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An empirical investigation into the external heat transfer of a U-bend in cross-flow

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Abstract

An experimental investigation into the global external heat transfer coefficients for circular U-bends (180° bends) in a cross-flow air stream was performed using six different curvature ratios (δ) covering a Reynolds number range of 3000–80 000 while imposing a uniform-wall-temperature (UWT) boundary condition. The measurements specifically focused on a zero-degree angle-of-incidence, in which the external flow direction is parallel to the bend plane. A straightforward method to measure the global external heat transfer that is not dependent on geometry but limits the testing to UWT was employed. The external heat transfer coefficients for circular U-bends in cross-flow were found to be consistently higher than those for corresponding straight cylinder flows. Also, a strong dependence on the curvature ratio, δ, was observed. Measurements presented demonstrate a peak enhancement in the external heat transfer for curvature ratios close to an optimum value of $\delta_{opt} \approx 0.23$. A geometrical analysis was performed to explain the experimental finding of a distinct optimum curvature ratio (δ_{opt}) as a self-inflicted wake effect. U-bend Nusselt numbers were found to be over four times larger than those for straight tubes of identical diameter and Reynolds number. An engineering correlation for the global external Nusselt number is offered. © 1998 Elsevier Science Ltd. All rights reserved.

Nomenclature

$C_{1,2,3}$ arbitrary constants
c_p constant pressure specific heat
d outer pipe diameter
$f(\delta)$ arbitrary function of δ
$g(\delta)$ arbitrary function of δ
h convection coefficient
j_{fg} heat of vaporization
k fluid thermal conductivity
L separation distance, or length
$\dot{m}$ condensation accumulation rate
Nu Nusselt number (hd/k)
Pr Prandtl number ($\mu c_p/k = \nu/\alpha$)
R radius of bend curvature
Re Reynolds number ($u\,d/\nu$)
s seconds
T temperature
U fluid velocity.

Greek symbols
α thermal diffusivity
δ curvature ratio (ratio of pipe radius to bend radius)
θ bend-plane arc angle
μ dynamic viscosity
ν kinematic viscosity
Π Buckingham Pi group
ρ density.

Subscripts
air air
amb ambient
bend bend pipe section
d diameter
steam steam
straight straight pipe section
o ambient or free stream
opt optimum.

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1. Introduction

The stimulant for the tests now reported was the need to determine the external heat transfer coefficient for a tubular ('serpentine') heat exchanger. U-bends are commonly found in serpentine heat exchangers, which are typically used in heating systems such as residential furnaces. Typical bend curvature ratios, δ, in these appliances range from 0.10 to 0.30. In a residential furnace the external (circulating) air moving across these U-bends will generally range between 3 and 6 m s^{-1}. This flow range gives a Reynolds number value between 5000 and 17 000 based on the external diameter of the pipe. A typical serpentine passage will contain three to five bends. These bends can account for up to 20% of the total running length of a serpentine passage. Therefore, U-bends have a significant influence on serpentine heat exchanger performance.

Data found in the literature highlights that the increase of the internal heat transfer for coiled-pipe flows could be three times the value of an equivalent straight pipe flow. Most studies have concentrated on continuously coiled-pipe flows. Studies that specifically focused on the internal heat transfer for U-bends are rare [1–3]. Although there are no reliable engineering correlations for flows through a sharp ($\delta > 0.1$) U-bend, a particular correlation for coiled pipe flows, given in Manafzadeh et al. [4], provides an adequate engineering estimate. However, there are no investigations in the literature reporting the external heat transfer across a U-bend. Representative results will now be presented.

The Buckingham Π theorem was used to derive the non-dimensional groups governing external heat transfer of a heated U-bend. These dimensionless groups were then used to reduce the experimental results. Since the heat transferred from an immersed body to an external ambient fluid is dependent upon the free stream velocity, bend geometry, and fluid properties, the heat transfer coefficient $h \sim h(U_o, d, R, \rho, k, c_p, \mu)$. The four Π groups identified by using U_o, d, ρ, and μ as the repeating variables are as shown below.

$$\Pi_1 = \frac{h \cdot d}{k}, \quad \Pi_2 = \frac{R}{d}, \quad \Pi_3 = \frac{\rho \cdot U_o \cdot d}{\mu}, \quad \Pi_4 = \frac{v}{\alpha} \tag{1}$$

Recognizing some familiar dimensionless groupings, the Nusselt number was expected to depend on the following parameters.

$$Nu_d = Nu_d(Re_d, \delta, Pr) \tag{2}$$

Both the Nusselt number and Reynolds number have the pipe diameter as the characteristic length.

2. Test procedure

Given the Nusselt dependence expected in equation (2) a test-rig was constructed capable of measuring external heat transfer rates for U-bends submerged in a cross-flow air stream. The external air stream ($Pr \sim 0.7$, 298 K) was delivered by a wind tunnel, which provided a uniform velocity profile across the U-bends. Therefore, the Prandtl dependence given in equation (2) was not identified and the results collected were valid for air only; accordingly, the behavior identified was $Nu_d = (Nu_d Re_d, \delta, Pr = 0.7)$. The external Reynolds number (based on diameter), Re_d, was controlled by varying the external air stream flow velocity over several U-bends fabricated from different sized diameter pipes. A brief account of the testing program is provided next. See Harris [6] for a detailed description of the test facility.

2.1. Wind tunnel facility

The wind tunnel used could deliver 54 m s^{-1} airflow with 0.1% free stream turbulence levels as described by Brown [5]. Therefore, the effects of free stream turbulence levels on the external heat transfer rate were not investigated. For the tube diameters used,which ranged from 1.25 to 4.45 cm, Reynolds numbers (based on the tube outer-diameter) as high as 10^5 were obtained. The wind tunnel test section was 60 cm high by 45 cm wide by 154 cm long. A pitot probe was placed 30 cm from the test section entrance and was 7.6 cm off the floor of the tunnel. The probe was used to measure the external flow velocity by using a MKS Baratron® pressure differential transducer.

2.2. U-bend test specimens

All U-bend sections fabricated for testing contained adjoining straight pipe sections needed for structural support. Since the purpose of the measurements was to determine the heat transfer rate for the curved section of the bend (i.e. the bend itself with no adjoining straight pipe sections) the influences from the straight pipe sections needed to be quantified. One bend section ($\delta = 0.292$) was fabricated that physically isolated the condensate formed in the curved section from the condensate formed in the straight pipe sections. That is, the condensation rate in the curved section of the U-bend was directly measured. Also, the condensation rates in the straight pipe sections were measured to benchmark the heat transfer rates associated within these straight pipe sections. All other U-bend test sections were fabricated to collect the condensate from the curved section in addition to the two adjoining straight pipe sections. The estimated contribution from the straight sections was then subtracted from the total amount of condensate collected.

In all, seven separate U-bends were constructed for this investigation. A U-bend fabricated from 4.45 cm outer diameter aluminized steel with a bend radius of 7.62 cm ($\delta = 0.292$) was designed to investigate the heat transfer rates at incidence angles of 0, 45 and 90° for

Reynolds numbers ranging from 3×10^3 to 8×10^4. For 0° (and 90°) angle-of-incidence the external flow impinged on the pipe surface from a directional parallel (and perpendicular) to the bend plane. The curved section of the U-bend ($0 \leqslant \theta \leqslant 180$) for this test specimen was physically separated with fiberglass insulators from two attached 17.8 cm long straight pipe sections. Isolators separated the curved to ensure that condensate created in the curved section was separated from the condensate created in the two straight pipe sections adjoining the bend. Separate condensate drains were attached to the curved section and the two straight pipe sections to segregate the condensate collected in the curved section of the U-bend. This was necessary since this test specimen was also used to measure the heat transfer over a circular cylinder to validate the experimental methodology. When situated in the wind tunnel, the test section was inclined at a slight angle to allow condensate to flow towards the rear of the test section where it was collected and measured. The end of the two straight sections also contained various access ports. These ports were used for the steam inlet and access for the vacuum pump and pressure gauge. Also, a 45.7 cm type K thermocouple used to measure the steam temperature was inserted at the end of a straight section, which penetrated into the inner volume of the curved section.

Six other U-bend test sections ($\delta = 0.100, 0.103, 0.167, 0.200, 0.250$ and 0.417) were designed to investigate the influence of bend curvature on the rate of heat transfer for 0° angle-of-incidence. These pipe bends were all made from standard refrigeration copper tubing with diameters of 1.91, 1.59, 1.27 and 0.95 cm. The bend diameters for these tubes were 18.50, 15.88, 7.62 and 4.76 cm, respectively, giving curvature ratios of 0.103, 0.100, 0.167 and 0.200. The 1.59 cm pipe bend was twice modified to have bend ratios of 0.250 and 0.417 by respectively changing the bend diameter to 3.18 and 1.90 cm. The straight sections on these bends were 7.62 cm long for the two larger diameter tubes and 5.08 cm long for the two smaller diameter tubes. The bends were all inclined slightly so that the condensate would flow to the rear of the test section. Unlike the $\delta = 0.292$ U-bend, the straight pipe sections of these U-bends were not isolated from the curved section. Therefore, the condensate from the curved section and both straight sections was collected and the heat transfer from both straight sections and the curved section was inferred from the amount of condensate. The contribution from the straight sections was easily estimated by use of various correlations for flow over a cylinder and subtracted from the total to obtain that attributable to the curved section only. The correlation offered by Zhukauskas [8] was used for both the leading and trailing cylinders; except for the $\delta = 0.417$ U-bend where the correlation offered by Kostic and Oka [11] was used for the trailing cylinder to account for significant (leading cylinder) wake effects.

2.3. *Methodology and data reduction*

The method used to measure the heat transfer occurring on the (U-bend) outer surface involved measuring the rate of condensing saturated steam along the (U-bend) inner surface under steady conditions. The raw data collected included the accumulation rate of the condensate, $\dot{m}$, along with the internal saturated steam temperature, T_{steam}, and the external air stream (ambient) temperature, T_{amb}. The heat of vaporization of the steam, j_{fg}, and the conductivity of the air, k_{air}, were (respectively) derived from the temperature measurements. The external Nusselt number was found by reducing the measured and derived data as shown in equation (3).

$$Nu = \frac{\dot{m} \cdot j_{\text{fg}}}{k_{\text{air}} \cdot \pi^2 \cdot R \cdot (T_{\text{steam}} - T_{\text{amb}})} \tag{3}$$

The data collection and reduction procedure was as follows. A container of water (used as the steam source) was heated to around 115°C. The inner bend volume was first evacuated and then filled with the saturated (pure) steam. The inner volume, and hence inner surface, was then held constant at the saturation temperature as a cross-flow air-stream flowed across the outer surface. As air flowed across the external U-bend surface condensation on the inner surface occurred. The rate of condensation throughout the inner surface was proportional to the heat transfer rate from the external surface to the ambient air-stream. This simplified method of data reduction to derive the Nusselt number for the external bend surface was possible since the conduction resistance to heat transfer in the bend material was insignificant to the convective resistance to heat flow on the outer surface. For instance, for a 4.45 cm outer diameter tube exposed to a convection coefficient of 100 W m^{-2} K^{-1} the conduction resistance was less than 0.3% of the convection resistance. The inner surface heat transfer could also be ignored since condensing water vapor at 1 atm produces convection coefficients in the range of 10^3–10^4 W m^{-2} K^{-1}. Conditions were allowed to become steady at every external airflow level. The condensate was then collected over a 10 min interval and the mass was measured directly using a scale sensitive to 0.01 g. The internal (saturated steam) pressures and temperatures were measured directly. The heat of vaporization was computed from the temperature measurement by use of a correlation derived from steam table data. The film temperature was computed based on the steam temperature and the ambient temperature. This film temperature was then used to find the external air-stream thermal conductivity using a curve-fit constructed from air table data.

3. Validation of test methodology

The experimental methodology was verified by measuring the heat transfer for a heated straight cylinder in cross-flow. The results were then compared to familiar engineering correlations in the literature including Hilpert [7], Zhukauskas [8] and Churchill and Bernstein [9]. The reduced results are shown in Fig. 1. A least squares (statistical) fit of the measurements produced equation (4).

$$Nu = 0.321 \cdot Re_d^{0.58} \quad (4)$$

The straight pipe measurements lie closest to the correlation reported by Zhukauskas [8], equation (5), for the boundary condition of uniform heat flux.

$$Nu_{\text{straight}} = 0.26 \cdot Re_d^{0.6} \cdot Pr^{0.37} \quad (5)$$

The data showed a slight positive bias in the measurement technique. The overall uncertainty in the measurements shown in Fig. 1 was 5.8% for the Reynolds number and 11.3% for the Nusselt number. The overall uncertainty level included both precision and bias errors for the 95% confidence level. However, there appeared an additional bias in the measurements not accounted for in the uncertainty analysis. The most probable source of the positive bias was condensation occurring due to parasitic heat losses. These parasitic losses are difficult to predict and were not accounted for in the uncertainty analysis. In an effort to minimize these losses all surfaces outside the wind tunnel were packed with fiberglass insulation. However, even with this positive bias the measurements all lie near the Zhukauskas correlation within their uncertainty range. Therefore, the measurement technique should have adequately predicted, within the stated uncertainty limits, the external heat transfer behavior of curved U-bend surfaces.

4. Test results

The application for the results presented is residential furnace tubular heat exchangers with U-bends having curvature ratios between 0.1 and 0.3. A family of test specimens was designed to measure heat transfer for bends with curvature ratios between 0.100 and 0.417. Figure 2 shows the reduced measurements using a curvature ratio of 0.292 at three angles-of-incidence. A 0° angle-of-incidence indicates that the external flow velocity vector was parallel to the bend-plane. The reduced measurements (Nu_{bend}) are presented with respect to the reduced values for the straight pipe sections (Nu_{straight}) made during the validation phase (Fig. 1). For 0° angle-of-incidence the external heat transfer for the U-bend was two to three times the rate observed for a corresponding straight cylinder in cross-flow ($Nu_{\text{bend}}/Nu_{\text{straight}}$). There also appeared a distinct Reynolds number dependence on the level of this increase. No general conclusions about the influence of the angle-of-incidence on the heat transfer rate could be inferred from these measurements since only three angles were tested. However, it was clearly evident that the U-bend at 90° angle-of-incidence behaved similarly to a cylinder in cross-flow.

Figures 3–9 show the reduced measurements for all

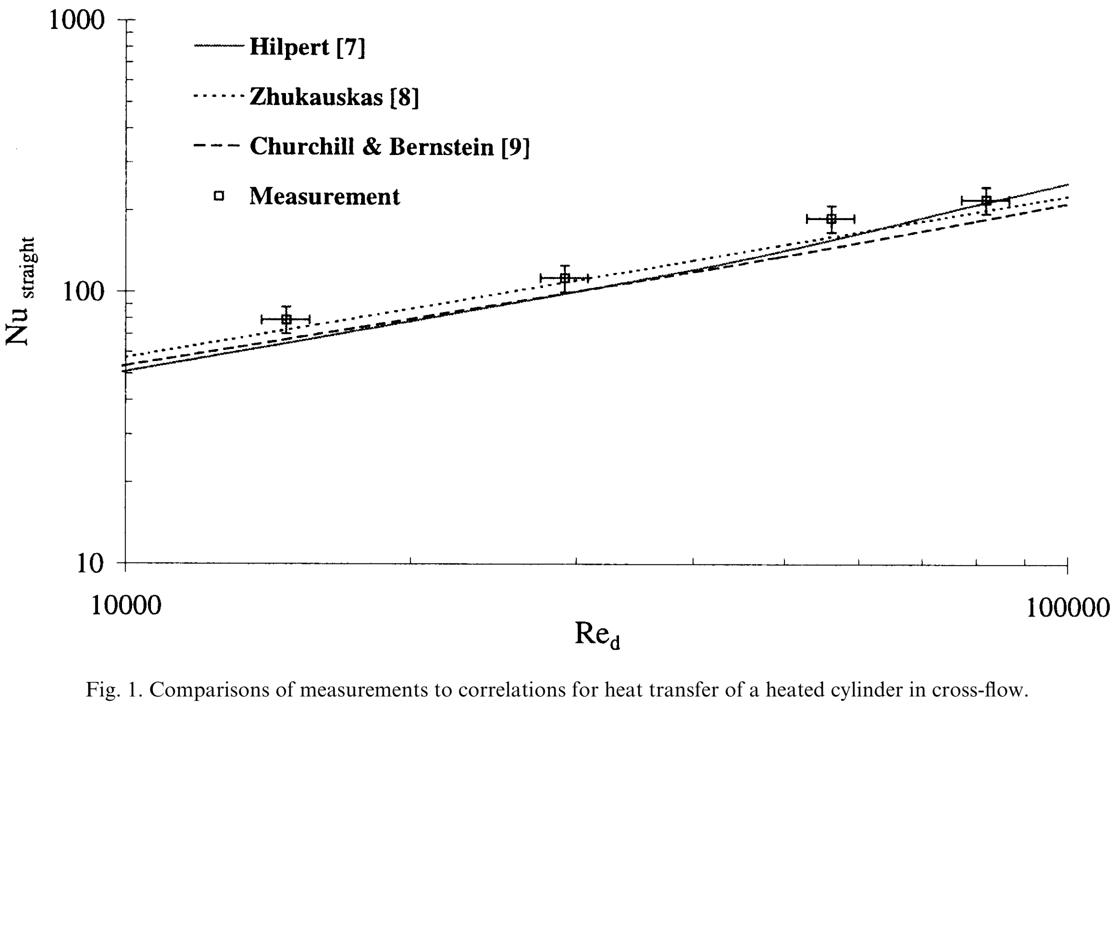

Fig. 1. Comparisons of measurements to correlations for heat transfer of a heated cylinder in cross-flow.

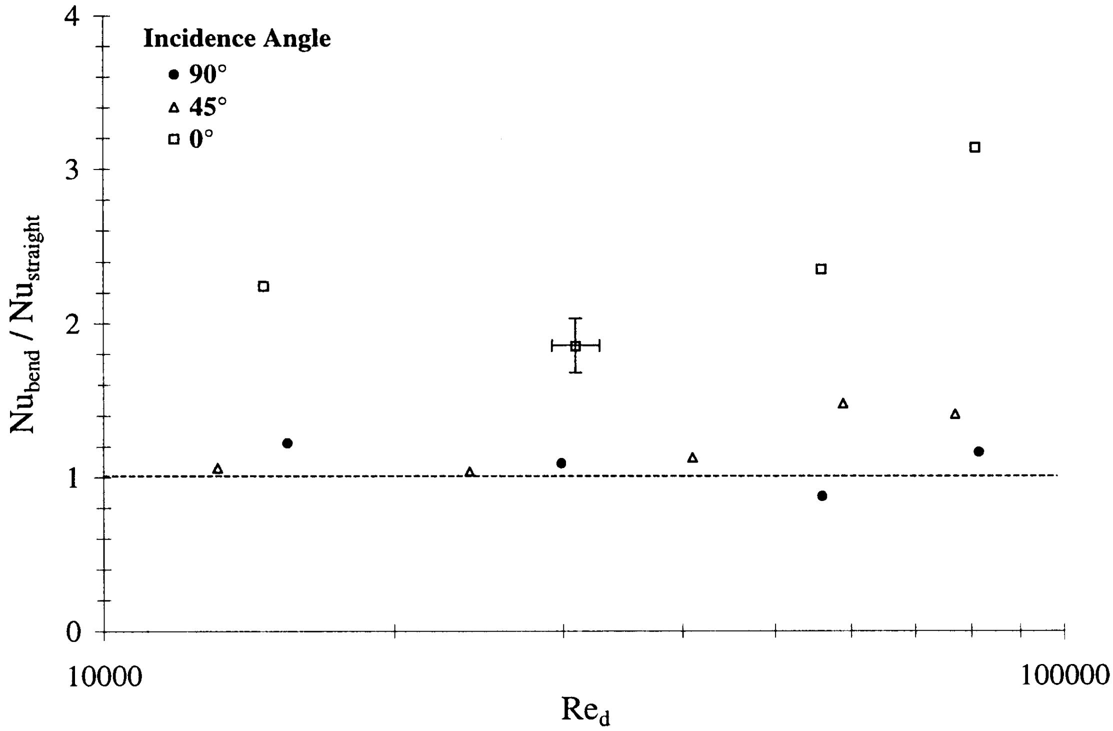

Fig. 2. Measured U-bend external heat transfer at various flow angles for $\delta = 0.292$.

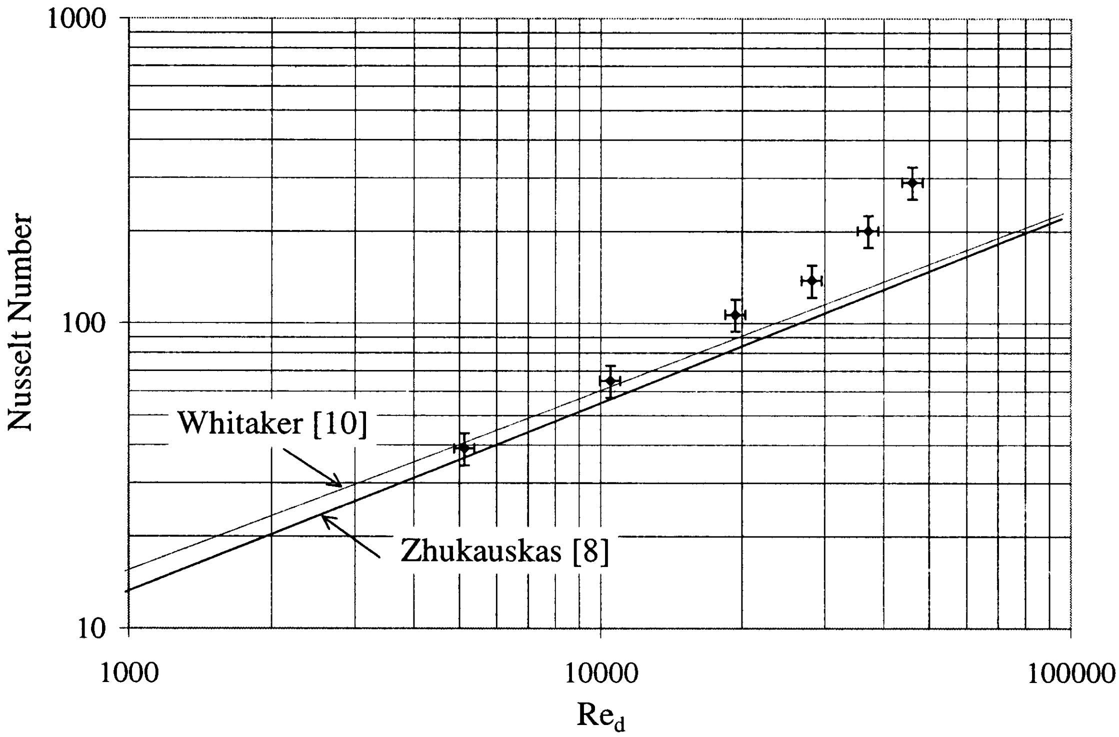

Fig. 3. Measured U-bend heat transfer for 0.100 bend curvature.

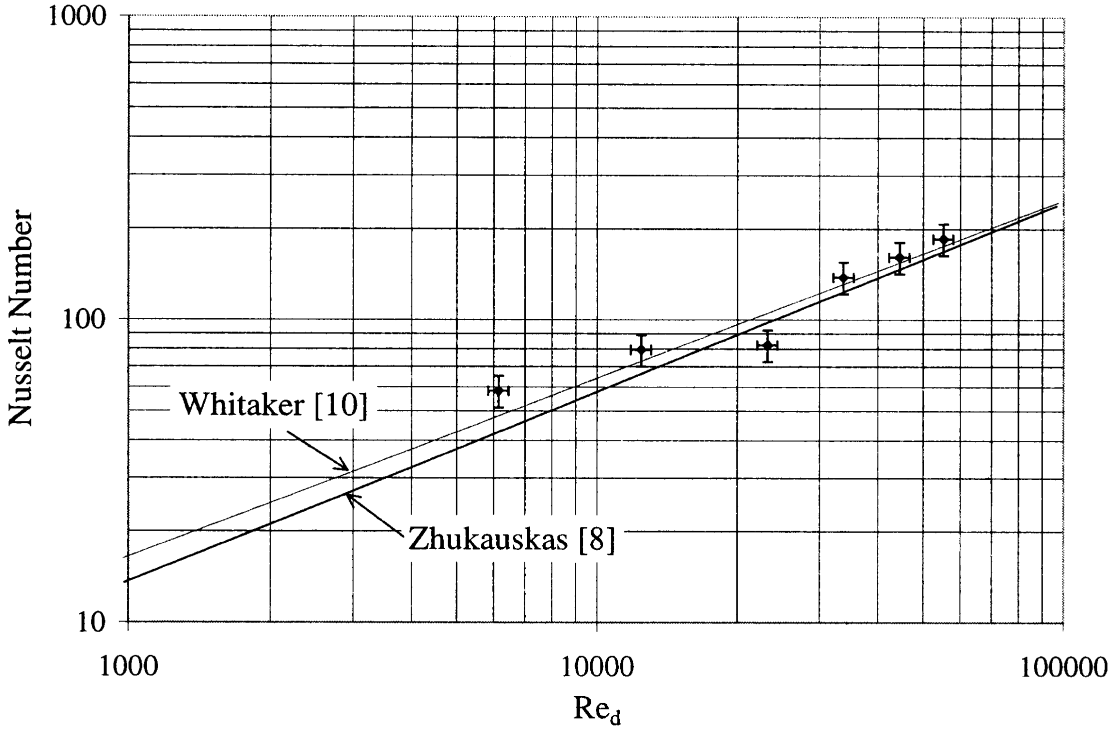

Fig. 4. Measured U-bend heat transfer for 0.103 bend curvature.

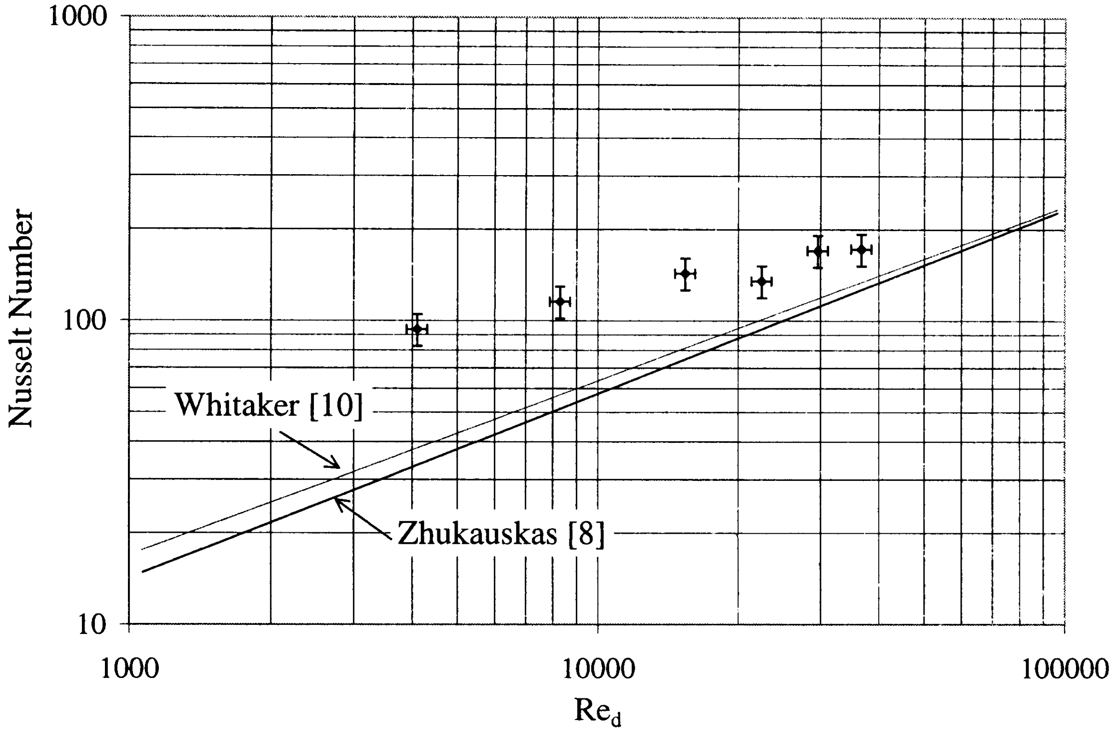

Fig. 5. Measured U-bend heat transfer for 0.167 bend curvature.

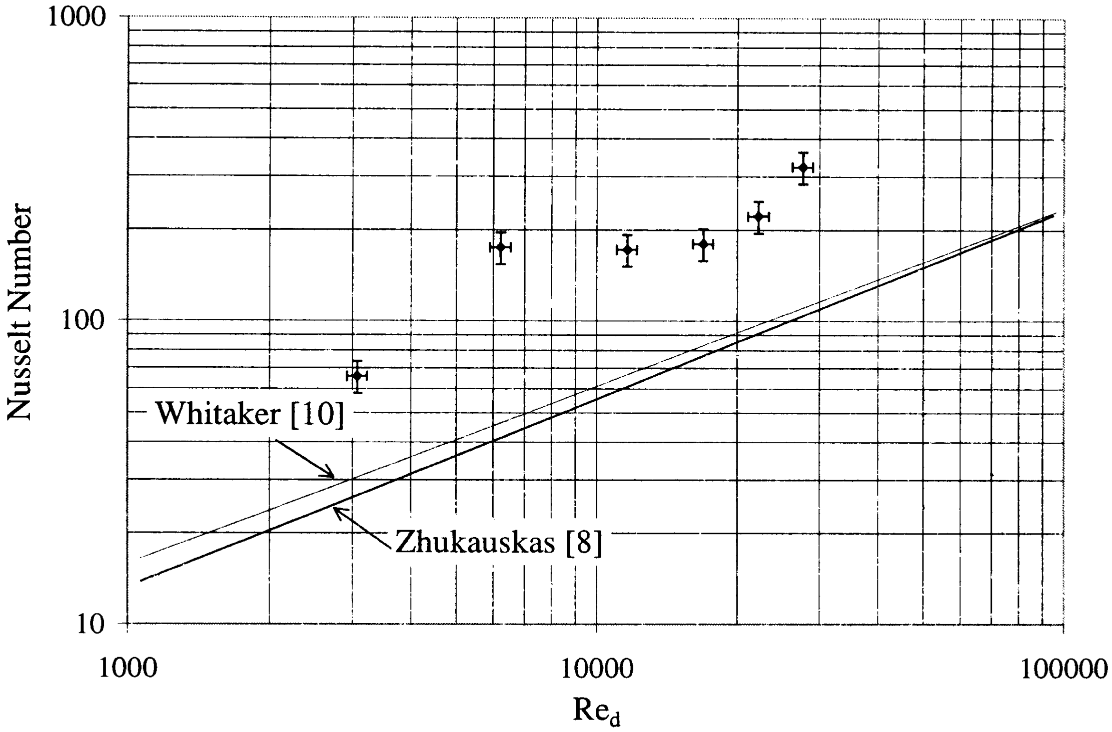

Fig. 6. Measured U-bend heat transfer for 0.200 bend curvature.

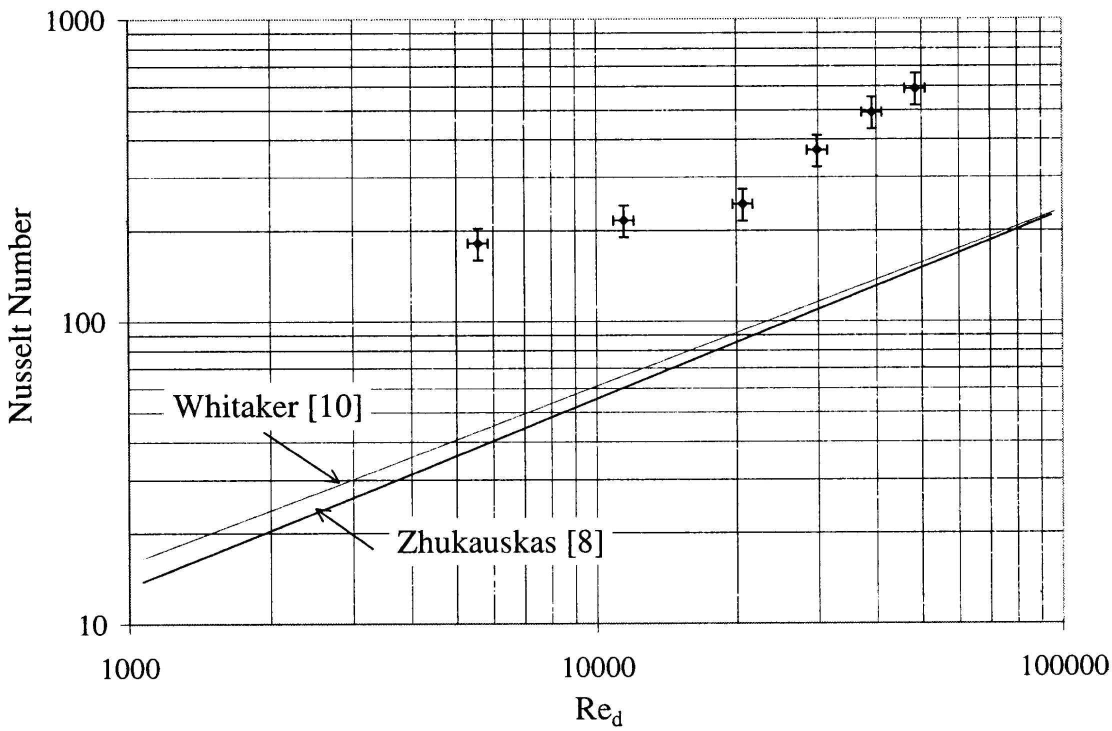

Fig. 7. Measured U-bend heat transfer for 0.250 bend curvature.

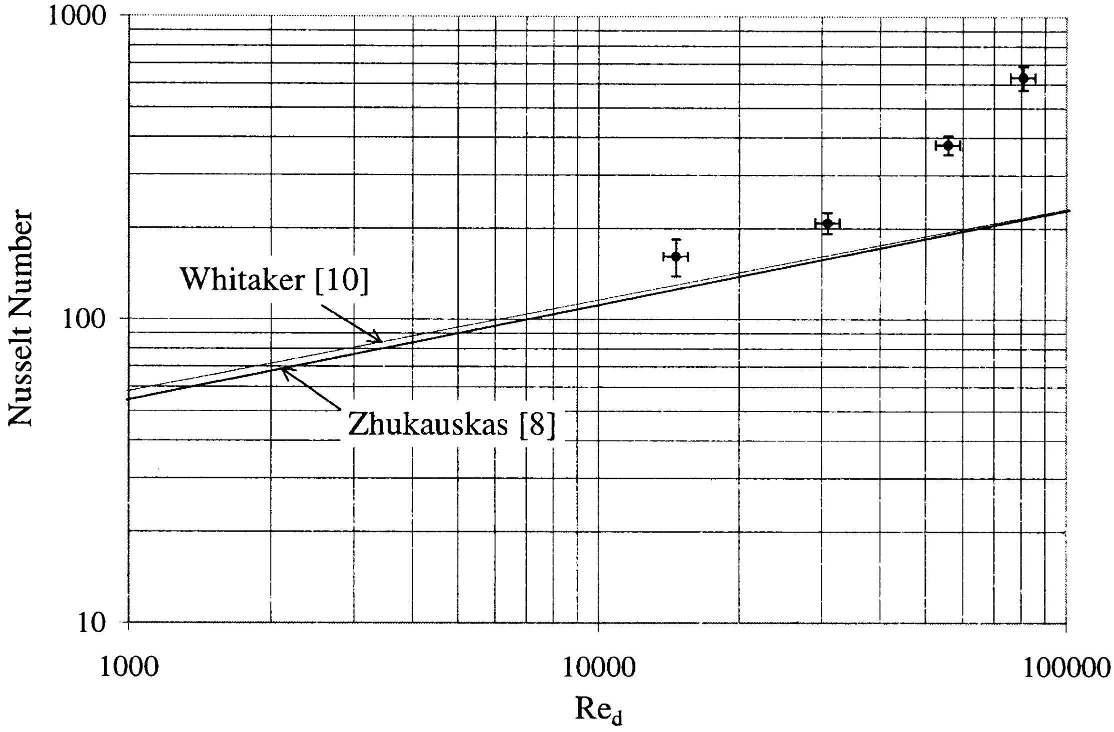

Fig. 8. Measured U-bend heat transfer for 0.292 bend curvature.

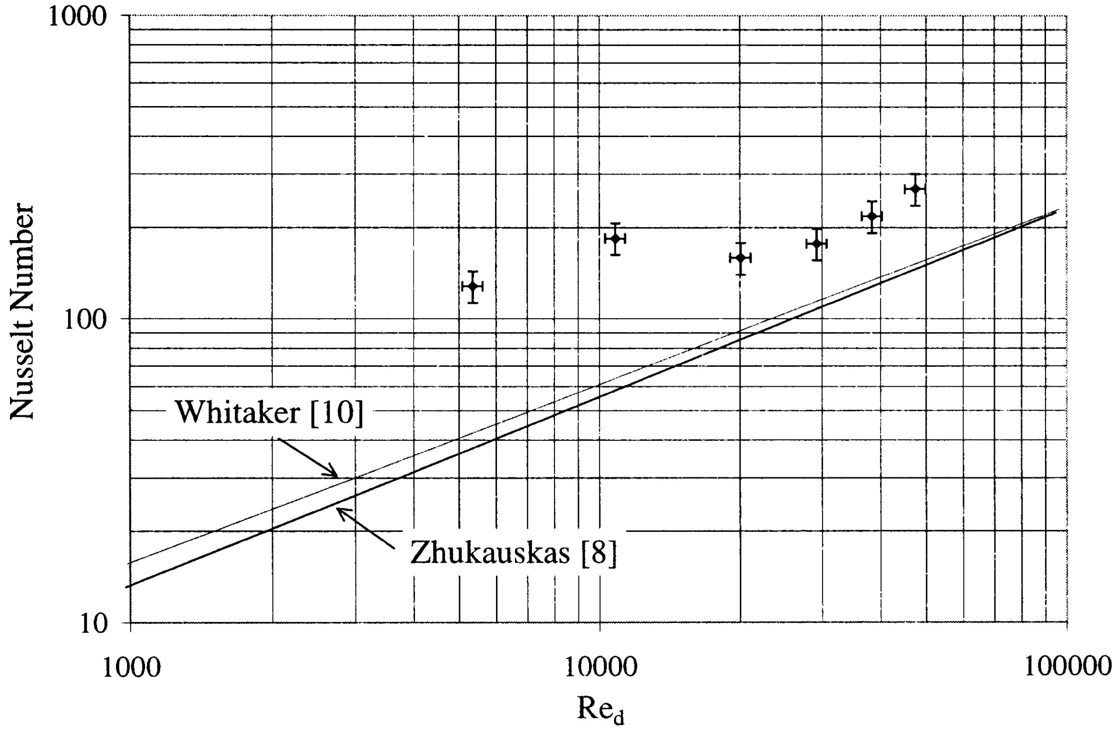

Fig. 9. Measured U-bend heat transfer for 0.417 bend curvature.

seven bends studied at an incidence angle of 0°. All figures also include Nusselt correlations (represented as straight lines) for both a cylinder in cross-flow and a submerged sphere as given respectively by Zhukauskas [8] and Whitaker [10]. For all the reduced measurements shown in Figs 3–9, the Nusselt values (for the U-bends) lie above the cylinder and sphere correlations. The error bars shown in Figs 3–9 represent the uncertainty associated with each individual data point. For all measurements taken the uncertainty in the Reynolds number was 5.8%. The uncertainty for the Nusselt numbers ranged from 28.5 to 7% with an average uncertainty of 12%. The different amounts of condensate collected was the cause for the large range in the level of uncertainty for the Nusselt numbers.

5. Discussion

The measurements shown in Figs 3–9 show that U-bend flows (at 0° incidence angle) always give higher heat transfer rates than (corresponding) straight cylinder flows. Only one data point in Fig. 4 fell below the Zhukauskas [8] correlation for a straight cylinder. In fact, the entire behavior of the $\delta = 0.103$ was troubling in that it was the only U-bend whose Nusselt values did not depart from those expected from a corresponding straight cylinder flow. Otherwise several clear trends in the data were present for all U-bends tested:

(i) The Nusselt value of the U-bend (Nu_{bend}) was always greater than that expected for a corresponding straight cylinder flow (Nu_{straight}).
(ii) The ratio $Nu_{\text{bend}}/Nu_{\text{straight}}$ was higher at the lower Reynolds numbers and decreased as the Reynolds number increased until around a Reynolds number of 15 000, where upon it began to increase again.
(iii) The $Nu_{\text{bend}}/Nu_{\text{straight}}$ ratio increased with increasing δ up to $\delta = 0.25$, where it then began to decrease.

Attempts at an overall Nusselt correlation relating the Reynolds number and δ while capturing the trends (i)–(iii) were not too successful. No correlation able to fit the data was found while satisfying the following general requirements:

- as δ approaches zero the Nusselt number goes to that for a straight tube, and does such with a slope approaching zero;
- as δ approaches unity the Nusselt number goes to that for a half sphere and does such with a slope approaching zero;
- the function is always positive and exhibits an optimum curvature ratio near 0.25.

Instead an easier-to-handle set of fits was constructed. The first correlation of this set was developed to mimic the general correlation structure typically used for cylinders but with an additional δ-dependence attached to the leading constant and the Reynolds exponent.

$$Nu_{\text{bend}} = C_1 \cdot [1+\delta]^{C_2} \cdot Re_{\text{d}}^{C_3} \tag{6}$$

In equation (6) C_1 was fixed to 0.228 in order to match the constant given by Zhukauskas and a least squares fit led to $C_2 = 3.53$ and $C_3 = 0.59 - 0.095\delta$. The standard error estimate (SEE) of this correlation to the actual measurements was 28.1%. Equation (6) collapses to the Zhukauskas correlation, shown in equation (5), as δ becomes small. The measurements are compared to this correlation in Fig. 10, which highlights that equation (6) was not able to capture the exponential behavior observed for the intermediate bend curvatures between 0.167 and 0.292. (The Reynolds number associated with the correlation lines and each data point are also shown in Fig. 10.)

Therefore, a second correlation (of a general exponential form) valid for the range of $0.16 < \delta < 0.30$ was developed to more closely resemble the measurements of intermediate bend curvature values. This correlation is shown in equations (7)–(9). This general form allowed for bend curvature influence on both the preceding constant, $f(\delta)$, and the exponent, $g(\delta)$, as was observed in the measurements.

$$Nu_{\text{bend}} = f(\delta) \cdot \exp[g(\delta) \cdot Re_{\text{d}} \cdot 10^{-5}] \tag{7}$$

$$f(\delta) = -5694.1\delta^2 + 2635.2\delta - 169.2 \tag{8}$$

$$g(\delta) = -418.7\delta^2 + 198.8\delta - 20.3 \tag{9}$$

The second correlation for intermediate bend curvatures is compared to the measurements in Fig. 11. There is an obvious inflection point in the correlation at approximately $\delta = 0.23$. This can be seen by computing the maximum values of the polynomial representations for both $f(\delta)$ and $g(\delta)$. These equations have a maximum at δ values of 0.230 and 0.236, respectively. A comparison of the predicted and measured Nusselt values using both bend correlations is shown in Fig. 12. The standard error estimate was 11.5% (shown as dashed lines), which signified a satisfactory fit to the data.

5.1. Geometrical analysis

A cylinder in cross-flow does not have any portion of its surface area exposed to its own wake, as is the case with a U-bend at 0° incidence angle. This self-inflicted wake effect was suggested to be the cause for the larger Nusselt values found for U-bends over those reported for cylinders in cross-flow. The geometrical arrangement of U-bends was evaluated in an effort to qualitatively construe the wake effect and to resolve the discrepancy between the measured and statistical δ_{opt} of 0.25 and 0.23, respectively.

Figure 13 depicts the U-bend geometry considered. The geometry was divided into three separate zones.

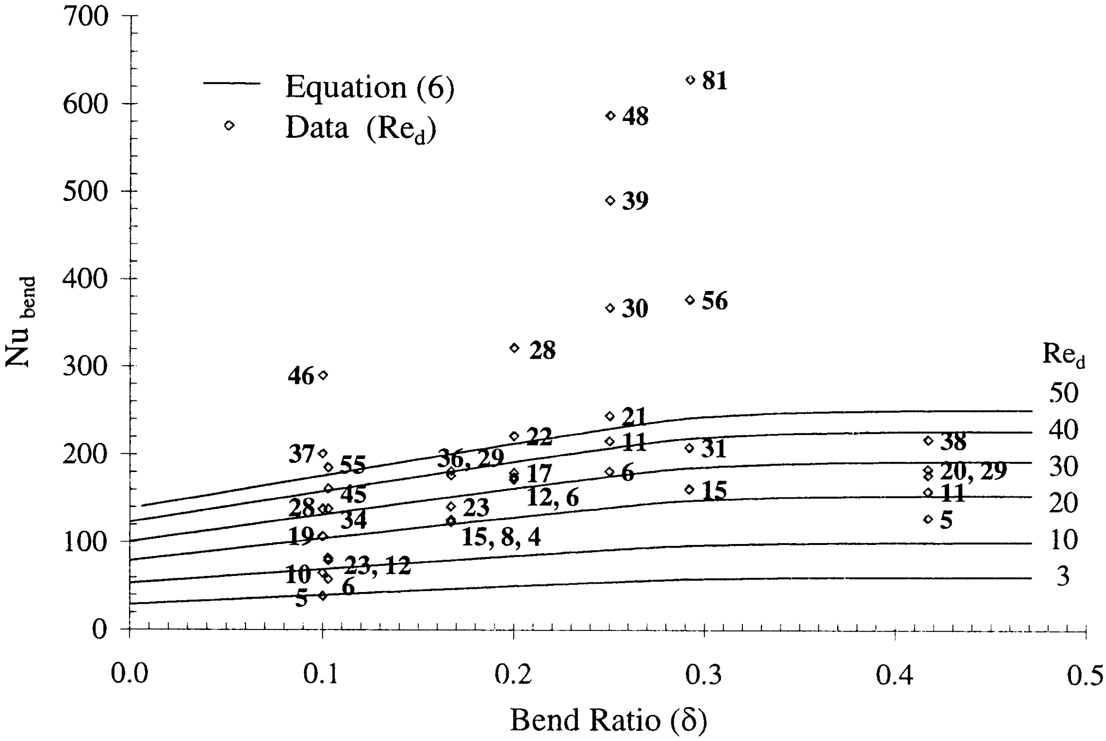

Fig. 10. Comparison of bend measurements and correlation for all bend curvatures measured at zero degree angle-of-incidence.

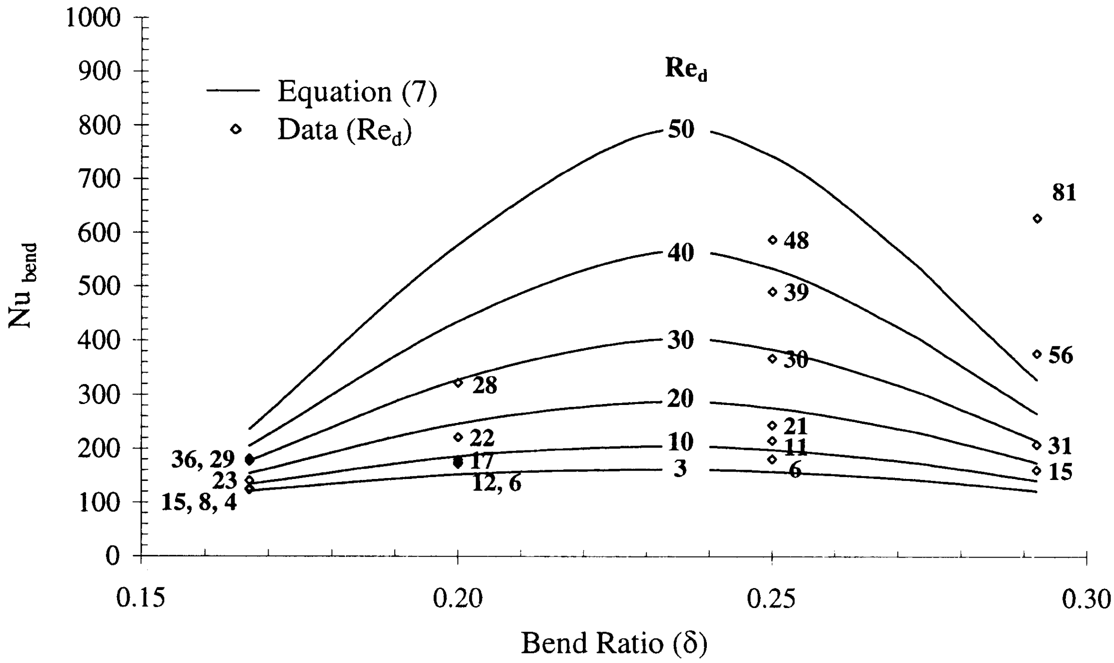

Fig. 11. Comparison of bend measurements and correlation for intermediate bend curvatures.

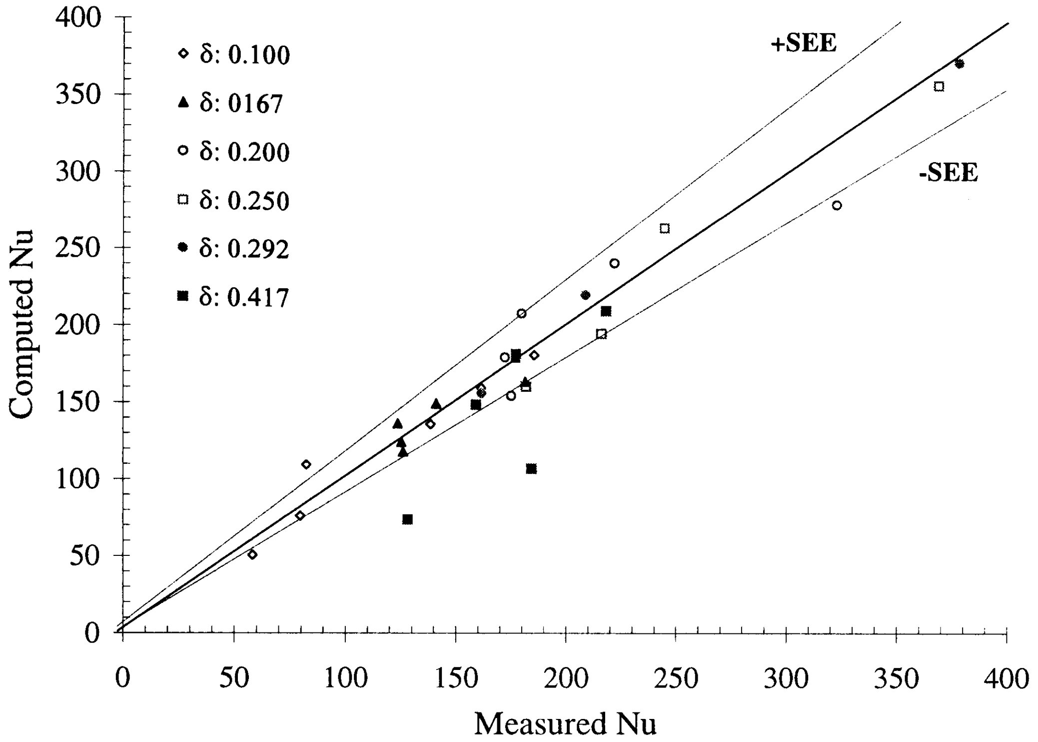

Fig. 12. Comparison of correlation to measurements.

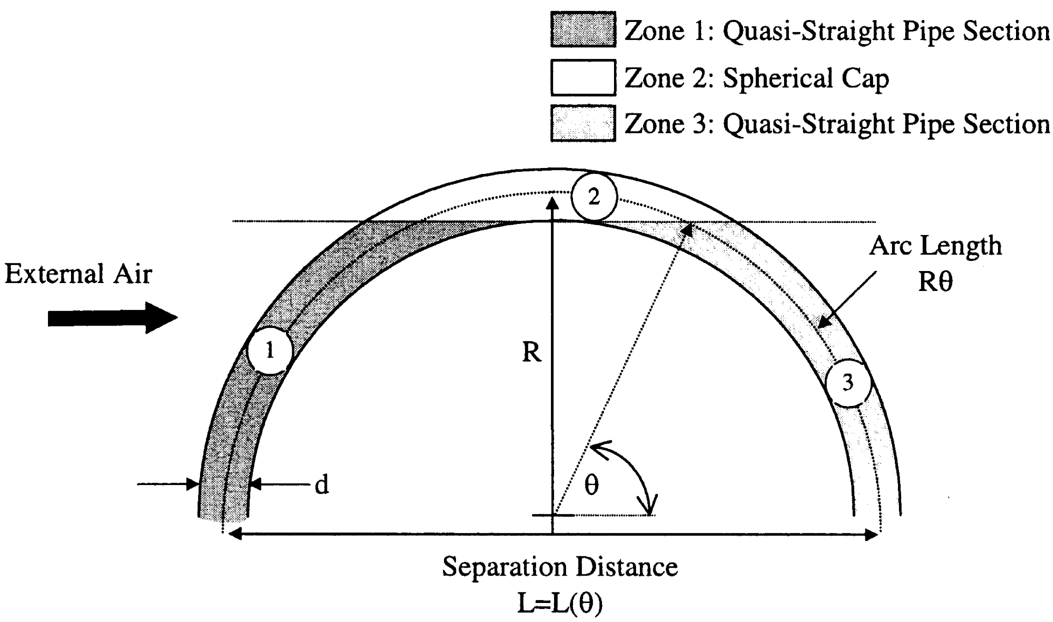

Fig. 13. U-bend geometry.

Zones 1 and 3 represented the portions of the U-bend surface that were viewed as quasi-straight pipe sections. Zone 2 was viewed as a spherical cap. The behaviors of the leading cylinder, Zone 1, and the spherical cap, Zone 2, are well known and documented. However, the behavior of the trailing 'cylinder', Zone 3, is dependent upon the separation distance from the leading 'cylinder'. The separation distance is defined as the distance between the axes of the two cylinders. The behavior of the trailing cylinder was estimated using the findings of Kostic and Oka [11]. In their investigation into the behavior of two cylinders in series, they found that the leading cylinder's heat transfer behavior was relatively independent of the separation distance. However, the trailing cylinder experienced an increase in heat transfer behavior for separation distances less than 2.7 diameters. This separation distance of $L/d \leqslant 2.7$ corresponds to a curvature ratio of $\delta \geqslant 0.37$. Therefore, increased heat transfer could be expected for any portion of Zone 3 that is within 2.7 diameters of Zone 1. The average separation distance, $\bar{L}$, between Zone 1 and Zone 3 (cylinders) is shown below.

$$\bar{L} = \frac{1}{\theta}\int_0^{\theta} R\cos\theta' \mathrm{d}\theta' = \frac{R}{\arcsin(1-\delta)}\int_0^{\arcsin(1-\delta)} \cos\theta' \mathrm{d}\theta' = \frac{R\cdot(1-\delta)}{\arcsin(1-\delta)} \quad (10)$$

However, as the bend curvature increases, the arc length of the trailing cylinder Zone 3 decreases. An inspection of Fig. 13 gives that the arc length of Zone 3 is as shown in equation (11).

$$\text{Arc length} = R\cdot\arcsin(1-\delta) \quad (11)$$

Therefore, there were two competing effects. The heat transfer behavior was expected to increase with an increasing δ according to equation (10) but decrease with an increasing δ according to equation (11). These two competing effects were combined by equating (10) and (11), which gave the following relationship for the optimum bend curvature ratio.

$$\sqrt{(1-\delta_{\mathrm{opt}})} = \arcsin(1-\delta_{\mathrm{opt}}) \quad (12)$$

A value of $\delta_{\mathrm{opt}} = 0.231$ satisfies equation (12). Although this geometrical analysis would not, on its own, convincingly identify 0.23 as the optimum bend ratio, it did support the empirical findings of an optimum bend curvature ratio near 0.25 and give credence to the correlation, equation (7), constructed.

6. Conclusions

The overall external convective heat transfer coefficients for flows around heated U-bends cooled by external air flows has been measured using a family of U-bends with varying bend curvature ratios. Intermediate bend curvature ratios ($0.10 \leqslant \delta \leqslant 0.30$) gave larger increases in the external heat transfer. In addition, a bend curvature value of 0.25 gave the highest level of measured external heat transfer rates. New Nusselt correlations were produced that represented the measurements within 11.5%. These correlations gave an optimum bend curvature ratio of 0.23. A geometrical analysis supported the value of 0.23 as being the optimum bend curvature ratio. Further work is needed in obtaining angular (circumference averaged) Nusselt values about the bend plane, $Nu(\theta)$. Also, a study is needed investigating the dependence on angle-of-incidence between the bend plane and the external air flow direction on the U-bend heat transfer rate.

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