EXPERIMENTAL INVESTIGATION OF SINGLE-PHASE HEAT TRANSFER IN HORIZONTAL MINI-TUBES WITH DIFFERENT BENDING INLETS

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Abstract. An experimental setup was constructed for this study to measure the heat transfer of the horizontal mini-tubes of 0.8 mm and 2 mm inner diameters under the uniform wall heat flux boundary condition. Three inlet configurations, hydrodynamically fully developed length, 90° bend, and 180° bend, were used in this study. The bending inlets were arranged horizontally and vertically. The Reynolds number range for the entire experiment was approximately between 800 and 11,000. From the heat transfer results, for the two tubes with different inner diameters, the heat transfer characteristics from laminar to turbulent region for the straight tube inlet was matched with those of the macro tube. However, the start of transition was inlet dependent and the bending inlet delayed the start of transition when compared with the hydrodynamically fully developed length. From the entrance to the fully developed region, the laminar region entry length was almost at the length-to-inside diameter ratio of 150 and the laminar heat transfer of all the horizontal and the vertical 90° and 180° bends was observed to be higher than that of the hydrodynamically fully developed length prominently in the entrance region.

1. Introduction

To design the devices such as miniature heat exchangers, pumps, or sensors, the understanding of the heat transfer inside the small scale tubes or channels is required. Due to the importance of investigating heat transfer inside the mini- and micro-tubes, Krishnamoorthy et al. [1] and Celeta [2] reviewed the published research works done on heat transfer in the micro-tubes and channels. Based on the review works of [1, 2], the past studies seldom discussed the effect of inlet on heat transfer inside the mini- and micro-tubes. However, the effect of inlet on laminar and transition regions heat transfer was observed in the macro-tubes [3-6]. In the macro-tube studies [3-5], it was clearly shown that the start and end of transition and the heat transfer characteristics in the transition region were influenced by three different inlet configurations, re-entrant, square-edged, and bell-mouth, installed before the horizontal plain tube with an inner diameter of 15.8 mm. Recently, Dirker et al. [7] analyzed the effect of different inlets (sudden contraction, bell-mouth, and swirl inlets) on heat transfer and friction factor inside the micro-and mini-channels of 0.57, 0.85, and 1.05 mm in hydraulic diameter. They concluded that the heat transfer and friction factor behaviors, especially, in the transition region, were influenced by the inlet configuration.

Besides of the above-mentioned inlets, bending inlets with different angles are also used in the traditional heat exchangers and the effect of the bends on heat transfer in macro-tubes can be found in some experimental studies [8-12]. Practically, the bending entrance before a straight tube is also important in miniature heat exchangers. Recently, Tam and his coworkers [13] analysed the effect of bending entrance on the mini-tube heat transfer with the 90^{o} and 180^{o} bends. However, their study was limited to the single diameter tube and the horizontal bending entrance. Therefore, the main objective of this experimental study is to investigate the effect of the inlets with various angles of horizontal and vertical bends on heat transfer characteristics inside the mini-tubes with different diameters in the laminar, transition, and turbulent regions.

2. Experimental Setup

An overall schematic for the experimental test apparatus is shown in Figure 1. The working fluid used in this study is distilled water. As the pressurized nitrogen is fed into the pressure vessel, the distilled water is forced out of the pressure vessel, and through the flow meter array and test section. Flow rate of the water entering the array is further regulated using a metering valve and is measured by a Coriolis liquid flow meter. After the working fluid passes through the flow meter, fluid enters the test section assembly. In the test section assembly, the entire straight tube with different inlets are arranged between the fittings of inlet and outlet thermocouples.

As shown in Figure 1, electric copper wires were soldered on to both ends of the test tube. A DC power supply was used to provide the uniform wall heat flux boundary condition. The range of wall heat flux of this study was from 9 kW/m^2 to 459 kW/m^2. For the temperature measurements, the inlet and exit bulk temperatures were measured by means of thermocouple probes (Omega TMQSS-125U-6) placed before and after the test section. Also, for the heat transfer experiments, the thermocouples (Omega 5TC-TT-T-40-36) were attached with the adhesive pads (Omega TAP) along the test section. All the thermocouples and thermocouple probes were calibrated by a NIST-calibrated thermocouple probe (±0.22°C) and a PolyScience constant temperature circulating bath. Therefore, the temperature sensors were as accurate as ±0.22°C. For data acquisition, a National Instruments SCXI-1000 data collection system was used. All digital signals–from the flow meter and thermocouples were acquired and recorded by the Windows-based PC with a self- developed LabVIEW program.

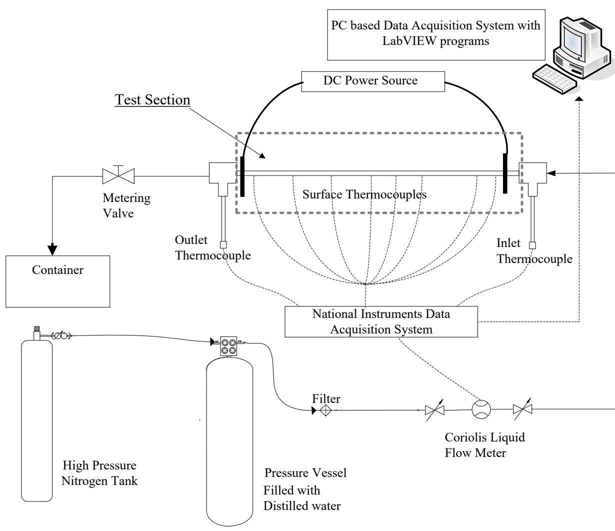
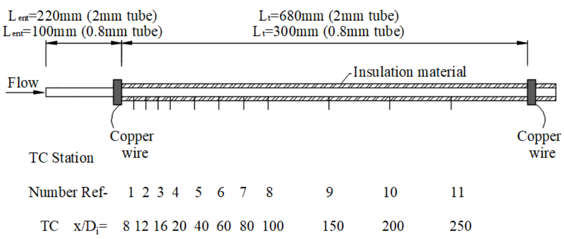


Figure 1. Schematic diagram of heat transfer measurement system.

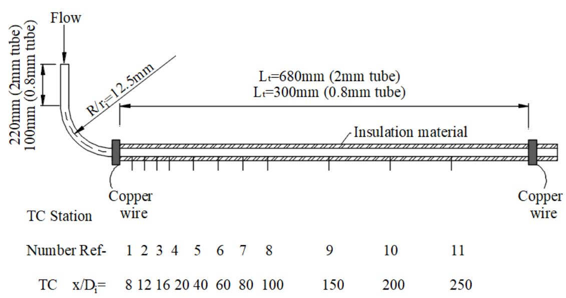
In this study, the test section was the horizontal stainless circular mini-tubes with two diameters: (i) 2 mm inside diameter and (ii) 0.8 mm inside diameter. For 2 mm and 0.8 mm tubes, the total length of the heating section (L_t) was 680 mm and 300 mm, providing a maximum heating length-to-inside diameter ratio (L_t/D_i) of 340 and 375, respectively. The tolerance of the diameters of the tested tubes measured by Scanning Electron Microscope (SEM, S3400N, HITACHI) was ± 0.12 mm (for 2 mm tube) and ± 0.03 mm (for 0.8 mm tube). The average roughnesses measured by Atomic Force Microscope (AFM, XE7, Park-Systems) was 0.63 µm (for 2 mm tube) and 0.998 µm (for 0.8 mm tube).

Figures 2 (a-d) show the three types of inlet configurations (straight tube, 90° bend, and 180° bend) of two different diameters installed before the horizontal tested tube and the arrangement of the thermocouples (TC) on the test section. In Figure 2(a), there is a straight tube length arranged horizontally before the heating section. In Figures 2 (b & c), the 90° and 180° bends, which are arranged horizontally and vertically in different tests, are formed by bending a straight tube. For the vertical bending inlet, the fluid flowed downward from the inlet to the horizontal test tube. For both bend orientations, the heating section is placed horizontally on the test table. The curvature ratio (R/r_i) of the bends is 12.5, where R is bend radius and r_i is the inner radius. To ensure a hydrodynamically developed flow condition, a certain length of straight tube is arranged before each bend. Figure 2 (d) shows the arrangement of the two thermocouples TC_{top} and TC_{bottom} (one on the top and one on the bottom of the tube) placed at each measuring station along the straight length of the mini-tube. After installation of the thermocouples, the tested tube was covered by self-adhesive elastomeric insulating material.

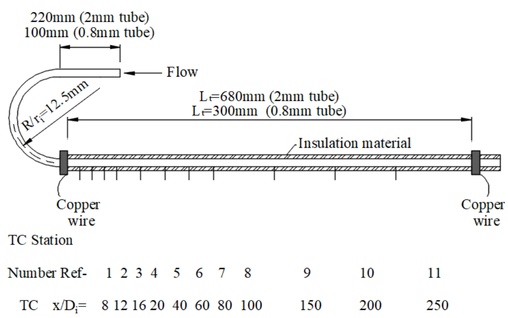
From the local peripheral wall temperature measurements at each axial location, the inside wall temperatures and the local heat transfer coefficients were calculated by the method shown in [14]. Also, the dimensionless numbers, such as Reynolds, Prandtl, Grashof, and Nusselt numbers, were computed by the computer program developed by [14]. The range of Reynolds number for this study was from 800 to 11,000. Heat balance errors was less than ±12%. The primary method was the one used in the computer program [14] for all heat flux and heat transfer coefficient calculations. By the calculation method based on Kline and McClintock [15], the maximum uncertainty of the heat transfer coefficients over the entire range of Reynolds numbers was ±15.5%.



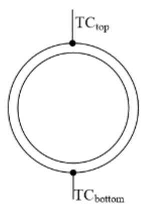
(a) Straight tube length



(b) 90° bend



(c) 180° bend



(d) Top and bottom thermocouples at each station

Figure 2. Test sections with different inlets and thermocouples

3. RESULTS AND DISCUSSION

To verify the new experimental setup, experiments with the straight tube length were conducted first. Figure 3 shows the comparison of the fully developed heat transfer data (at the x/D_i = 200) of the current 2 mm and 0.8 mm tubes with the data of Ghajar and Tam [3] (at the x/D_i = 200) for a 15.8 mm stainless steel tube with square-edged and re-entrant inlets. Basically, the present laminar, transition, and turbulent heat transfer data could follow the data trend of [3]. Especially, the start of transition and transition trend of the current data were closer to the re-entrant inlet data although the different tube size and inlet were used in the current and past studies. Therefore, the experimental setup and the heat transfer data were confirmed to be reliable. In the figure, it should be noted that the parallel shift from the classical fully developed value of Nu = 4.364 for the uniform wall heat flux boundary condition in the laminar region is due to the buoyancy effect [3, 5] for 2 mm tube and the roughness effect [16] for 0.8 mm tube.

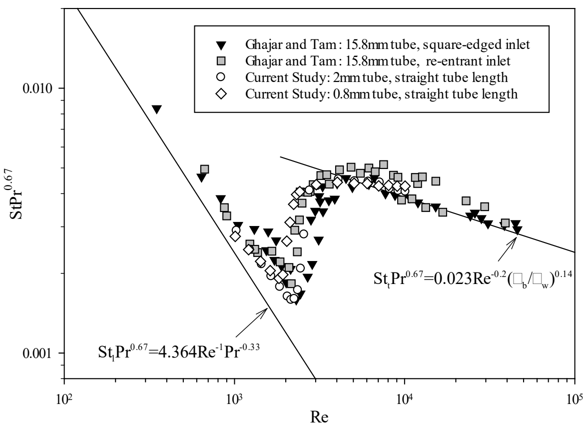
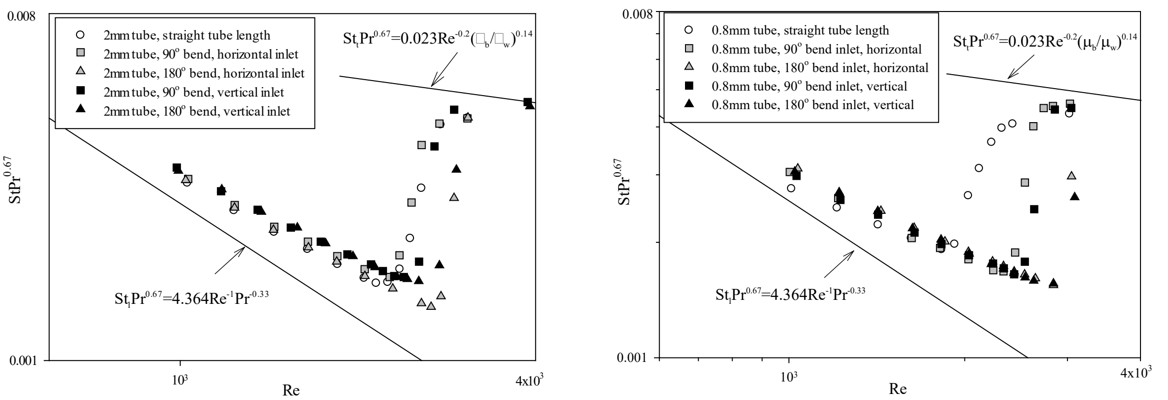


Figure 3. Comparison of the present heat transfer data with the experimental data of [3].



(a) 2 mm tube (b) 0.8 mm tube

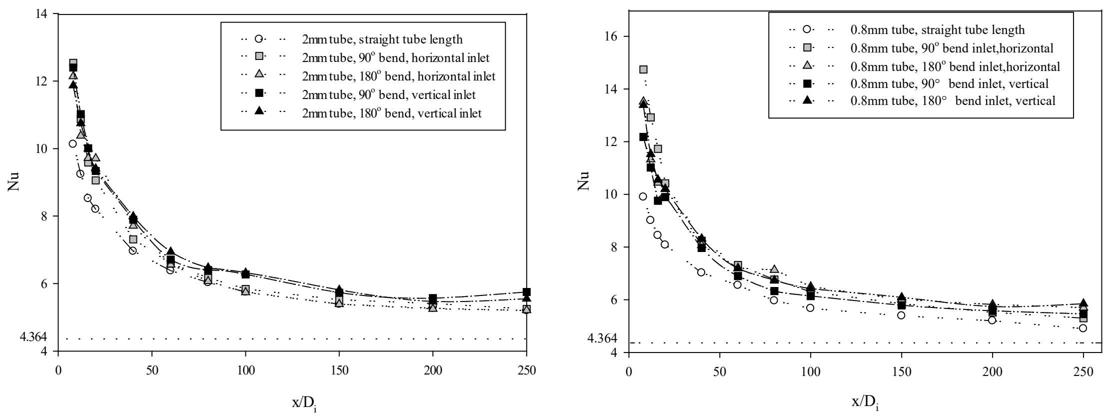
Figure 4. Heat transfer characteristics for the five different inlets at x/D_i of 200.

Figures 4(a & b) show the heat transfer data of the 2 mm and 0.8 mm diameter tubes with the straight tube inlet and the four 90° and 180° bending inlets (horizontal and vertical). As seen in the figure, the start of transition for those inlets is defined as the point of sudden departure from the laminar Nu line parallel to the classical Nu = 4.364 line. The transition Reynolds numbers for the start of transition are summarized in Table 1. Referring to Figure 4(a) and Table 1, for the 2 mm tube, it can be observed that the start of transition is found to be inlet dependent. It can be observed that the start of transition Reynolds number for each horizontal and vertical inlet at the 2 mm tube is determined to be about 2100 for the straight tube length, 2300 for the 90º bend, and 2600 for the 180º bend inlets. Referring to Figure 4(b) and Table 1, for the 0.8 mm tube, it can be observed that the start of transition is also found to be inlet dependent. It can be observed that the start of transition Reynolds number for each horizontal and vertical inlet at the 0.8 mm tube is determined to be about 1800 for the straight tube length, 2300 for the 90º bend, and 2800 for the 180º bend inlets. Regardless of the horizontal or vertical bending inlets, the results indicate that the larger bending angle inlet delays the start of transition when the straight tube length is treated as the base case. The delay of start of transition by the bending inlet may be explained as the fewer disturbances due to a momentum flow through the bend. After the start of transition, the transition trends for those inlets move upward in different ways.

Table 1. Start of transition for the five different inlets of 2 mm and 0.8 mm tubes at x/D_i of 200.

Inlet		2 mm Tube Heat Transfer		0.8 mm Tube Heat Transfer	
		$\mathbf{Re_{start}}$	$\mathbf{StPr^{0.67}}$	$\mathbf{Re_{start}}$	$\mathbf{StPr^{0.67}}$
Horizontal	Straight tube	2149	$1.587e^{-3}$	1826	$1.914e^{-3}$
	90°bend	2264	$1.651e^{-3}$	2331	$1.680e^{-3}$
	180°bend	2662	$1.381e^{-3}$	2819	$1.546e^{-3}$
Vertical	90°bend	2373	$1.875e^{-3}$	2333	$1.710e^{-3}$
	180°bend	2517	$1.561e^{-3}$	2838	$1.562e^{-3}$

Figure 5 shows the variation of local Nusselt number in the laminar region. As shown in Figures 5 (a & b), for all five inlets, the local Nusselt number first decreases along the tube and then stays fairly constant after x/D_i of about 150. Before the x/D_i of 150, it is obvious that the local heat transfer coefficients of the 2 mm and 0.8 mm test tubes with bend inlets are higher than that of the straight tube inlet. This is mainly due to the heat transfer enhancement by the complicated flow out from the bend inlets. Especially, the superposition of the secondary flow on the primary and tertiary flows of the horizontal tube downstream from the 180º bend (or U-bend) was taken into consideration for the heat transfer in the laminar region [10]. After the x/D_i of 150, it is observed that the local heat transfer coefficients of the 2 mm and 0.8 mm test tubes with bend inlets are slightly higher than that of the straight tube inlet. From the results, it is also observed that the heat transfer is not further increased by the vertical bending inlets when compared to the horizontal bending inlets.



(a) 2 mm tube (b) 0.8 mm tube

Figure 5. Comparison of variation of local Nusselt number with length of the different inlets in the range of the Reynolds number of 892 and 1061.

4. Conclusions

From this study, the following conclusions can be made:

- For the 2 mm and 0.8 mm tubes, the start of transition was inlet dependent.
- Regardless of the type of inlet, the similar local Nusselt number behaviour was observed in the laminar region and the entry length was around the length-to-inside diameter ratio (x/D_i) of 150.
- In the laminar region, the local heat transfer coefficients for the bend inlets was higher than that of the straight tube length inlet, especially, in the entrance region.

5. Acknowledgements

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6. References

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