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The unusual behavior of local heat transfer coefficient in a circular tube with a bell-mouth inlet

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Abstract

The local heat transfer characteristics for ethylene glycol water mixtures flowing in a horizontal circular straight tube with a bell-mouth inlet have been determined experimentally over a flow Reynolds number range of 1500–27,000. A wall-boundary heating condition of uniform heat flux was imposed. The variation of local heat transfer coefficient with length in the transition and turbulent flow regimes is very unusual. For the bell-mouth inlet, the boundary layer along the tube wall is at first laminar and then changes through a transition region to the turbulent condition causing a dip in the Nu–x/D curve. The length of the dip in the transition region is much longer than that in the turbulent region. For our experiments with a fixed inside diameter of 1.584 cm, the length of the dip in the transition region varied from $x/D = 100$–175 in comparison to an $x/D < 25$ for the turbulent region. The presence of the dip in the transition region causes a significant influence on both the local and the average heat transfer coefficients. This is particularly important for heat transfer calculations in short tubes with a bell-mouth inlet.

Keywords: Local heat transfer coefficient; Forced convection; Mixed convection; Transition region; Circular tube; Uniform wall heat flux; Heat exchangers

1. Introduction

The effect of entrance shapes normally used in heat exchangers on turbulent forced convection heat transfer in a tube has been studied by many investigators. These works have been reviewed by Al-Arabi [1]. Two entrance conditions can generally be distinguished. In the first and the most common condition, the entrance shape causes extra turbulence as the fluid enters the heat transfer tube. Such as the square-edged or reentrant entrances employed in tubular heat exchangers. For these entrances, the local heat transfer coefficient decreases with tube length asymptotically. Fig. 1(a) qualitatively shows the general shape of variation of local dimensionless heat transfer coefficient (Nu) with dimensionless tube length (x/D) for these inlets.

The second entrance condition is characterized by uniform velocity and temperature distributions at the entrance. This condition occurs in practice when the tube entrance is in the form of a bell-mouth. This entrance shape is used in some heat exchangers mainly to avoid the presence of eddies which are believed to be one of the reasons for tube inlet-end erosion. For the bell-mouth inlet, the variation of local heat transfer coefficient with length is very unusual. Fig. 1(b) qualitatively shows the general shape of variation of local dimensionless heat transfer coefficient (Nu) with dimensionless tube length (x/D) for the bell-mouth inlet. In this case, the boundary layer along the tube wall is at first laminar and then changes through a transition region to the turbulent condition causing a dip in the Nu–x/D curve. The length of the dip decreases with the increase of the turbulent Reynolds number and is relatively short (less than 25 tube diameters from the tube entrance), as observed by Mills [2] and Sukomel and Velichko [3]. The presence of the dip in this case significantly influences the local turbulent heat transfer characteristics of the tube. However, its effect on the average heat transfer is much less.

In practice it is not always feasible or possible to design or operate the heat exchangers in the turbulent re-

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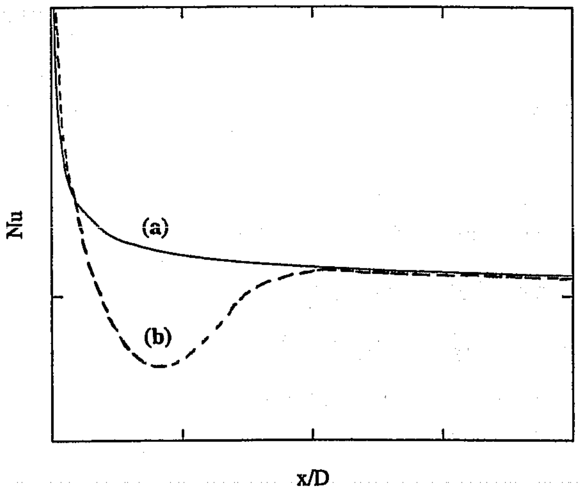

Fig. 1. General shape of variation of local Nusselt number with dimensionless tube length in the turbulent region for (a) square-edged and re-entrant inlets and (b) bell-mouth inlet.

gion and the flow inside the tubes falls into the transition region. However, very little has been done to investigate this unusual behavior of the local heat transfer coefficient in the transition region in which the Reynolds number is relatively low in comparison to the turbulent region. Since the length of the dip decreases with the increase of the turbulent Reynolds number, it is anticipated that the length of the dip in the transition region would be much longer than in the turbulent region. This would then cause a significant influence on both the local and average heat transfer characteristics of the tube.

The objective of this study was to experimentally investigate the behavior of local heat transfer coefficient in the transition region for a circular tube with a bell-mouth entrance. For this purpose, local heat transfer measurements under uniform wall heat flux boundary condition were made across all flow regimes (laminar, transition, and turbulent) for several different levels of disturbance (different entrance shapes) at the inlet to the heat transfer tube.

2. Heat transfer experiments

A schematic diagram of the overall experimental apparatus and the heat transfer test section used for heat transfer experiments is shown in Fig. 2. The overall experimental setup shown was also used for pressure drop measurements [4]. A detailed description of the heat transfer experimental apparatus and procedures used in this study were reported in [5]. In this paper only a brief description of the experimental setup and procedures will be provided. The local forced and mixed convective heat transfer measurements were made in a horizontal electrically heated stainless steel circular

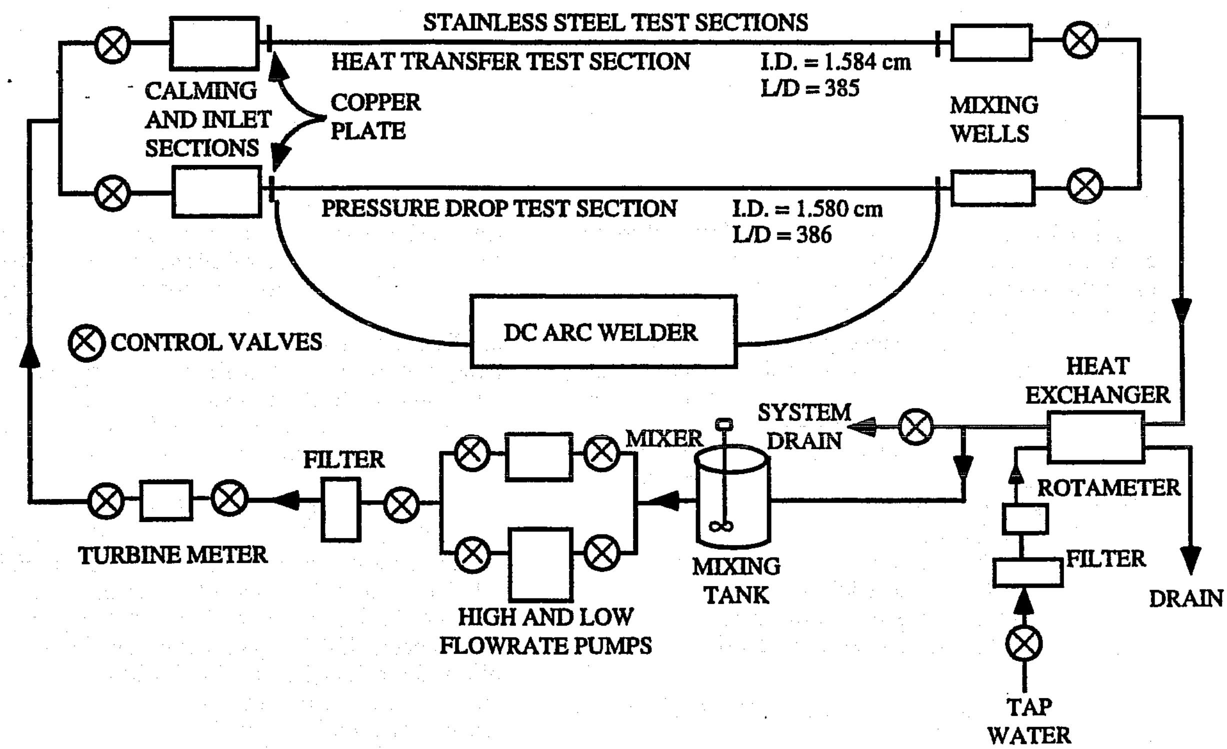

Fig. 2. Schematic diagram of the experimental setup.

straight tube with reentrant, square-edged, and bell-mouth inlets under uniform wall heat flux condition. The pipe had an inside diameter of 1.58 cm and an outside diameter of 1.90 cm. The total length of the test section was 6.10 m, which provided a maximum length-to-diameter ratio (L/D) of 385. A uniform wall heat flux boundary condition was maintained by a dc arc welder. Thermocouples (T-type) were placed on the outer surface of the tube wall at close intervals near the entrance and at greater intervals further downstream. Twenty-six axial locations were designated with four thermocouples at each location. The thermocouples were placed 90° apart around the periphery. From the local peripheral wall temperature measurements at each axial location, the inside wall temperatures and the local heat transfer coefficients were calculated using a data reduction computer program [6]. In these calculations, axial conduction was assumed negligible ($\mathrm{Re}\,\mathrm{Pr} > 42{,}000$ in all cases), but peripheral and radial conduction of heat in the tube wall were included. In addition, the bulk fluid temperature was assumed to increase linearly from the inlet to the outlet.

To ensure a uniform velocity distribution at the entrance of the test section, the flow passed through calming and inlet sections (see Fig. 3). The calming section consisted of a 17.8-cm-outer-diameter acrylic plastic cylinder with three perforated acrylic plastic plates with an open area ratio of 0.312 (73 holes per plate, hole diameter 1.1 cm) followed by tightly packed soda straws (inside diameter 0.57 cm, length 10.2 cm, open area ratio 0.915) sandwiched between fine plastic mesh screens (strand diameter 0.03 cm, mesh width 0.14 cm, open area ratio 0.65). Two more fine plastic mesh screens were placed at the exit of the calming section. The total length of the calming section is 61.6 cm. Test fluid leaving the calming section entered the inlet section and flowed undisturbed through 23.5 cm of a 16.5-cm-outer-diameter acrylic plastic tube before it entered the test section. This section was added to ensure a uniform velocity distribution upon entering the test section. The inlet section had the versatility of being modified to incorporate a reentrant or a bell-mouth entrance (see Fig. 3). The reentrant entrance was simulated by sliding 1.91 cm of the tube entrance length into the inlet section (Fig. 3), which was otherwise the square-edged (sudden contraction) entrance. For the bell-mouth entrance, a nozzle had to be constructed to replace the inlet section of Fig. 3. The nozzle was designed according to the method suggested by Morel [7] and was constructed of fiberglass. The nozzle had a contraction ratio of 10.7 and a total length of 23.5 cm.

Various mixtures of ethylene glycol and distilled water were used as the test fluid throughout the experiments to cover the laminar, transition, and turbulent flow regimes. For the experiments with all three inlets (reentrant, square-edged, and bell-mouth), the local bulk Reynolds number ranged from about 280 to 49000, the local bulk Prandtl number varied from about 4 to 158, the local bulk Grashof number range was from 1000 to 2.5×10^5, and the local bulk Nusselt number varied from 13 to 258. The uniform wall average heat flux for the experiments ranged from about 4 to 670 $\mathrm{kW/m^2}$. Heat bal-

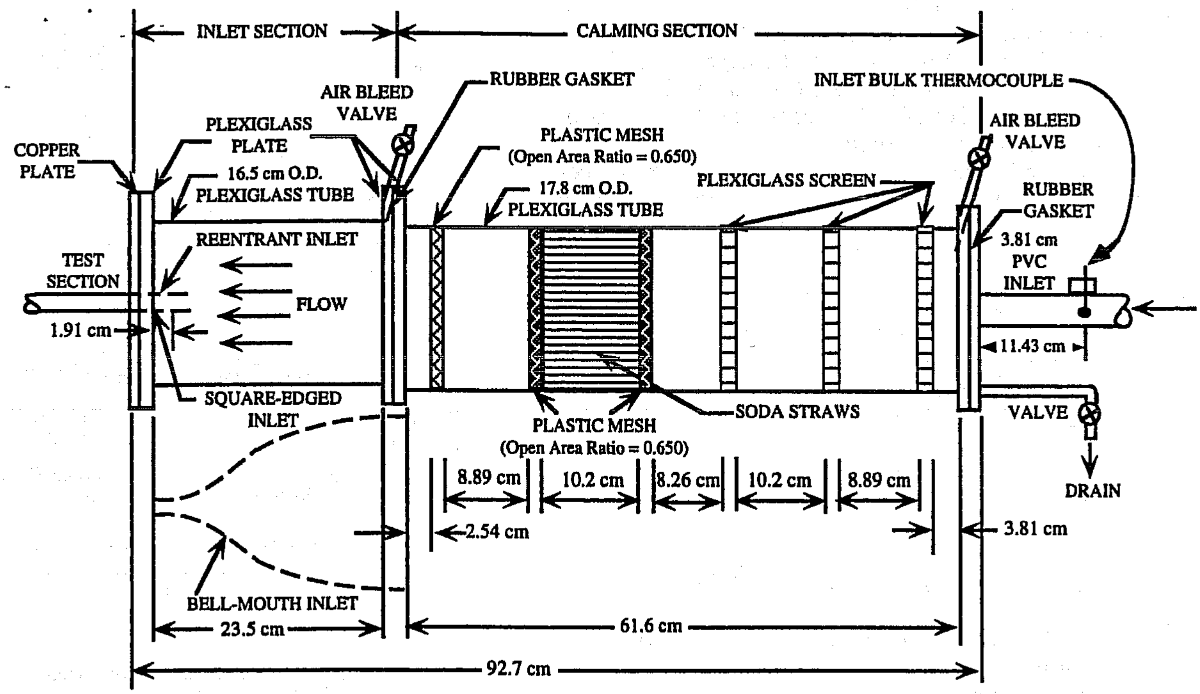

Fig. 3. Schematic diagram of the calming and inlet sections.

ance errors were calculated for all experimental runs by taking a percent difference between two methods of calculating the heat addition. The product of the voltage drop across the test section and the current carried by the tube was the primary method, while the fluid enthalpy rise from inlet to exit was the second method. The heat balance error between the two methods in all cases was less than 5%. The primary method was the one used in the computer program [6] for all heat flux and heat transfer coefficient calculations.

The reliability of the flow circulation system and of the experimental procedures were checked by making several calibration runs with distilled water and the results were compared with several of the well-established heat transfer correlations for fully developed turbulent pipe flow presented by Bhatti and Shah [8]. The experimental data were within ±8% of the predicted values. The uncertainty analysis of the overall experimental procedures using the method of Kline and McClintock [9] showed that there is a maximum of 9% uncertainty for heat transfer coefficient calculations. Experiments under the same conditions were conducted periodically to ensure the repeatability of the results. The difference between the duplicated experimental runs were within ±9%.

3. Experimental results

The heat transfer experimental results presented in this section are for both forced and mixed convection in a tube under uniform wall heat flux boundary condition. In our previous works [5,10], it was established that secondary flow is present in both laminar and transition flow regimes. The starting length necessary for the establishment of the free convection effects is influenced by the entrance shape and is somewhere between 20 and 70 tube diameters from the entrance to the tube for the inlets considered in this study. The strong influence of buoyancy forces (free convection) on forced convection in laminar and transition regions gives rise to mixed convection flow in the tube. This in turn results in a much higher fully developed laminar uniform wall heat flux Nusselt number than the accepted 4.364 value. As shown in Figs. 4, 6–8 the Nusselt number values are in the order of 20. However, in the turbulent region, the secondary flow effect is suppressed by the turbulent motion and the heat transfer mode is pure forced convection.

In order to investigate the behavior of local heat transfer coefficient in the transition region with a bell-mouth entrance it is first necessary to establish the influence of flow regime and different levels of inlet disturbance on the local heat transfer coefficient.

4. Entry effects in the laminar and turbulent regions

Fig. 4 shows some of our local forced convective heat transfer measurements along the tube in the laminar and turbulent regions. In these experiments the entry effects were investigated by using three different entrance shapes with different levels of disturbance. The reentrant and bell-mouth inlets provided the most and the least disturbance to the fluid, respectively. The square-edged inlet fell somewhere between the two other inlets. The heat transfer results presented in Fig. 4 for the three different entrance shapes are at comparable laminar and turbulent Reynolds numbers.

As shown in Fig. 4, the reentrant and square-edged inlets that cause extra turbulence as the fluid enters the heat transfer tube have no influence on the laminar and turbulent heat transfer coefficients. For these entrance shapes the dimensionless local average peripheral heat transfer coefficient decreases with tube length asymptotically. This trend is also true in the laminar flow regime with a bell-mouth inlet. However, local heat transfer measurements made in the turbulent flow regime with a bell-mouth entrance show an unusual be-

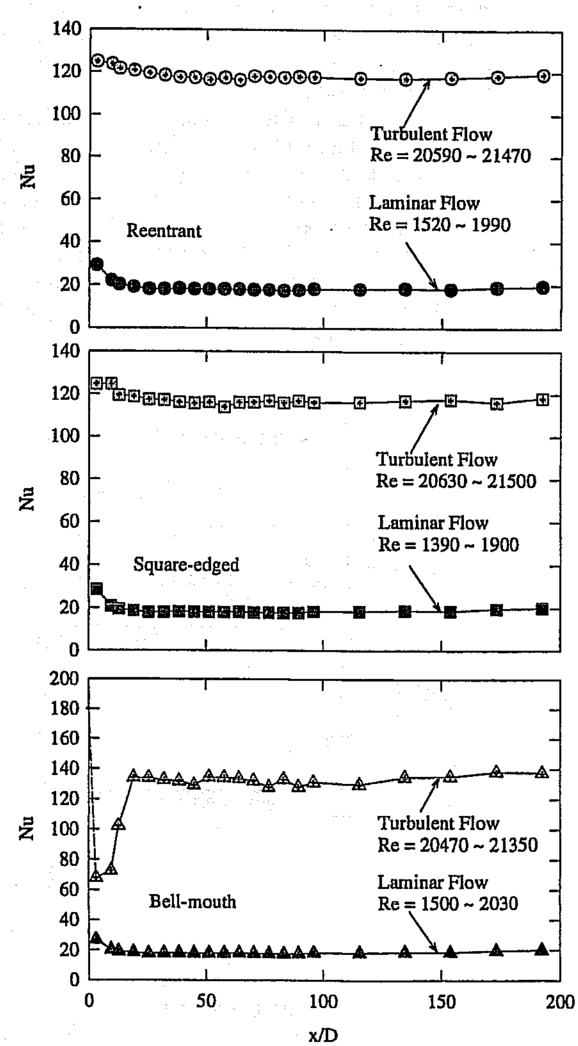

Fig. 4. Variation of local Nusselt number with length for the reentrant, square-edged, and bell-mouth inlets in the laminar and turbulent regions.

havior. In this case the heat transfer coefficient goes through a sharp dip before it behaves similar to the other two inlets. The length of the dip extends to about 25 tube diameters from the tube entrance.

The unusual behavior of the turbulent heat transfer coefficient along the tube for the bell-mouth inlet can be attributed to the characteristics of this particular inlet. Due to the low levels of turbulence caused by the bell-mouth inlet, the velocity and temperature distributions at the entrance to the heat transfer tube are uniform. In this case as the fluid enters the tube the boundary layer along the tube wall is at first laminar and then changes through a transition region to the turbulent condition causing a dip in the Nu–x/D curve. However, for the other two inlets this boundary layer changing behavior does not exist. This is due to the extra turbulence caused by the reentrant and square-edged inlets which leads to a turbulent boundary layer at the entrance to the tube.

5. Entry effects in the transition region

As reported in our previous works [5,10], the type of inlet configuration influences the start and end of the heat transfer transition region. The range of heat transfer transition Reynolds numbers along the pipe ($3 \leqslant x/D \leqslant 192$), were reported to be about 2000–8500 for the reentrant inlet, 2400–8800 for the square-edged inlet, and 3400–17,000 for the bell-mouth inlet. Fig. 5 shows the variation of local average peripheral Nusselt number along the tube length in the transition region for the three entrance shapes at comparable transition Reynolds numbers.

In the transition region the reentrant and square-edged entrances show no influence on the local heat transfer coefficients. For these inlets as shown in Fig. 5, the local Nusselt number has a minimum value at x/D approximately equal to 25 and increases monotonically along the tube rather than staying at a relatively constant value as was observed in Fig. 4. Considering Newton's law of cooling for a uniform wall heat flux boundary condition, the increase in the Nu values with the tube length is associated with the decrease in the difference between the inside wall and fluid bulk temperatures ($T_w - T_b$) along the tube. The inside wall temperature and the fluid bulk temperature increase monotonically after they have reached their fully developed values but the fluid bulk temperature has a larger rate of increase. This causes the heat transfer coefficient to increase along the tube.

For a bell-mouth inlet, Fig. 5 clearly shows that the dip in the Nu–x/D curve observed in the turbulent region is much more significant in the transition region. In this case, the length of the dip is much longer and extends to about 140 diameters from the tube entrance. This means that the boundary layer along the tube wall over the first 140 diameters from the tube entrance is laminar and for $x/D > 140$ changes through a transition region to the turbulent condition. The large variation in the value of Nusselt number along the tube, from about 30 to 80, makes a significant impact on both the local and the average heat transfer characteristics of the tube. This impact is particularly important for heat transfer calculations in short tubes with a bell-mouth inlet.

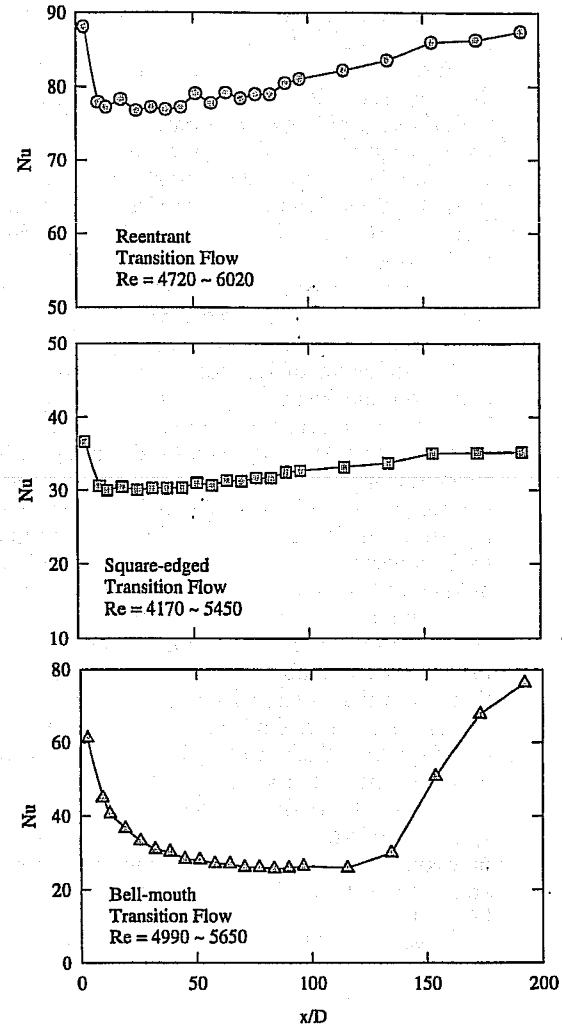

Fig. 5. Variation of local Nusselt number with length for the reentrant, square-edged, and bell-mouth inlets in the transition region.

6. Heat transfer characteristic of the bell-mouth inlet

The heat transfer results presented in Figs. 4 and 5 demonstrated that the heat transfer characteristic of the bell-mouth inlet in the transition and the turbulent regions is very different than those of the reentrant and square-edged inlets. In addition, it was established that the dip in the Nu–x/D curve for the transition region has a much greater influence on both the local and the average heat transfer characteristics of the tube. Therefore, further investigation into the unusual heat transfer characteristic of the bell-mouth inlet in the transition region is warranted.

Further investigation into the unusual behavior of the bell-mouth inlet will be aimed at determination of the influence of Reynolds number and inlet disturbance on the length of the dip in the transition region. To investigate the effect of inlet disturbance on the bell-mouth entrance, three different screen sizes were used in the last section of the calming section before the fluid entered the bell-mouth entrance (see Fig. 3). The three different plastic mesh screens used are referred to as coarse, medium, and fine mesh screens with open area ratios of 0.825, 0.759, and 0.650, respectively.

The influence of Reynolds number and inlet disturbance on the local heat transfer characteristic of the bell-mouth entrance is shown in Figs. 6–8. For the coarse mesh screen (see Fig. 6), the unusual heat transfer behavior is observed in the transition Reynolds number range from about 3470 to 7700. From the results presented in Fig. 6, the local Nusselt number starts to shift upward at approximately $x/D = 175$ for a Reynolds number of about 3470. This means that the laminar boundary layer becomes unstable and it starts to change from laminar to turbulent at this location. As the Reynolds number increases, the dimensionless location for which the local Nusselt number starts to shift upward decreases. This behavior indicates that the location for which the boundary layer changes from laminar to turbulent decreases as Reynolds number increases. The band of the dip reaches the smallest value when x/D approximately equals to 100 at a Reynolds number of about 7700. For Reynolds numbers less than 3470 and greater than 7700, this unusual behavior can no longer be observed. As seen in Fig. 6 for Reynolds numbers less than 3470, the local Nusselt numbers along the dimensionless x/D locations behave exactly the same as they behaved in the laminar region (see Fig. 4). However, in the turbulent region ($Re > 7700$) the dip observed in Fig. 4 for the bell-mouth inlet is no longer present. The absence of the dip is due to the extra turbulence caused by the coarse mesh screens placed in front of the bell-mouth entrance. In this case, the extent of the

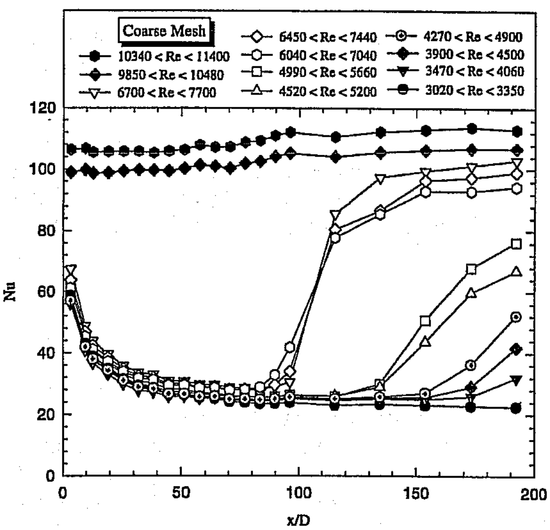

Fig. 6. Variation of local Nusselt number with length for a bell-mouth inlet with coarse mesh screens in the laminar, transition, and turbulent regions.

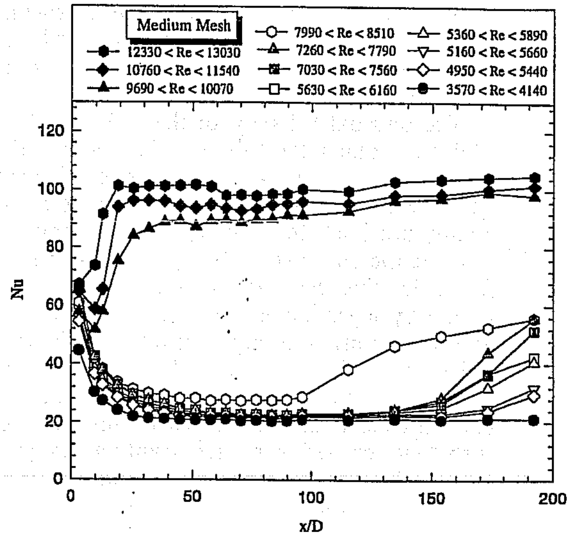

Fig. 7. Variation of local Nusselt number with length for a bell-mouth inlet with medium mesh screens in the laminar, transition, and turbulent regions.

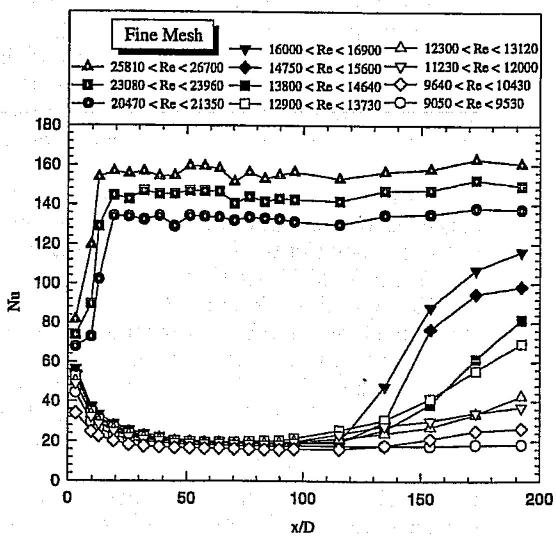

Fig. 8. Variation of local Nussel number with length for a bell-mouth inlet with fine mesh screens in the laminar, transition, and turbulent regions.

laminar boundary layer is very short ($x/D < 3$) and the boundary layer is practically turbulent at the entrance.

For the medium mesh screen (see Fig. 7) the unusual heat transfer behavior is observed when Reynolds number ranges from about 4950 to 8510. For Reynolds numbers less than 4950, the local Nusselt numbers along the dimensionless x/D locations behave exactly the same as they did in the laminar region. When Reynolds number is greater than 4950, the local Nusselt number starts to shift upward at x/D approximately equal to 175 and causes a dip in the Nu–x/D curve. Similar to the situation observed for the coarse mesh screen. As the Reynolds number increases, the x/D location for which the local Nusselt number starts to shift upward decreases. When Reynolds number reaches approximately 8510, the dip reaches its smallest value at x/D approximately equal to 100. For Reynolds numbers greater than 8510, the local heat transfer characteristics of the tube are very similar to what was observed in Fig. 4 for a bell-mouth entrance. In this case, the length of the dip extends to about 25 tube diameters from the tube entrance and decreases with an increase in the turbulent Reynolds number. These observations are consistent with the findings of Mills [2] and Sukomel and Velichko [3].

For the fine mesh screen (see Fig. 8), the unusual heat transfer behavior is observed when Reynolds number ranges from about 9640–16,900. For Reynolds numbers less than 9640, the local Nusselt numbers along the dimensionless x/D locations behave exactly the same as they behaved in the laminar region. When Reynolds number is greater than 6940, the local Nusselt number starts to shift upward at x/D approximately equal to 175 and cause a dip in the Nu–x/D curve. Similar to the situation for the coarse and medium mesh screens. As the Reynolds number increases, the x/D location for which the local Nusselt number starts to shift upward decreases. When Reynolds number reaches approximately 16,900, the dip reaches its smallest value at x/D approximately equal to 100. For Reynolds numbers greater than 16,900, the length of the dip decreases considerably and the local heat transfer behavior is very similar to the turbulent behavior observed for the medium mesh screen (see Fig. 7) and those depicted in Fig. 4 for the bell-mouth inlet. Once again, the length of the dip is about 25 tube diameters and decreases with an increase in the turbulent Reynolds number.

Referring to Figs. 6–8, there are several common key factors that can be summarized as follows:

1. The dimensionless location x/D for which the local Nusselt number starts to shift upward is approximately a constant ($x/D \approx 175$). This means that the laminar boundary layer exists up to $x/D \approx 175$. Then, the boundary layer becomes unstable and starts to change from laminar to turbulent.
2. The dimensionless location x/D for which the dip becomes smallest is approximately a constant ($x/D \approx 100$). The band of the dip gets smaller as the Reynolds number increase. This means that the distance required for the boundary layer change decreases as the Reynolds number increases.
3. For all three different screen sizes, the dip can no longer be detected when it reaches the smallest value at $x/D \approx 100$.
4. The local heat transfer coefficient has laminar behavior for these three different screen sizes for the first 100 x/D locations. This means that the boundary layer does not change from laminar to turbulent until the flow goes beyond $x/D \approx 100$.
5. The Reynolds number for which the unusual heat transfer behavior begins is inlet condition dependent. For a high inlet turbulence level (coarse screen), the Reynolds number is 3470. For a low inlet turbulence level (fine screen), the Reynolds number is 9640. For a moderate inlet turbulence level (medium screen), the Reynolds number is 4950.

7. Practical significance

Bell-mouth inlet is used in some heat exchangers mainly to avoid the presence of eddies which are believed to be one of the reasons for tube inlet-end erosion. For the bell-mouth inlet, the variation of local heat transfer coefficient with length in the transition and turbulent flow regions is very unusual. For this inlet, the boundary layer along the tube wall is at first laminar and then changes through a transition to the turbulent condition causing a dip in the Nu–x/D curve. The length of the dip in the transition region is much longer ($x/D = 100$–175) than in the turbulent region ($x/D < 25$). This causes a significant influence on both the local and the average heat transfer coefficients. This is particularly important for heat transfer calculations in short tube heat exchangers with a bell-mouth inlet.

8. Conclusions

In this study, through careful measurements, it was established that for a bell-mouth entrance the variation of local heat transfer coefficient with length in the transition and turbulent flow regimes is very unusual. In this case, the boundary layer along the tube wall is at first laminar and then changes through a transition to the turbulent condition causing a dip in the Nu–x/D curve. The length of the dip decreases with the increase of Reynolds number. The length of this dip is very short (about 25 diameters) in the turbulent region and its effect on the average heat transfer is much less than on the local heat transfer. However, the length of the dip in the transition region is much longer than that in the turbulent region. For our experiments with a fixed inside diameter of 1.584 cm, the length of the dip varied from $x/D = 100$ to 175 depending on the level of disturbance at the inlet and the transition flow Reynolds number. This causes a significant influence on both the local and the average heat transfer characteristics of the tube.

Nomenclature

c_p	specific heat of the test fluid evaluated at T_b, J/(kg K)
D	inside diameter of test section (tube), m
Gr	local bulk Grashof number [$= g\beta\rho^2 D^3 (T_w - T_b)/\mu^2$], dimensionless
g	acceleration of gravity, m/s^2
h	local average peripheral heat transfer coefficient, W/(m^2 K)
k	thermal conductivity of the test fluid evaluated at T_b, W/(m K)
L	length of the test section (tube), m
Nu	local average peripheral Nusselt number ($= \mathrm{h}D/k$), dimensionless
Pr	local bulk Prandtl number ($= \mu c_p/k$), dimensionless
Re	local bulk Reynolds number ($= \rho VD/\mu$), dimensionless
T_b	local bulk temperature of the test fluid, °C
T_w	local average peripheral tube inside wall temperature, °C
V	average velocity in the test section, m/s
x	local distance along the test section from the inlet, m

Greek symbols

β	coefficient of thermal expansion of the test fluid evaluated at T_b, K^{-1}
μ	absolute viscosity of the test fluid evaluated at T_b, Pa s
ρ	density of the test fluid evaluated at T_b, kg/m^3

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