Experimental Analysis of the Single-Phase Heat Transfer and Friction Factor inside the Horizontal Internally Micro-Fin Tube

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Abstract— **To increase heat transfer, internally micro-fin tubes are widely used in commercial HVAC applications. It is commonly understood that the micro-fin enhances heat transfer but at the same time increases the pressure drop as well. In the previous studies, majority of the works were focused on the development of correlations in a particular flow regime, especially in the turbulent region. There are only a few works that fundamentally studied the continuous change in the characteristic behavior of heat transfer and pressure drop from laminar to transition and eventually the turbulent regions. Therefore, more in-depth study is necessary. In this study, heat transfer and pressure drop were measured simultaneously in a single test section and compared with the data of a plain tube. From the results, the transition from laminar to turbulent was clearly established. The transition from laminar to turbulent was found to be inlet dependent. Furthermore, it was observed that the buoyancy effect is present in the laminar region. The efficiency index (the ratio of the heat transfer and the friction factor of enhanced tube to those variables for the plain tube) was examined and it was found to have a value larger than one when Reynolds number is larger than 5000 regardless of the type of inlet configuration used. Therefore, the application of the micro-fin tube used in this study is suitable when Reynolds number is larger than 5000.**

Keywords-heat transfer; friction factor; micro-fin tube

I. INTRODUCTION

Single-phase liquid flow in internally enhanced tubes is becoming more important in commercial HVAC applications, where enhanced tube bundles are used in flooded evaporators and shell-side condensers to increase heat transfer. This enables water chillers to reach high efficiency, which helps mitigate global warming concerns of HVAC systems. One kind of internally enhanced tube is the micro-fin tube. Jensen and Vlakancic [1] defined the micro-fin tube to have a height less than $0.03D_i$ (i.e. $2e/D_i<0.06$), where D_i is the inside diameter and e is the fin height. Basically, such kind of tube is widely used in high flow rate applications because the heat transfer enhancement in high flow rates (turbulent region) is more pronounced than that in the low flow rates (laminar region). Khanpara et al. [2] for turbulent heat transfer in micro-fin tubes reported an increase of 30 to 100% with Reynolds numbers between 5,000 and 11,000. Brognaux et al. [3] indicated that there was a 65 to 95% increase in heat transfer for the micro-fin tube over the smooth tube. However, there is also a 35 to 80% increase in the isothermal pressure drop. The work of [1] indicated that the micro-fins increased heat transfer ranging from 20 to 220% in the turbulent flow region. However, there was a penalty due to friction factor increase ranging from 40 to 140%. Webb et al. [4] calculated the "efficiency index", defined as the ratio of the heat transfer and the friction factor of enhanced tube to those variables for the plain tube, to vary from 0.98 to 1.18 for the seven different micro-fin tubes with Reynolds numbers between 20,000 and 80,000. For the laminar flow, several researchers [3, 5-7] concluded that the heat transfer and pressure drop were not greatly affected by micro-fins. Trupp and Haine [8] indicated that the secondary flow inside the tube with longitudinal fins was insignificant in the laminar flow and the thermal entry length was shown not to be relevant to the fin geometry.

The tube side roughening increases heat transfer surface area resulting in high efficiency heat exchangers. The increase in the surface area causes low flow rates in the heat exchanger tubes, resulting in the flow to be at Reynolds numbers that are between laminar and turbulent, that is in the transition region. Owing to the high efficiency requirements, it is likely that more HVAC units will operate in the transition region where the understandings of the heat transfer and pressure drop is limited. In the study of [5] the sudden changes in the heat transfer and friction factor with Reynolds number was observed. Different tube size had different onset point of transition. Moreover, the effect of heat transfer on pressure drop is significant within the transition region. However, the end of transition was not evaluated in that study. Also, there was no further discussion on transition region in that study. Jensen and Vlakancic [1] observed that the micro-fin tubes represented a long transitional period in friction factor before becoming fully turbulent at Re ≅ 20,000. During the transitional period, the friction factors are insensitive to Reynolds number. Also, this behavior was observed in [3, 7, 9].

According to the plain tube studies done by Ghajar and his coworkers [10-12], the heat transfer and friction factor characteristics, the buoyancy effect, the entry length effect, the start and end of transition, and the effect of heating on friction factor are the important points to be considered in transition flow. However, in the past studies with micro-fins, some of the above-mentioned points were always ignored. Therefore, the objective of this study is to analyze the heat transfer and friction factor characteristics of the micro-fin tubes in detail based on the heat transfer and pressure drop data measured simultaneously from a horizontal micro-fin tube.

II. EXPERIMENTAL SETUP

The heat transfer and pressure drop experimental data used in this study were obtained from the experimental apparatus shown in Figure 1. The local forced and mixed convective heat transfer measurements were made in a horizontal, electrically heated, copper circular straight tube under uniform wall heat flux boundary condition. As shown in Figure 2, two types of inlet configurations (re-entrant and square-edged) were installed before the test section. A calming and inlet section similar to that used in [10] was used to ensure a uniform velocity distribution at the entrance of the test section. Water and a mixture of water and ethylene glycol were used as the test fluid in the closed-loop experimental system. The DC arc welder provided the uniform heat flux boundary condition to the test section.

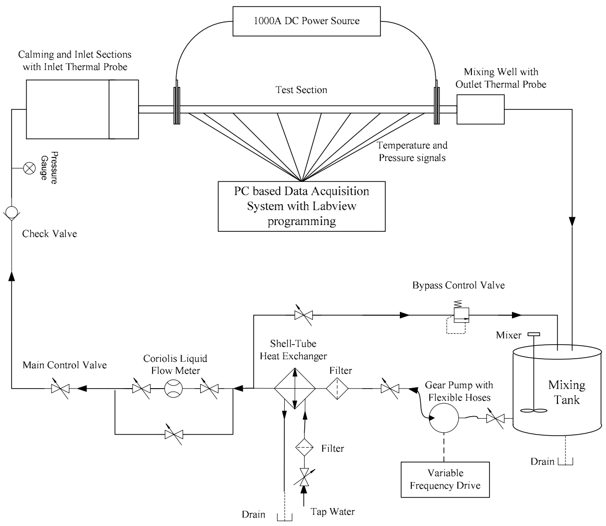


Figure 1. Experimental setup.

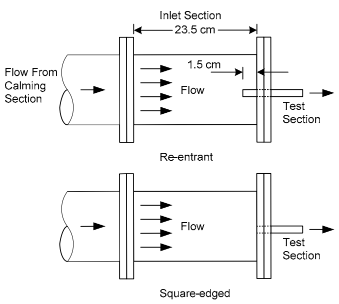
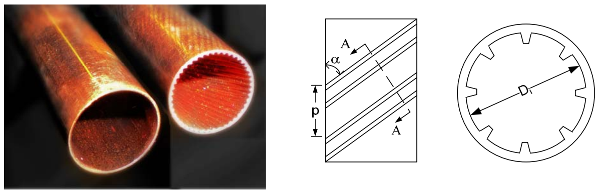


Figure 2. Type of inlet.

In this study, the plain tube and micro-fin tube (see Figure 3) were used as the test section. Table I represents the specification of the tubes. Both of the tubes had an inside diameter of 1.48 cm and an outside diameter of 1.58 cm. The total length of the test section was 6 m, providing a maximum length-to-inside diameter ratio of 403.

TABLE I. SPECIFICATIONS OF THE TEST SECTIONS

Tube Type	Tube Material	Outer Dia., D_o (mm)	Inner Dia., D_i (mm)	Spiral angle, α	Fin height, e (mm)	Number of starts, N_s
Smooth	Copper	15.88	14.88	–	–	–
Micro-fin	Copper	15.88	14.88	18°	0.5	25



(a) (b)

Figure 3. (a) Test tubes; (b) Sectional view of the micro-fin tube.

As shown in Figure 4, a pressure tap was also placed near the TC1 and TC2 thermocouples. 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 [13]. In these calculations, the axial conduction was assumed negligible (RePr > 4,200 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. Also, the dimensionless numbers, such as Reynolds, Prandtl , Grashof, and Nusselt numbers were computed by the computer program [13]. In the present study, the experiments covered a local bulk Reynolds number range of 1000 to 22000, a local Prandtl number range of 5.2 to 45.3, a local bulk Grashof number range of 4496 to 23797, a local bulk Nusselt number range of 11.1 to 252.9, and a friction factor range of 7.3×10^{-3} to 2.4×10^{-2}. The wall heat flux for the experiments ranged from 3.4 to 6.9 kW/m^2.

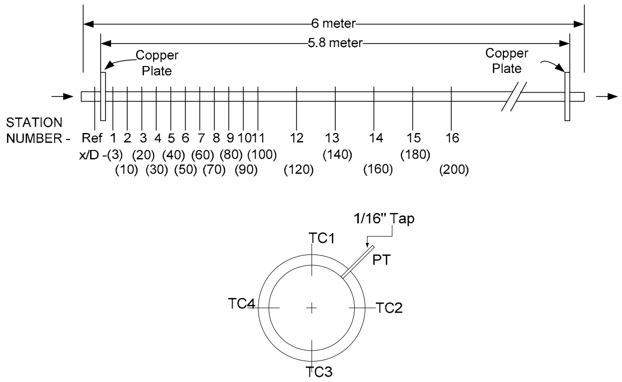


Figure 4. Arrangement of the thermocouples and pressure tap on the test section.

All the thermocouples and the pressure transducers were calibrated before the experimental runs. The high-precision Coriolis flowmeter was calibrated by the manufacturer. Table II lists the uncertainties of the measured variables and calculated variables over the entire range of Reynolds numbers studied. The calculation method is based on [14].

TABLE II. UNCERTAINIES IN EXPERIMENTAL DATA

Measured variables		Calculated variables	
Variable	**Uncertainty**	**Variable**	**Uncertainty**
Temperature	0.22 °C	C_f *(friction factor)*	2.1%
Mass flow rate	0.77% of full range (0.3-30USGPM)	*h (heat transfer coefficient)*	12.6%
Density	0.2 kg/m^3	*Nu (Nusselt number)*	12.6%
Diameter	0.02mm		
Length	0.1mm		
Voltage	1%		
Current	1%		

III. RESULTS AND DISCUSSION

To verify the new experimental setup and later to compare with the micro-fin tube data, experiments for plain tube were conducted first. Figure 5 shows the recently collected plain tube heat transfer data for the square-edged and re-entrant inlets compared with those collected by [10]. Since the deviations between the new data and old data were below 10%, the experimental setup and the data were confirmed to be reliable. It should be noted that the parallel shift from the classical fully developed value of Nu = 4.364 for uniform heat flux boundary condition in the laminar region is due to the buoyancy effect as will be explained later.

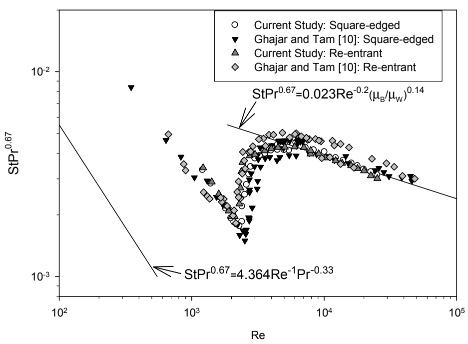


Figure 5. Heat transfer characteristics for the plain tube at x/D of 200.

The measurements of friction factor were also verified by comparing the recently collected data with the classical friction factor equations for laminar and turbulent flows. As shown in Figure 6, all the recently collected friction factor data matched well (within 10%) with the classical equations.

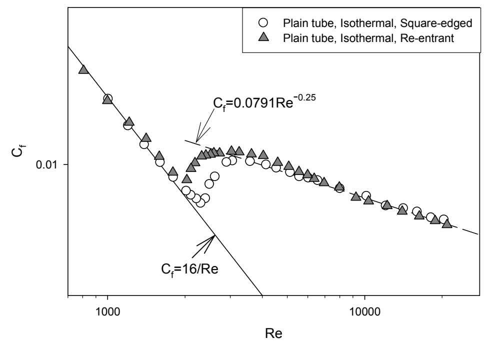


Figure 6. Friction factor characteristics for the plain tube at x/D of 200 under isothermal boundary condition.

After the verifications of the experimental setup, heat transfer and pressure drop data for the micro-fin tube were measured simultaneously in a single test section. The results are shown in Figure 7 for heat transfer and Figure 8 for friction factor, respectively. As seen in these figures, the start and end of transition for heat transfer (with uniform heat flux boundary condition) and friction factor (with isothermal and uniform heat flux boundary conditions) in plain and micro-fin tubes for the two inlet configurations (square-edged and re-entrant) were established. The transition Reynolds numbers for heat transfer and friction factor are summarized in Table III.

TABLE III. START AND END OF TRANSITION FOR PLAIN AND MICRO-FIN TUBES AT X/D OF 200.

Tube,	Heat Transfer				Friction Factor			
Condition	Re_{start}	$StPr^{0.7}$	Re_{end}	$StPr^{0.7}$	Re_{start}	C_f	Re_{end}	C_f
Plain, Isothermal (Square-edged)	-	-	-	-	2306	7.6e-3	3065	0.0102
Plain, Heating (Square-edged)	2298	1.7e-3	8357	4.1e-3	2200	7.4e-3	3585	0.0100
Plain, Isothermal (Re-entrant)	-	-	-	-	2032	9.0e-3	3031	0.0110
Plain, Heating (Re-entrant)	2001	2.0e-3	7919	4.12e-3	2012	7.6e-3	3250	0.0105
Micro-fin, Isothermal(Square-edged)	-	-	-	-	2675	8.4e-3	8006	0.0144
Micro-fin, Heating (Square-edged)	2841	1.5e-3	9264	7.5e-3	2764	7.3e-3	8104	0.0138
Micro-fin, Isothermal(Re-entrant)	-	-	-	-	2057	0.0128	8022	0.0143
Micro-fin, Heating(Re-entrant)	2215	1.7e-3	8282	7.8e-3	2167	9.2e-3	7997	0.0143

For heat transfer, it can be clearly seen from Table III and Figure 7 that the lower transition is delayed for micro-fin tube and the lower transition Reynolds number is 2,841 instead of 2,298 for the plain tube fitted with the square-edge inlet. Same observations can be made for the re-entrant inlet and the lower transition Reynolds number is 2,215, instead of 2,001. For the upper transition for heat transfer, it was observed that the transition is also delayed for micro-fin tube. It is also obvious that the onset of the transition is inlet dependent. Referring to Figure 7, it is observed that the heat transfer behavior in the upper transition region for the plain and micro-fin tubes is very different. Around Reynolds number of 4,000 the plain tube data follows the Sieder and Tate correlation [15] for the turbulent region. However, the micro-fin data continues to increase up to a Reynolds number of about 8,000 and then a parallel shift from the Sieder and Tate correlation. This increase in the Colburn j factor (St $Pr^{0.67}$) and the parallel shift from the Sieder and Tate correlation in the turbulent region is due to the swirling motion induced by the micro-fin. It should be noted that the end of the transition for micro-fin tube is defined as the first point where the heat transfer data landed on a line parallel to the Sieder and Tate correlation.

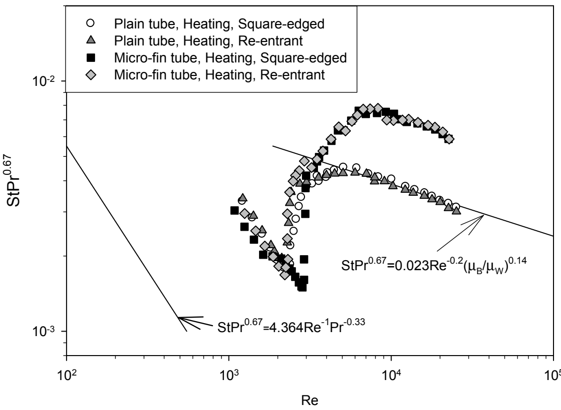


Figure 7. Heat transfer characteristics for plain and micro-fin tubes.

For the friction factor, as seen in Figure 8, in the laminar region, the friction factor for micro-fin tube, either heated or isothermal, exhibits different behavior from the plain tube. For the micro-fin tube a parallel shift from the classical laminar equation was observed. The laminar friction factor for heated or isothermal micro-fin tube has the same trend in the laminar region. In the other words, the friction factor in that region is insensitive to the different boundary conditions. For the isothermal data, the lower transition Reynolds numbers of the micro-fin tube (2,675 and 2,057) for the square-edged and reentrant inlets are different from the plain tube (2,306 and 2,032) and late transition was observed. Comparing the isothermal and heated friction factor for micro-fin tube, it can be observed that heating delays the start of transition. In the transition region, the friction factors go through a steep increase followed by a constant behavior and then a parallel shift from the classical Blasius turbulent flow friction factor correlation. Similar results were also observed in [1, 3, 7, 9]. It should be noted that the end of the transition for micro-fin tube is defined as the first point where the friction factor data landed on a line parallel to the Blasius correlation. Compared with [1], where the transition ended at a Reynolds number of about 20,000, the end of transition in the present study was at a Reynolds number of about 8,000.

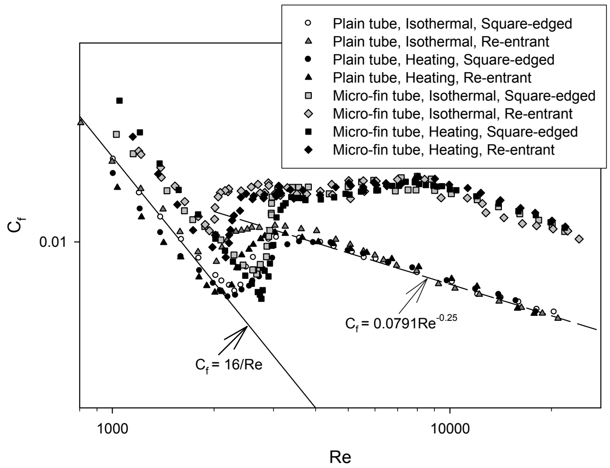


Figure 8. Friction factor characteristics for the plain and micro-fin tubes at x/D of 200 under isothermal and heating boundary conditions.

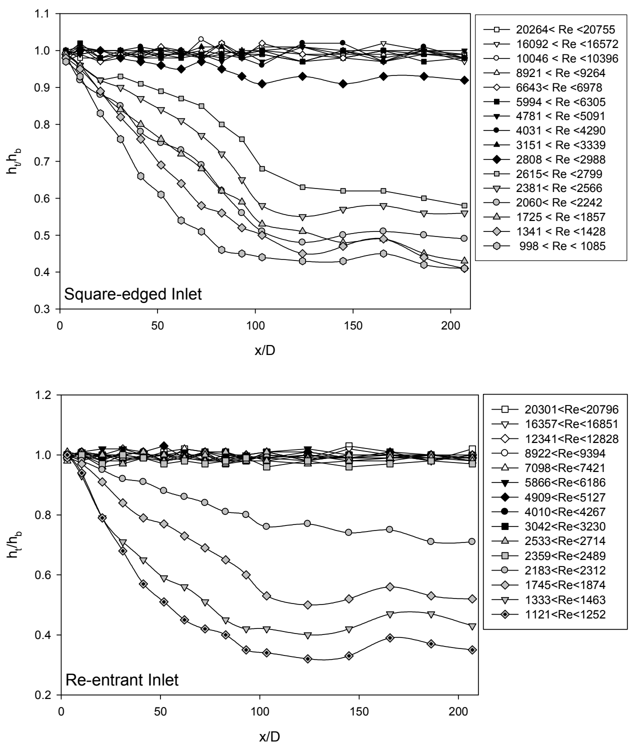


Figure 9. The ratio of heat tranfer coefficients at the top and bottom on the micro-fin tube with different inlets.

As seen in Figure 4, four thermocouples were used at each location. Ghajar and Tam [10] established when the ratio of the heat transfer coefficient of the top and bottom is less than 0.8, strong buoyancy effect is present. As seen in Figure 9, for the micro-fin tube with square-edged inlet when the Reynolds number is less than 2,800, the ratio started at one and decreased rapidly and finally stabilized when x/D was larger than 100. Recall from Table III that the lower transition Reynolds number for this inlet was delayed to 2,841 from 2,298 for the plain tube. This delay in the transition is the direct effect of the precence of buoyancy forces in the laminar region and it is also the main reason for the parallel shift from the classical Nu value of 4.364 observed in Figures 5 and 7. No buoyancy effect was observed in the transition and turbulent regions. For the micro-fine tube with a re-entrant inlet, the buoyancy effect occurs when the Reynolds number is less than 2,300. As shown in Table III, the lower transition Reynolds number for this inlet was delayed to 2,215 from 2,001 for the plain tube.

The efficiency index, $\eta = (j/C_f)_{micro-fin}/(j/C_f)_{plain}$ was calculated for the measured Reynolds number range and is plotted in Figure 10. It was observed that the index is larger than one when Reynolds number is larger than 5,000. Therefore, it can be concluded that micro-fin tube should not be used in the laminar and even the lower transition regions.

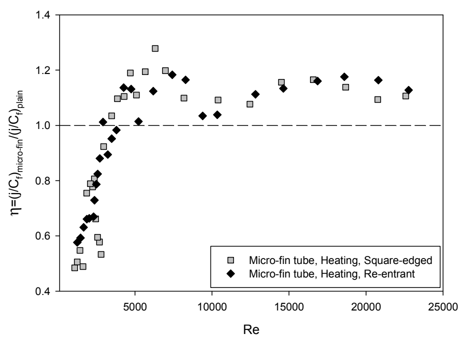


Figure 10. Efficiency index.

IV. CONCLUSIONS

In this study, heat transfer and friction factor for horizontal plain and micro-fin tubes were obtained simultaneously under uniform wall heat flux boundary conditions. From the results, the transition regions for heat transfer and pressure drop were obtained. For the heat transfer, the transition is inlet dependent and it has a broader transition region in comparison to the plain tube. In addition, a parallel shift in the heat transfer data due to the swirling motion induced by the micro-fin was observed in the turbulent region. For the micro-fin tube friction factor, a parallel shift in the laminar and turbulent regions was observed. The transition region for the friction factor is composed of a steep increase followed by a constant behavior and comparable with the observations of other researchers. Moreover, buoyancy effect is only present in the laminar region. Finally, from the calculated efficiency index values it was concluded that when the Reynolds number is larger than 5,000, regardless of the inlet configuration used, the efficiency index is always greater than 1.

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