects. We therefore propose a correction function for upflow according to a Gaussian curve equation of the following type:

$$\frac{Nu_{\mathrm{m,up}}}{Nu_{\mathrm{m,df}}} = \psi; \quad \psi = 1 - a\exp\left\{-0.8\left[\log\left(\frac{Bo}{b}\right)\right]^2\right\} \quad (9)$$

Constants a and b are obtained with a best-fit procedure through experimental data, finding:

$$a = 0.36 + 0.0065\frac{L}{D} \quad b = 869\left(\frac{L}{D}\right)^{-2.16} \quad (10)$$

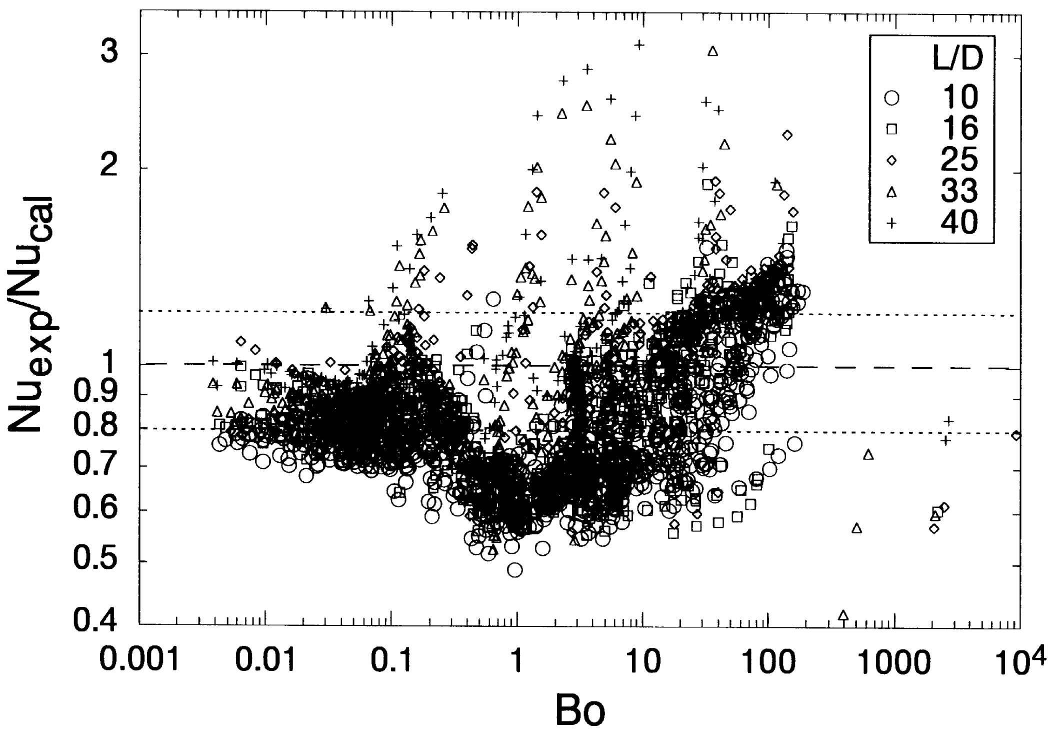


Fig. 8. Predictions of present experimental data obtained using the Aicher and Martin [1] method.

In the Gaussian curve, $1-a$ represents the ordinate of the Gaussian vertex, while b is its abscissa. It is therefore also of interest the trend of the $1-a$ and b parameters versus the channel geometry, L/D, which has been reported in Fig. 10. Figure 10(a) shows the $1-a$ parameter as a function of L/D. The variable a corresponds to the maximum deviation between the experimental heat transfer and that given by equation (5), and then the maximum effect of the laminarization on the heat transfer in upflow mixed convection, with respect to downflow. As expected, see Fig. 5 and Aicher and Martin [1], the laminarization effect is small for low L/D values, while it increases as the flow thermal stabilisation increases. Figure 10(b) shows instead the b parametric trend.

A first comparison of experimental data with predictions obtained using equation (9) is reported in Fig. 11, where the ratio between the experimental Nusselt number and that obtained using equation (5), $Nu_{m,df}$, i.e. the left-hand side term of equation (9), is plotted vs the buoyancy parameter, *Bo*, together with the ψ function, for different values of L/D.

Also here, the double representation of the logarithmic and linear abscissa scale is used to show the comparison more clearly. The agreement is quite good, the proposed method showing a good precision and accuracy for buoyancy and geometric effects.

A more general view of the comparison between present experimental data and predictions obtained using the new proposed method is shown in Fig. 12, where the ratio between experimental and calculated Nusselt number is plotted versus the buoyancy parameter, *Bo* (top graph), and the heat flux q'' (bottom graph), respectively. The comparison is quite good, with most of data points predicted within $\pm 20\%$, with a standard deviation of 17%, and no systematic error as a function of *Bo* and q'' is observed.

In order to verify the performance of the newly developed method with different data than that of the present, we tried to predict Aicher and Martin [1] data. First of all, we used equation (9), together with equations (1), (2) and (5), i.e., exactly the same equations used for the prediction of the present data. The prediction is reported in Fig. 13, where the calculated-to-experimental Nusselt number is plotted versus the buoyancy parameter. The calculation refers to full (grey) triangles and circles, for $L/D = 25$ and 54. As Aicher and Martin [1] used different correlations to predict their data in pure forced and natural convection, we accomplished a second calculation using equation (9) in combination with the equations used by Aicher and Martin for pure forced and natural convection. In other words we used the Gnielinski [35] correlation for forced convection, and equation (2), slightly modified by Aicher and Martin [1] for natural convection. The result of this second calculation is plotted

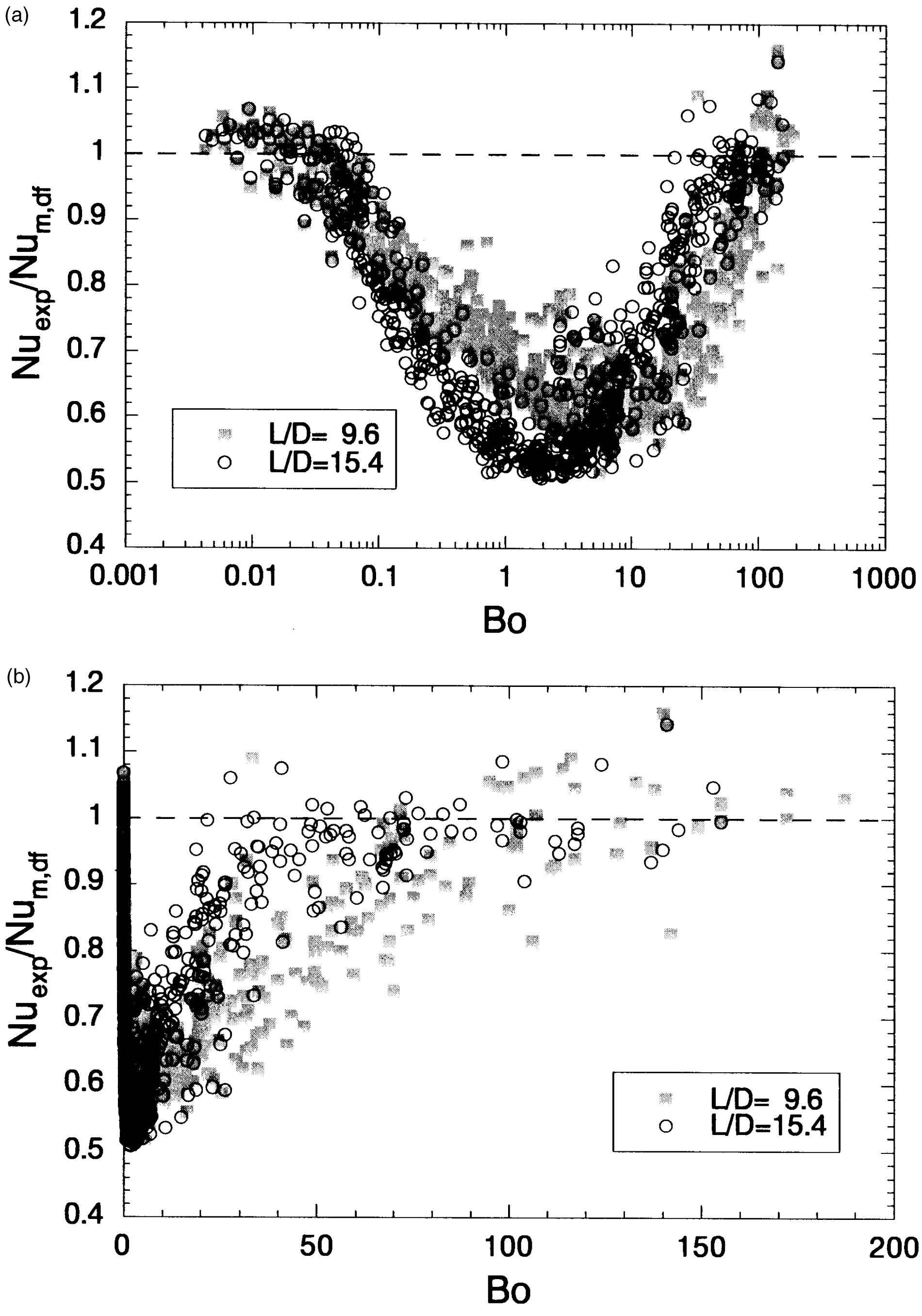


Fig. 9. Experimental-to-calculated Nusselt number using equation (5), valid for downflow, vs buoyancy parameter, for $L/D = 9.6$ and 15.4, using a logarithmic abscissa (a), and a linear abscissa (b).

in Fig. 13, and refers to empty symbols (triangles and circles).

Although the first calculation gives a good prediction in the mixed convective region of Aicher and Martin data, $0.1 < Bo < 10$, a general underprediction is exhibited in the forced convection region, $Bo < 0.1$. Using the cor-

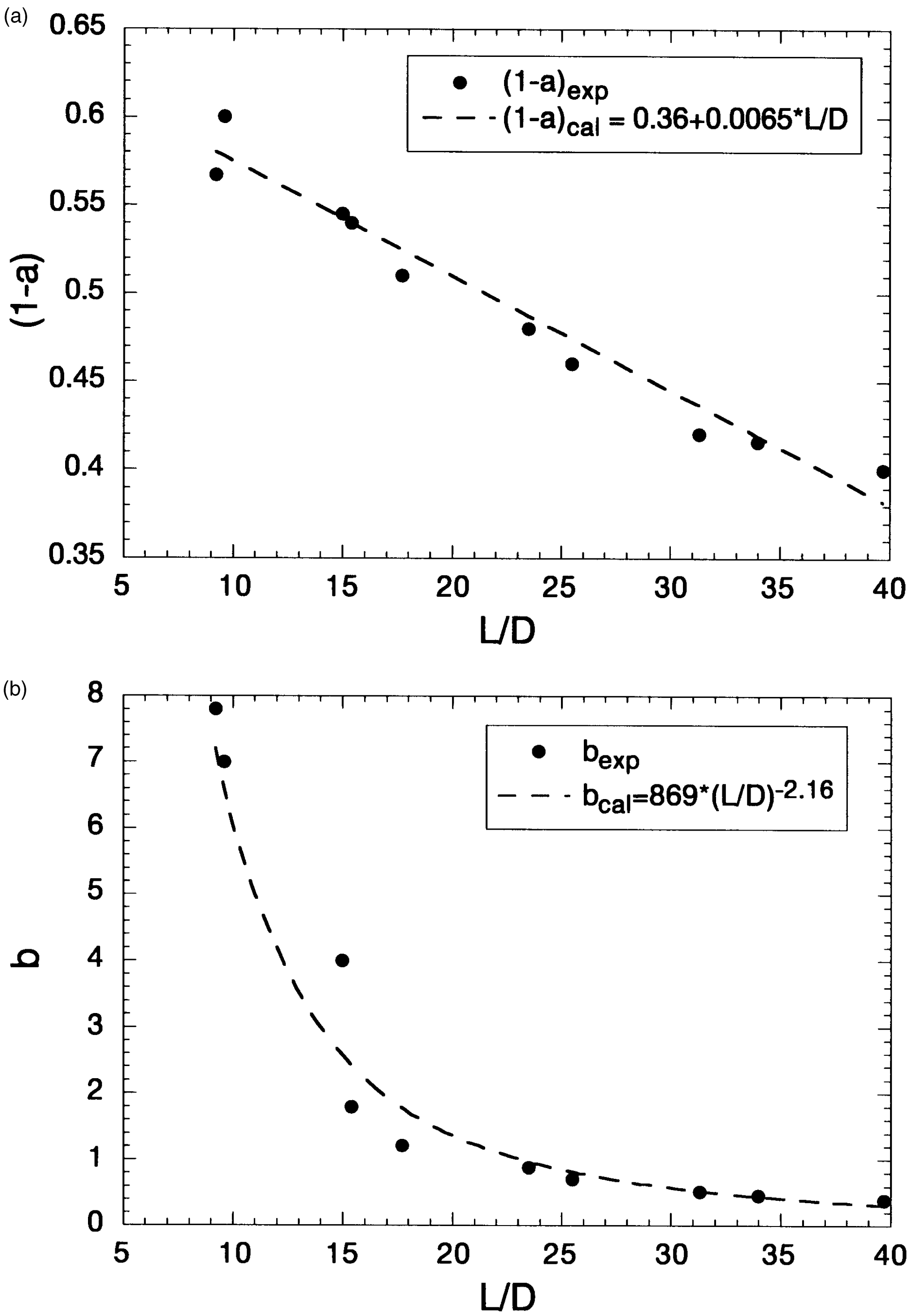
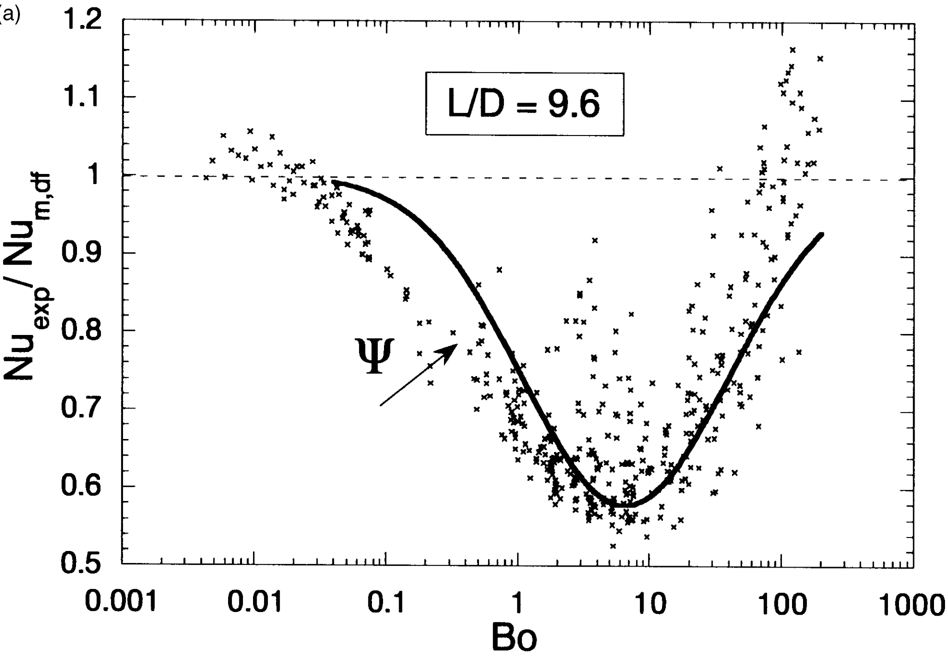


Fig. 10. Representation of a (a) and b (b) parameters in equation (9) vs L/D.

relations suggested by Aicher and Martin for pure forced and natural convection, which may depend on the characteristics of the pipes employed in the experiment, the general result is very good for all the flow regimes, with most of the data predicted within $\pm 20\%$.

Therefore, the newly proposed method for the calculation of the heat transfer coefficient in vertical upflow turbulent mixed convection beyond its capacity to provide a good prediction of present experimental data, can be also used to predict data from other authors in differ-

(a)



(b)

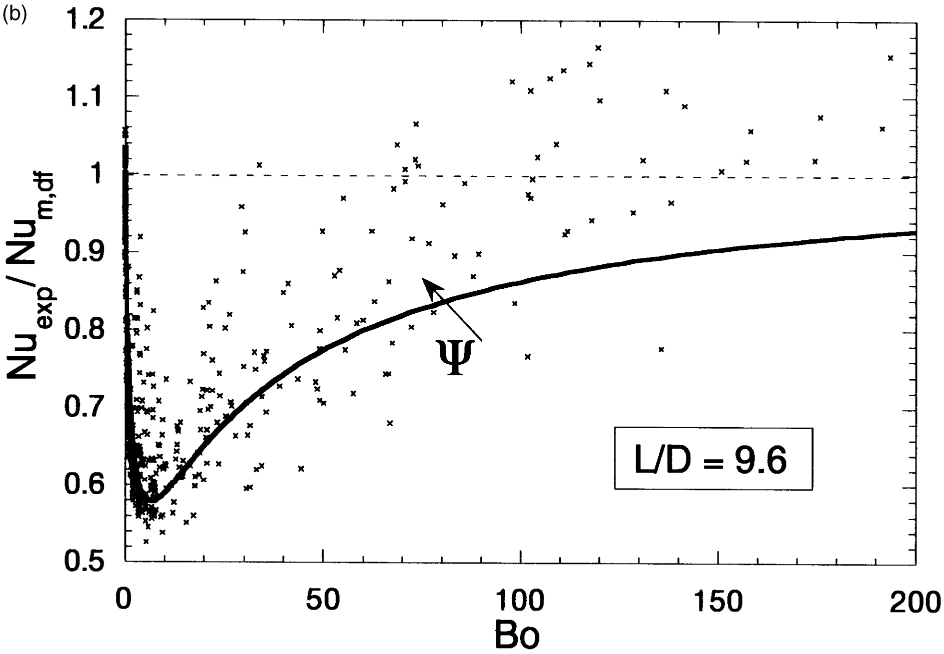
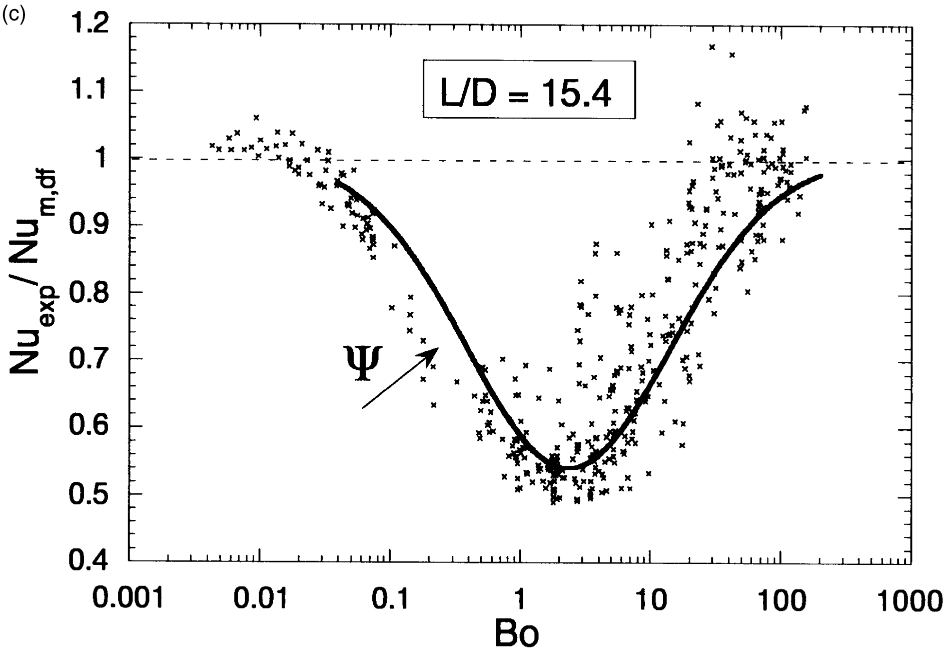


Fig. 11. Experimental-to-calculated Nusselt number using equation (5), valid for downflow, vs buoyancy parameter, together with the corrective function ψ, described in equation (9), for $L/D = 9.6$ using a logarithmic (a) and a linear abscissa (b), and for $L/D = 15.4$ using a logarithmic (c) and a linear abscissa (d).

(c)



(d)

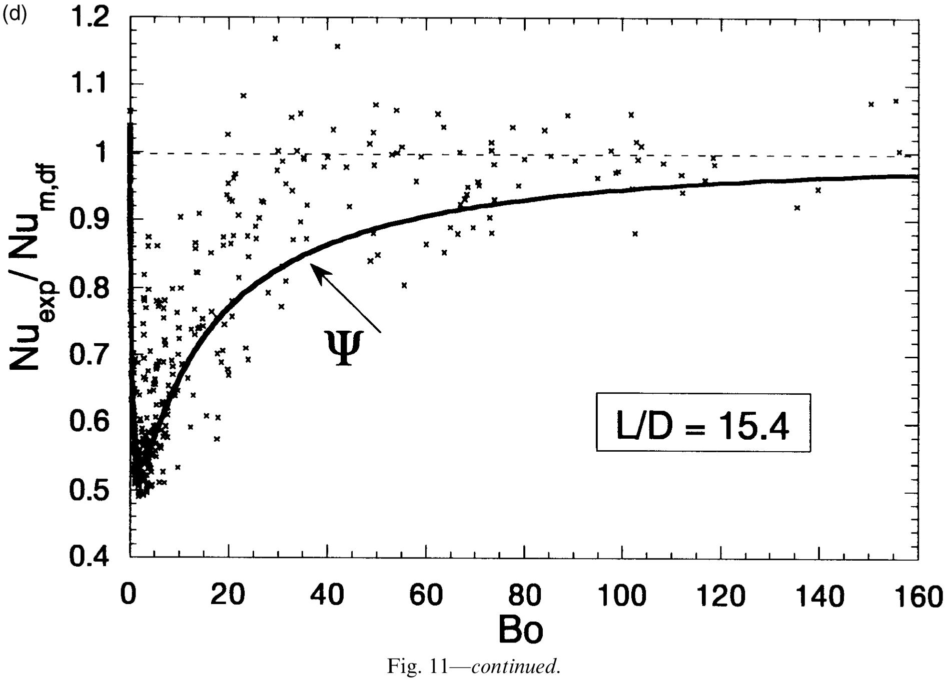


Fig. 11—*continued.*

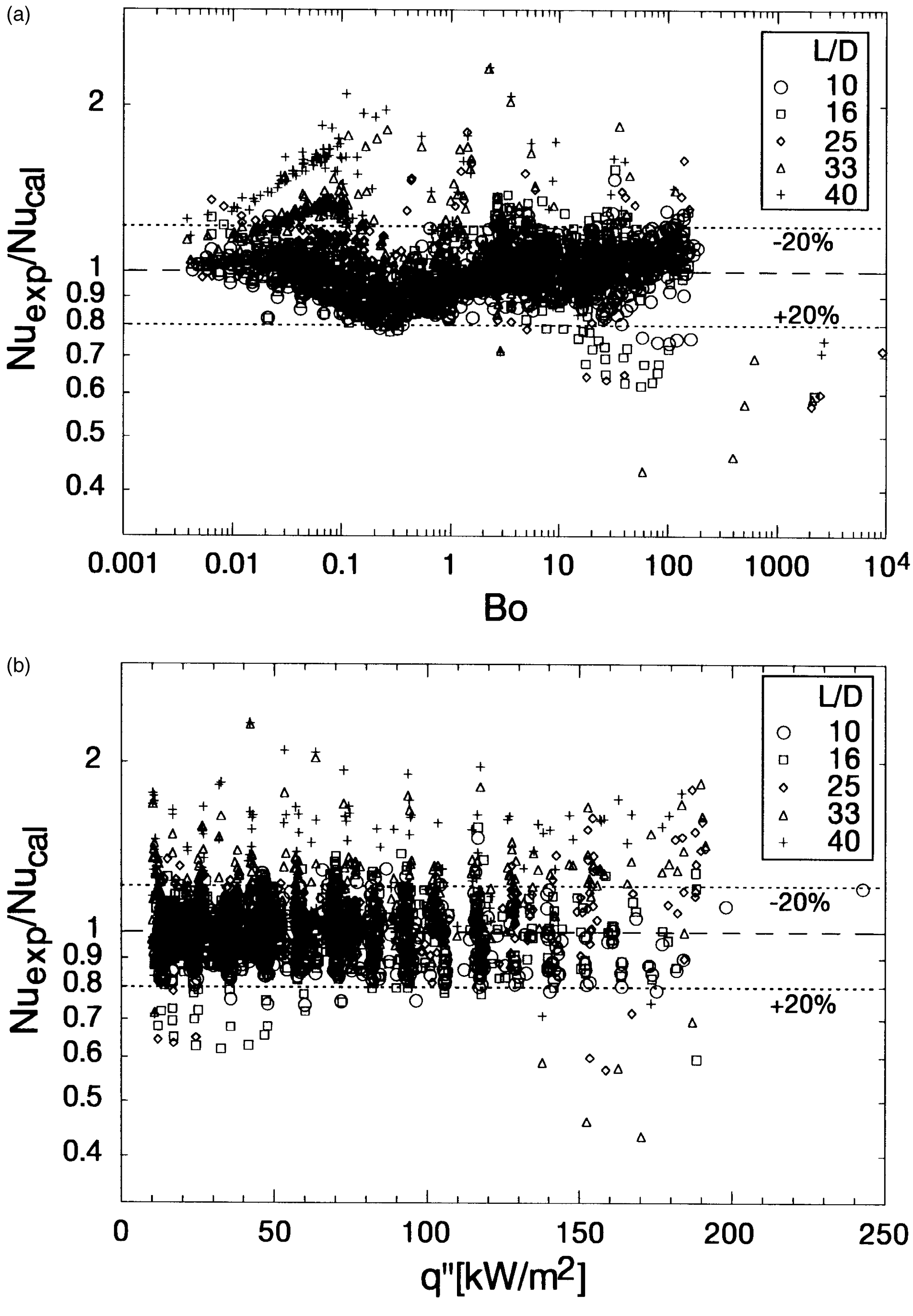


Fig. 12. Predictions of present experimental data obtained using equation (9) vs buoyancy parameter (a) and heat flux (b).

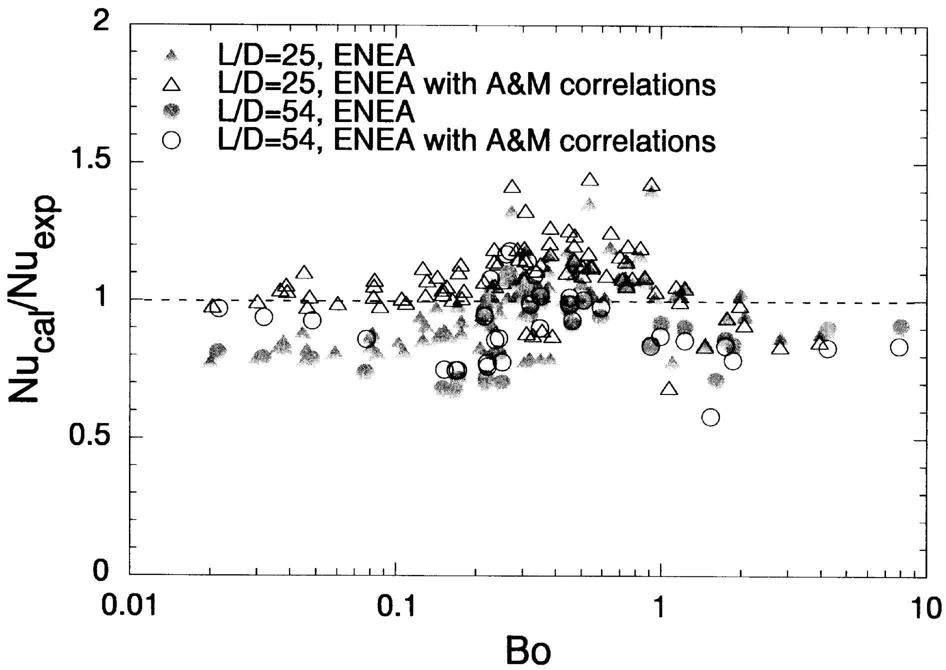


Fig. 13. Predictions of Aicher and Martin [1] data obtained using equation (9) vs buoyancy parameter.

ent geometries and different channels, bypassing the main drawbacks presented by the Aicher and Martin [1] method, with special regard to points, L/D effect, and replacement of pure forced and natural convection correlations, reported in Section 5.1.

6. Conclusions

New experimental data of heat transfer in turbulent upward mixed convection of water in a vertical channel have been performed, with special emphasis on L/D effect.

In aided mixed convective flow, the heat transfer decreases as L/D increases, the reduction showing a maximum when the buoyancy parameter, *Bo*, is around unity. Under these conditions, the laminarization effect is maximum.

In view of the scarce availability of reliable design tools (apart from a complex numerical approach), a new method for the prediction of the heat transfer coefficient in aided mixed convection is presented.

The new method provides a good prediction of present experimental data, taking into account the L/D and laminarization effects. The data of Aicher and Martin [1] are also predicted by the new method. Besides, whenever the method has to be used to predict data, for which different correlations for the calculation of pure forced and natural convection are necessary, this can be easily done simply without any further modification of the expression which accounts for buoyancy and geometric effects.

References

[1] T. Aicher, H. Martin, New correlations for mixed turbulent natural and forced convection heat transfer in vertical tubes. International Journal of Heat and Mass Transfer 40 (1997) 3617–3626.
[2] B.S. Petukhov, L.D. Nolde, Heat Transfer during the viscous–gravitational flow of a liquid through pipes. Teploenergetika 6 (1959) 72–80.
[3] B. Metais, Criteria for mixed convection, HTL-Techn. Report no. 51; Heat transfer Lab., University of Minnesota, 1963.
[4] B.S. Petukhov, Turbulent flow and heat transfer in pipes under considerable effect of thermo-gravitational forces. In: D.B. Spalding, N. Afgan (Eds.), Heat Transfer and Turbulent Buoyant Convection, vol. 2. Hemisphere, New York, 1977.
[5] G.T. Polley, Comprehensive correlating equations for convective heat transfer in circular tubes; HTFS RS 384; National Engineering Laboratory/HTFS 9, Glasgow, 409421, 1981.
[6] W. Aung, Mixed convection in internal flow. In: S. Kakaç, R.K. Shah, W. Aung (Eds.), Handbook of Single Phase Convective Heat Transfer, Wiley, New York, 1987.
[7] L.W. Swanson, I. Catton, Surface renewal theory for turbulent mixed convection in vertical ducts. International Journal of Heat and Mass Transfer 30 (1987) 2271–2279.

[8] J.D. Jackson, M.A. Cotton, B.F. Axcell, Studies of mixed convection in vertical tubes. International Journal of Heat and Fluid Flow 10 (1989) 2–15.

[9] S.W. Churchill, Combined free and forced convection in channels. In: G.F. Hewitt (Ed.), Handbook of Heat Exchanger Design, Begell–House, 1992.

[10] D. Joye, Comparison of aiding and opposing mixed convection heat transfer in a vertical tube with Grashof number vanation. International Journal of Heat and Fluid Flow 17 (1996) 96–101.

[11] D. Joye, Comparison of correlations and experiment in opposing flow, mixed convection heat transfer in a vertical tube with Grashof number vanation. International Journal of Heat and Mass Transfer 39 (1996) 1033–1038.

[12] A. Watzinger, D.G. Johnson, Wärmeübertragung von Wasser an Rohrwand bei senkrechter Strömung im Übergangsgebeit zwischen laminarer und turbulenter Strömung. Forschung im Ing.-Wesen 10 (1939) 182–196.

[13] R.H. Norris, M.W. Sims, A simplified heat transfer correlation for semi-turbulent flow of liquids in pipes. Transactions AIChE 38 (1942) 469–492.

[14] B. Metais, Wärmeubergang am waagrechten Rohr: Report no. 205A. Inst. fiir therm. Verfahrenstechnik, University of Karlsruhe, 1957.

[15] L.S. Herbert, U.J. Sterns, Heat transfer in vertical tubes—interaction of forced and free convection. Chemical Engineering Journal 4 (1972) 46–52.

[16] B.P. Axcell, W.B. Hall, Mixed convection to air in a vertical pipe. Proceedings of 6th International Heat Transfer Conference, vol. 1, Toronto, 1978, pp. 37–42.

[17] P.E. Saylor, D.D. Joye, Pressure drop in mixed convection heat transfer in vertical tube. Research proposal. Villanova University, Villanova, PA, 1988.

[18] A. Temu, W. Blumberg, V. Gnielinski, H. Martin, Überlagerte erzwungene und freie Konvektion in einem senkrechten Rohr. Wiss. Abschlußber. 26. International Seminar d. Univ. Karlsruche, University of Karlsruche, 1989, pp. 49–58.

[19] L. Prandtl, Führer durch die Strömungslehre. 3rd ed. Vieweg, Braunschweig, 1942.

[20] B. Metais, E.R.G. Eckert, Forced, mixed, and free convection regimes. Journal of Heat Transfer 86 (1964) 295–296.

[21] A. Steiner, On the reverse transition of a turbulent flow under the action of buoyancy forces. Journal of Fluid Mechanics 47 (1971) 503–512.

[22] S.W. Churchill, Comprehensive correlating equations for heat, mass and momentum transfer in fully developed flow in smooth tubes. Industrial Engineering and Chemistry Fundamentals 15 (1972) 789–800.

[23] S.W. Churchill, H.H.-S. Chu, Correlating equations for laminar and turbulent free convection from a vertical plate. International Journal of Heat and Mass Transfer 18 (1975) 1823–1829.

[24] J.D. Jackson, J. Fewster, Enhancement of turbulent heat transfer due to buoyancy for downward flow of water in vertical tubes. In: D.B. Spalding, N. Afgan (Eds), Heat Transfer and Turbulent Buoyant Convection, vol. 2. Hemisphere, New York, 1977.

[25] J.D. Jackson, W.B. Hall, Influences of buoyancy on heat transfer of fluids in vertical tubes under turbulent conditions. In: S. Kakaç, D.B. Spalding (Eds), Turbulent Forced Convection in Channels and Bundles, vol. 2, Hemisphere, New York, 1979.

[26] A. Bejan, Convection Heat Transfer. Wiley, New York, 1984.

[27] A.F. Polyakov, S.A. Shindin, Development of turbulent heat transfer over the length of vertical tubes in presence of convection of a liquid in a vertical tube. International Journal of Heat and Mass Transfer 31 (1988) 987–992.

[28] H. Tanaka, S. Maruyama, S. Hatano, Combined forced and natural convection heat transfer for upward flow in a uniformly heated, vertical pipe. International Journal of Heat and Mass Transfer 30 (1987) 165–174.

[29] B. Weigand, H. Beer, Wärmeübertragung in einem axial rotierenden, durchströmten Rohr im Bereich des thermischen Einflaus. Wärme Stoffübertr 24 (1989) 191–202.

[30] M.A. Cotton, J.D. Jackson, Vertical tube air flows in the turbulent mixed convection regime calculated using a low-Reynolds-number k–ε model. International Journal of Heat and Mass Transfer 33 (1990) 275–286.

[31] W.B. Hall, Heat transfer near the critical point. Advances in Heat Transfer 7 (1971) 1–86.

[32] W.B. Hall, J.D. Jackson, Laminarization of a turbulent pipe flow by buoyancy forces. ASME Paper 69-HT-55, 1969.

[33] B.E. Launder, B.I. Sharma, Application of the energy-dissipation model of turbulence to the calculation of flow near a spinning disc. Lett. Heat Mass Transfer 1 (1974) 131–138.

[34] R.C. Martinelli, L.M.K. Boelter, An analytical prediction of superimposed free and forced viscous convection in a vertical pipe. Univ. Calif. Publ. in Eng. 5(2) (1942) 23–58.

[35] V. Gnielinski, Ein neues Berechnungsverfahren für die Wärmeübertragung im Übergangsbereich zwischen laminarer und turbulenter Rohrströmung. Forschung im Ingenieurwesen-Engineering Research 61 (1995) 240–248.

[36] VDI Heat Atlas. VDI-Verlag, Düsseldorf, 1993.

[37] G.P. Celata, A. Chiaradia, M. Cumo, F. D'Annibale, Heat transfer enhancement by air injection in forced and mixed convective flow of water. Proceedings of 15th UIT National Heat Transfer Conference, vol. II, 1997, pp. 557–570.