An Experimental Study on Laminar Heat Transfer in Flat Aluminum Extruded Tubes Having Small Hydraulic Diameter

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ABSTRACT: Laminar heat transfer experiments were conducted in flat extruded aluminum tubes. Three different flat tubes-two with smooth inner channel, one with micro-finned inner channel-were tested. Smooth tube data were in reasonable agreement with the predictions by simplified theoretical models. The heat transfer coefficients of the micro-fin tube were significantly smaller than those of the smooth tube. The reason was attributed to the decelerating flow in the inter-fin region. Heat transfer correlations were developed from the data.

Nomenclature

A : heat transfer area [mm^2]
A_c : cross-sectional flow area [mm^2]
A_i : total internal surface area [mm^2]
b : thickness of the flat tube [mm]
c_p : specific heat [J/kg/K]
D_h : hydraulic diameter [mm]
e : fin height [mm]
G : mass flux [kg/m^2/s]
Gz : Greatz number, $\pi D_h RePr/4L$ [dimensionless]
h : heat transfer coefficient [W/m^2/K]
k : thermal conductivity [W/m/K]
L : length of the test section [mm]
Nu_{Dh} : Nusselt number based on hydraulic diameter, hDh/k [dimensionless]
p : fin pitch [mm]
Pr : Prandtl number, $\mu c_p / k$[dimensionless]
P_w : wetted perimeter [mm]
Re : Reynolds number, GD_h/μ_l [dimensionless]
r_b : fin base radius [mm]
r_o : fin tip radius [mm]
T : temperature [K]
t : tube wall thickness [mm]
U : overall heat transfer coefficient [W/m^2/K]
w : width of the flat tube [mm]

Greek Symbols

γ : apex angle of the fin [deg]
μ : dynamic viscosity [kg/m/s]
ρ : density [kg/m^3]

Subscripts

i : tube-side
l : liquid
m : mean
o : annular-side

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1. Introduction

Fin-and-tube heat exchangers having circular copper tubes and aluminum fins have widely been used in air-conditioning and refrigeration applications. However, they have inherent shortcomings such as thermal contact resistance between fins and tubes, low heat transfer region at the downstream of the tube, etc. Aluminum heat exchangers, which are made by brazing flat aluminum tubes and folded aluminum louver fins, are known to significantly improve the thermal performance by eliminating the thermal contact resistance and reducing the low heat transfer region downstream of the tube. Webb and Jung[1] have shown that the aluminum heat exchanger can reduce the heat exchanger volume by 50%, and the refrigerant charge by 30% as compared with the conventional fin-and-tube heat exchanger. The aluminum heat exchangers have been used as condensers of the automotive air-conditioning system, and are recently adopted as condensers of the household air-conditioning system.

To design an air-conditioning heat exchanger properly, it is important to accurately assess the air-side heat transfer coefficient. The flow and heat transfer behavior on the air-side is very complex, which limits theoretical or numerical predictions. The air-side heat transfer coefficient is usually obtained by testing a sample heat exchanger. The test procedure includes set-up of the sample heat exchanger in a wind tunnel, and circulation of hot water into the tube-side.[2] The air-side heat transfer coefficient is obtained by subtracting the tube-side heat transfer coefficient (or tube-side thermal resistance) from the measured total thermal resistance. Thus, it is necessary to know the tube-side heat transfer coefficient a priori. In addition, it is important to minimize the tube-side thermal resistance to improve the accuracy of the air-side heat transfer coefficient. The tube-side heat transfer coefficient is proportional to the tube-side Reynolds number, and thus, the Reynolds number should be maintained large to reduce the thermal resistance. Generally, the tube-side thermal resistances do not exceed 10% of the total thermal resistance.[2] The hydraulic diameter of the flat aluminum extruded tube is 1 mm~2 mm, which is much smaller than that of the conventional copper tube. Reynolds number is proportional to the diameter, and to maintain the same Reynolds number as that in the fin-and-tube heat exchanger, the flow velocity in the aluminum heat exchanger should increase in proportion to the diameter ratio. This requires an unacceptable pump specification, which results in the tube-side Reynolds number during the heat exchanger test usually laminar.

The laminar heat transfer coefficient is, different from the turbulent flow heat transfer, sensitive to the channel geometry, boundary condition, etc. In this study, laminar heat transfer coefficients were obtained for three flat aluminum tubes-two with smooth inner channel, one with micro-finned channel. Data are also compared with theoretical predictions.

2. Experimental methods

Cross-sectional photos of the sample tubes are shown in Fig. 1. Detailed specifications of the tubes and the annular flow channels are listed in Table 1. Tube A has 10 smooth inner channels, and the width and height are 20mm and 2 mm, respectively. The corresponding hydraulic diameter is 1.33 mm. Tube B has 7 smooth inner channels, 18 mm width and 2 mm

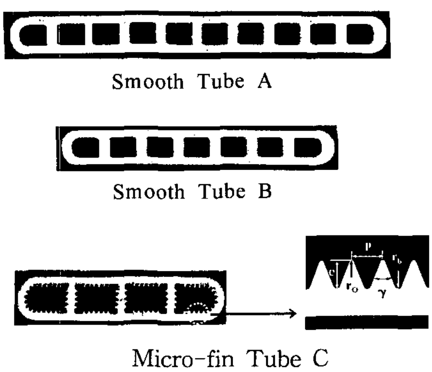

Fig. 1 Flat aluminum tubes tested in this study.

Table 1 Geometric details of flat tubes and test section

Item	Smooth Tube A		Smooth Tube B		Micro-fin Tube C	
	tube	annulus	tube	annulus	tube	annulus
w(mm)	20	22.3	18	20.3	16	18
b(mm)	2	4.2	2	4.2	3	5.0
$A_c(mm^2)$	16.08	48.7	13.0	51.6	22.7	39.6
P_w(mm)	43.4	92.3	38.4	84.0	58.0	77.3
D_h(mm)	1.33	2.11	1.35	2.46	1.56	2.05
t(mm)	0.4	-	0.4	-	0.5	-

height. The hydraulic diameter 1.35mm is approximately the same as that of Tube A. Tube C has 4 micro-finned channels, 16 mm width and 3 mm height. The micro-fin height (e) is 0.2mm with apex angle (γ) 40 degree. The mocro-fin pitch (p) is 0.4 mm and fin tip radius(r_o) is 0.013 mm and fin base radius(r_b) is 0.15mm. These tubes are currently used as automotive condensers.

A schematic drawing of the apparatus is shown in Fig. 2. The apparatus consists of the test section and two constant temperature water supply system. Each water supply system consists of a heat exchanger, a water heater, a pump and a float-type flow meter. A detailed drawing of the test section is shown in Fig. 3. The flat aluminum tube is mounted at the center of the test section, and an annular water channel is formed around the flat tube using inserts and water jackets. To obtain accurate tube-side heat transfer coefficients, it is important to reduce the thermal resistance on the annular-side. This was accomplished by maintaining the annular gap small (approximately 1mm). Detailed dimensions of the annular channels are listed in Table 1. The length of the test section is 455 mm.

The tube-side heat transfer coefficient was obtained by subtracting the annular-side heat transfer coefficient from the measured overall heat transfer coefficient. The annular-side heat transfer coefficient was obtained using the Wilson plot method. The Wilson plot method graphically determines the tube-side or annular-side heat transfer coefficient without measuring the tube wall temperature, and is widely used, especially when the tube wall temperature measurement is difficult. In this study, a

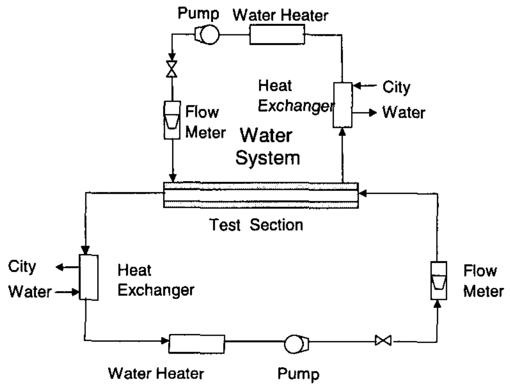

Fig. 2 Schematic drawing of the experimental apparatus.

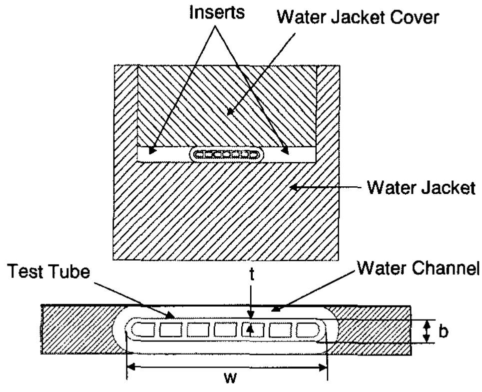

Fig. 3 Detailed drawing of the test section.

modified Wilson plot method proposed by Farrell et al.[3] was used to obtain the annular-side heat transfer coefficient. The method proposes a Sieder-Tate[4] type heat transfer correlation, and obtains relevant coefficients from experimental data. Experiments are conducted for a range of annular-side water velocities with tube-side water velocity and temperature fixed. This method has advantage over conventional Wilson plot that less experiments are necessary to obtain the heat transfer correlation. One thing to note about the Wilson plot is that both the annular-side and tube-side flow should be maintained turbulent. To promote turbulence on the annular-side, a thin Ni-Cr wire of 0.3 mm diameter was wrapped around the flat tube.

To obtain the overall heat transfer coefficient, inlet and outlet water temperatures and flow rates of each stream were measured. The water temperatures were measured using 5 thermocouple thermopiles, and the flow rate was measured using calibrated float-type flow meters. The tube-side heat transfer coefficient is obtained from the following equation.

$$h_i = \frac{1}{\left[\frac{1}{U_0} - \frac{1}{h_0}\right]\frac{A_i}{A_0} - \frac{tA_i}{kA_m}} \tag{1}$$

In Eq. (1), U_o is the overall heat transfer coefficient, h_o is the annular-side heat transfer coefficient, and A_m is the heat transfer area measured at the center of the tube wall. Experimental uncertainties were analyzed following the method by Kline and McClintock.[5] The uncertainty on the heat transfer coefficient ranged from ±4.1% to ±11.2%. The uncertainty increased as the Reynolds number decreased. During the test, the heat balance between the tube-side and the annular-side matched within 5%.

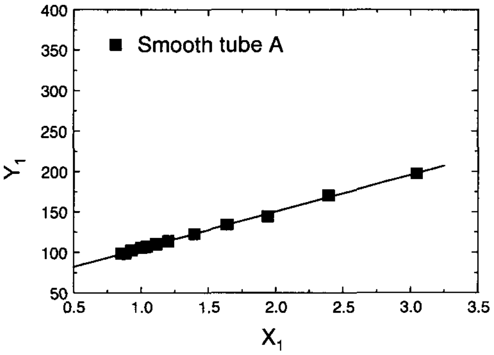

Fig. 4 Typical modified Wilson plot graph.

3. Results and discussions

3.1 Annular-side heat transfer coefficient

As discussed previously, modified Wilson plot tests were conducted to obtain the annular-side heat transfer correlations. During the test, the tube-side mean water temperature was maintained at 55 ℃, and the Reynolds number was maintained at 18,000. The annular-side Reynolds number was varied from 1,000 to 3,000 with the mean water temperature maintanied at 43 ℃, To promote turbulence at the annular-side, Ni-Cr wire of 0.3 mm diameter was wrapped around the flat tube at 3 mm pitch. Typical Wilson plot results are shown in Fig. 4, which corresponds to Tube A. In the figure, X_1 and Y_1 are the parameters to be optimized to obtain the exponent of the Reynolds number and proportional coefficient of the Sieder-Tate type correlation. The X_1 and Y_1 are defined as follows.

$$X_1 = A_i / [A_o (k_w / D_h)(Re_{Dh}^m \mathrm{Pr}^{1/3})] \tag{2}$$

$$Y_1 = \left(\frac{1}{U_o A_o} - \frac{1}{kA_m}\right) A_i \tag{3}$$

The final annular-side correlations are as follows.

smooth tube A : $N_{Dh} = 0.022 Re^{0.99} \mathrm{Pr}^{1/3}$ (4)

smooth tube B : $N_{Dh} = 0.032 Re^{0.92} \mathrm{Pr}^{1/3}$ (5)

micro-fin tube C : $N_{Dh} = 0.013 Re^{1.09} \mathrm{Pr}^{1/3}$ (6)

3.2 Tube-side laminar heat transfer coefficient

During the test, the tube-side mean water temperature was maintained at 35 ℃, and the annular-side mean water temperature was maintained at 55 ℃. The annular-side Reynolds number was maintained at 1,700, which corresponds to the flow rate of 1.0 l/min. Tests were conducted for the tube-side Reynolds number 300 to 1,000, which corresponds to tube-side water flow rate 0.1 l/min to 0.7 l/min depending on the flow cross-sectional area of the tubes. To measure the small flow rate accurately, the tube-side flow was drained and weighed every time using a precision balance. For the test range, the annular-side water temperature difference was from 3 ℃ to 9 ℃. The tube-side water temperature difference was from 13 ℃ to 30 ℃.

Fig. 5 shows the results for smooth tube A. The heat transfer coefficient increases as the Graetz number increases. The reason could be attributed to the increase of the flow developing region for the increased Graetz number. The heat transfer coefficient is higher at the flow developing region compared with that at the fully developed region. A theoretical predictions are also shown in Fig. 5. The present flow in the flat tube may be defined as both

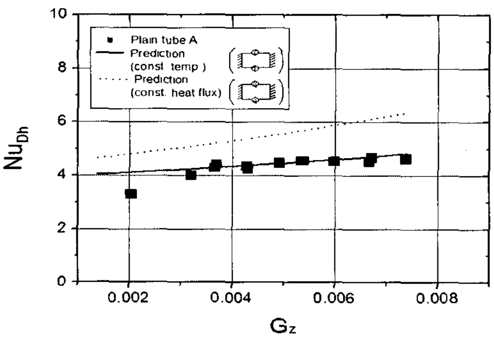

Fig. 5 Nusselt number of smooth flat tube A. Theoretical prediction is for simultaneously developing laminar flow in a simplified channel.

hydrodynamically and thermally (simultaneously) developing laminar flow with the wall temperature increasing from inlet to outlet. The literature does not provide appropriate theoretical solutions for the present flat tube geometry and the wall temperature boundary condition.

An attempt was made to simplify the channel geometry and the boundary condition. The flat tube has rectangular channels except for the two end channels. Thus, the rectangular channel was assumed to represent the flat tube channels. Two side-walls of the rectangular channel was assumed to be adiabatic because rectangular channels have common side-walls. Due to the experimental condition, the top and bottom wall temperature of the rectangular channel increase from inlet to outlet. However, the literature only provide two extreme solutions for the simultaneously developing flow - constant wall temperature and constant heat flux[(6)]. In this study, it is assumed that the ratio of the heat transfer coefficient at the fully developed region to that of the developing region is not affected by the temperature boundary condition. For example, if this ratio is 1.2 for the constant wall temperature condition, this ratio is assumed to apply to the other condition (e.g., two constant temperature walls and two adiabatic walls). This assumption appears plausible considering that the ratios between the constant wall temperature condition and the constant heat flux boundary condition agree within 3%.

Fig. 5 shows that the experimental data agree within 3.5% with the theoretical predictions for constant temperature top and bottom walls and two adiabatic side walls. Theoretical predictions for constant heat flux top and bottom walls highly overpredict the data. The tube wall temperature difference from inlet to outlet is estimated to be 3 ℃~9 ℃, which is much smaller (approximately 1/4) than the tube-side water temperature difference (13 ℃~30 ℃). It

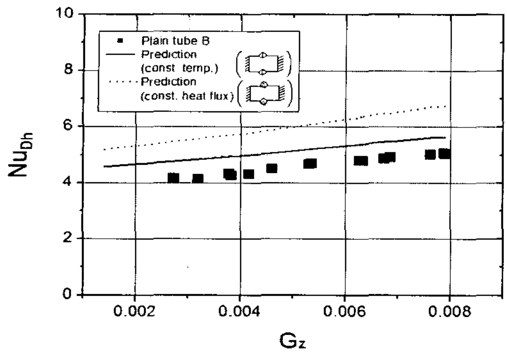

Fig. 6 Nusselt number of smooth D_h=1.35 mm Theoretical prediction is for simultaneously developing laminar flow in a simplified channel.

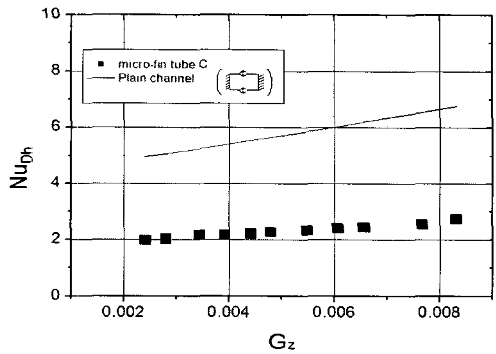

Fig. 7 Nusselt number of D_h=1.56 mm Theoretical prediction is for simultaneously developing laminar flow in a simplified smooth channel.

appears that the small temperature difference is simulated appropriately by the constant temperature boundary condition.

Fig. 6 shows the results for smooth tube B. Similar to the tube A, the heat transfer coefficient increases as the Graetz number increases. The constant temperature top and bottom wall boundary condition predicts the data within 13.6%.

Fig. 7 shows the results for micro-fin tube C. The heat transfer coefficient increases as the Graetz number increases. The literature doe not provide theoretical solution for laminar heat transfer in the micro-fin channel. Thus, theoretical prediction was made for the smooth rectangular channel having the same aspect ratio as the micro-fin channel. Fig. 7 shows that the smooth channel predictions significantly (approxiamtely 150%) overpredict the heat transfer coefficients in the micro-fin channel. The reason could be attributed to the low flow velocity in the inter-fin region. Similar trend was reported in circular micro-fin tubes.[7] The following correlations are obtained from the data.

smooth tube A : $Nu_{Dh} = 3.85 + 128.716 Gz$ (7)

smooth tube B : $Nu_{Dh} = 4.33 + 166.8388 Gz$ (8)

micro-fin tube C : $Nu_{Dh} = 1.70 + 114.966 Gz$ (9)

4. Conclusions

In this study, laminar heat transfer characteristics in flat aluminum tubes are experimentally investigated. Listed below are major conclusions.

(1) For flat tubes having smooth inner channels, the laminar heat transfer data are reasonably simulated by simplified theoretical models.

(2) The laminar heat transfer coefficient in the micro-fin flat tube is significantly lower than that in the smooth flat tube. The low flow velocity in the inter-fin region appears to be responsible.

(3) Laminar heat transfer correlations are proposed for the flat tubes.

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