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Jiyang Li
Department of Human and Engineered Environmental Studies, The University of Tokyo, lijiyangxjtu@gmail.com

Chaobin Dang
Department of Human and Engineered Environmental Studies, The University of Tokyo, dangcb@k.u-tokyo.ac.jp

Eiji Hihara
Department of Human and Engineered Environmental Studies, The University of Tokyo, hihara@k.u-tokyo.ac.jp

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Jiyang Li*, Chaobin Dang, Eiji Hihara
Department of Human and Engineered Environmental Studies, The University of Tokyo
5-1-5 Kashiwanoha, Kashiwa-shi, Chiba 277-8563, Japan
(TEL/FAX: +81-47-136-4360, lijiyang@hee.k.u-tokyo.ac.jp)

ABSTRACT

The characteristics of local heat transfer and pressure drop were investigated experimentally for the vertical up-flow boiling of refrigerant R1234yf in two types of aluminum multi-port extruded tubes having 16 channels with a cross-section of 0.91 × 0.21 mm (tube A) and 40 channels with a cross-section of 0.34 × 0.21 mm (tube B). At an evaporating temperature of 15 ℃, the local heat transfer coefficient and pressure drop for heat flux ranging from 3-16 kW/m^2 and mass flux ranging from 60-240 kg/m^{2}s were measured. The heat transfer coefficient almost linearly decreased with increasing vapor quality in both tubes. As the vapor quality increased, the dry patch area became lager in these rectangular channels, which deteriorated the heat transfer performance significantly. Compared with the multi-port extruded tube (tube C) having 16 channels with a cross-section of 0.9 × 0.9 mm, at a mass flux of 60 kg/m^{2}s, tube C demonstrated better heat transfer performance than tubes A and B. When the mass flux increased to 120 kg/m^{2}s, the dry-out was alleviated in tubes A and B and better performance was observed for low vapor quality regions, while in high vapor quality regions the dry patch reoccupied the heat transfer area, thereby resulting in worse heat transfer performance in tubes A and B. The pressure drop was not effectively predicted by any of the three correlations investigated.

1. INTRODUCTION

All-aluminum parallel multi-port heat exchangers are being developed for air conditioning systems to improve compactness and performance, and to reduce refrigerant charge amount, pressure drop, and total cost. When working as an evaporator, the refrigerant side flow boiling heat transfer and pressure drop performances are the crucial characteristics that need to be studied in-depth. Although numerous studies of flow boiling in mini-channels have been previously conducted, only a limited number have employed low heat and mass flux conditions that are typically found in air conditioning systems. As a result, the flow boiling mechanism inside multi-port extruded tubes of air conditioning systems has not yet been elucidated.

Beginning in 2017, the use of refrigerants with global warming potential (GWP) exceeding 150 in all types of mobile air conditioners will be banned in the EU market, which has triggered a research and development initiative for new low-GWP refrigerants. Recently, R1234yf, developed by Honeywell and DuPont, has emerged as a promising candidate because of its low GWP value of 4 and similar thermal properties to R134a (Saitoh et al., 2011). Due to increasing concerns regarding all-aluminum parallel multi-port heat exchangers, the heat transfer performance of R1234yf in multi-port channels must be investigated.

Using R1234fy as refrigerant under low heat and mass flux conditions, Tanaka et al. (2014) reported the local heat transfer and pressure drop performances of two kinds of multi-port extruded tubes having five channels with a rectangular cross-section of 1.05 × 2.6 mm, and having 16 channels with a square cross-section of 0.9 × 0.9 mm. They concluded that under the specific working conditions, the vertical up-flow boiling heat transfer performance was significantly influenced by the channel geometries and the effect of surface tension. The tube with a square cross-section showed better heat transfer performance than that with a rectangular cross-section. These results imply that there is potential to improve performance by decreasing the cross-sectional area.

In this study, two types of rectangular aluminum multi-port extruded tubes termed tube A and tube B were manufactured, and the local heat transfer coefficient and pressure drop were measured. Tube A had 16 channels with a cross-section of 0.91 × 0.21 mm while tube B had 40 channels with a cross-section of 0.34 × 0.21 mm. The tested results were compared to Tanaka's results.

2. EXPERIMENTAL SETUP AND METHODOLOGY

2.1 Experimental Setup

Figure 1 shows a schematic of the experimental setup. The liquid refrigerant was circulated using a gear pump, and the flow rate was measured with a Coriolis flow meter. The inlet vapor quality was controlled using a constant temperature bath and an electrically heated pre-heater. The test section was set vertically and heated by DC power. The flow patterns before and after the test section were observed by two sight glasses. After leaving the test section, the vaporized refrigerant flowed to a condenser and a sub-cooler before finally returning to the gear pump.

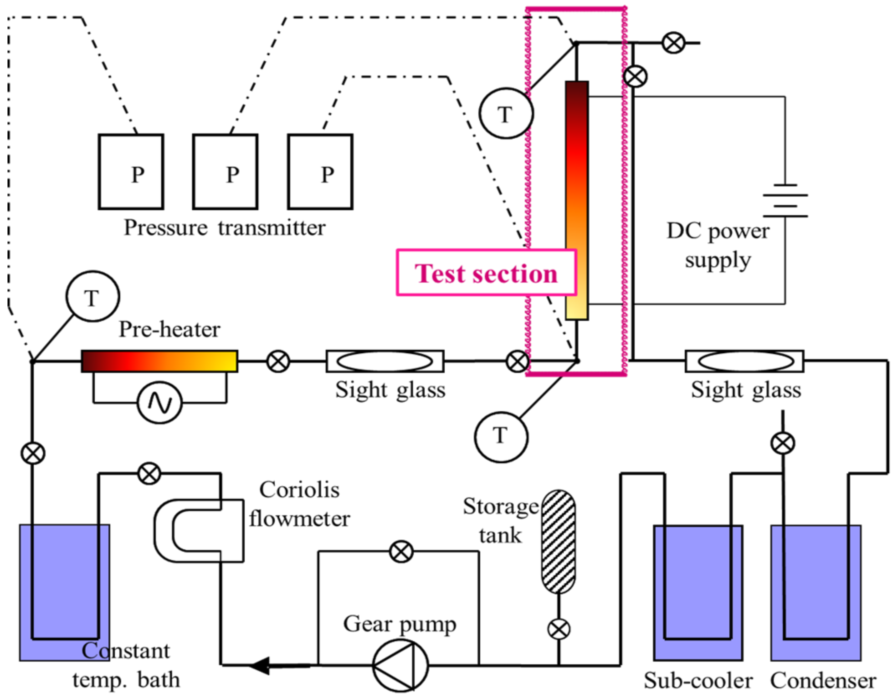

Figure 1: Schematic of experimental setup

2.2 Experimental methodology

Figure 2a shows the whole test section. There was a visualization section just before the aluminum tube, and an orifice plate with four holes of 1 mm diameter was mounted as shown in Figure 2b to enhance the mixture of vapor and liquid refrigerant. Figure 2c shows the mixture pattern of refrigerant after flowing through the orifice plate in experiments. The vapor and liquid mixed intensively, which provided a more uniform distribution of liquid for each channel. The surfaces of the electrodes were coated with an electroless nickel plating to reduce the electric resistance. To minimize heat loss, the whole test section excluding the visualization section was covered with thermal insulation material. The test tubes were made of aluminum A1050. The cross-sections of tubes A and B are shown in Figure 3, and the detailed dimensions are listed in Table 1.

The pressures and temperatures of the refrigerant at the inlet and outlet of the test tube and at the inlet of the preheater were measured by a pressure sensors and a sheathed thermocouples, respectively. As shown in Figure 4, 24 0.1 mm OD T-type thermocouples were attached to the surface of the test tube along the tube length in eight groups. All the experimental data were collected under steady working conditions. The accuracy of the thermocouples and pressure sensors were ±0.1 K and ±0.1 kPa, respectively. The accuracy of the Coriolis flow meter was ±0.1%. The heat loss was confirmed to be less than 5%. The properties of the refrigerant were determined by REFPROP version 9.1.

Table 1. Tube dimensions

Tube type	Type A	Type B
Number of channels	16	40
Tube width [mm]	16.6	16.6
Tube height [mm]	0.6	0.6
Wetted perimeter length [mm]	35.8	44
Cross-sectional area [mm^2]	3.06	2.86
Hydraulic diameter [mm]	0.34	0.26
Tube length [mm]	181, 266, 436	158, 220, 344
Heated length [mm]	85, 170, 340	62, 124, 248

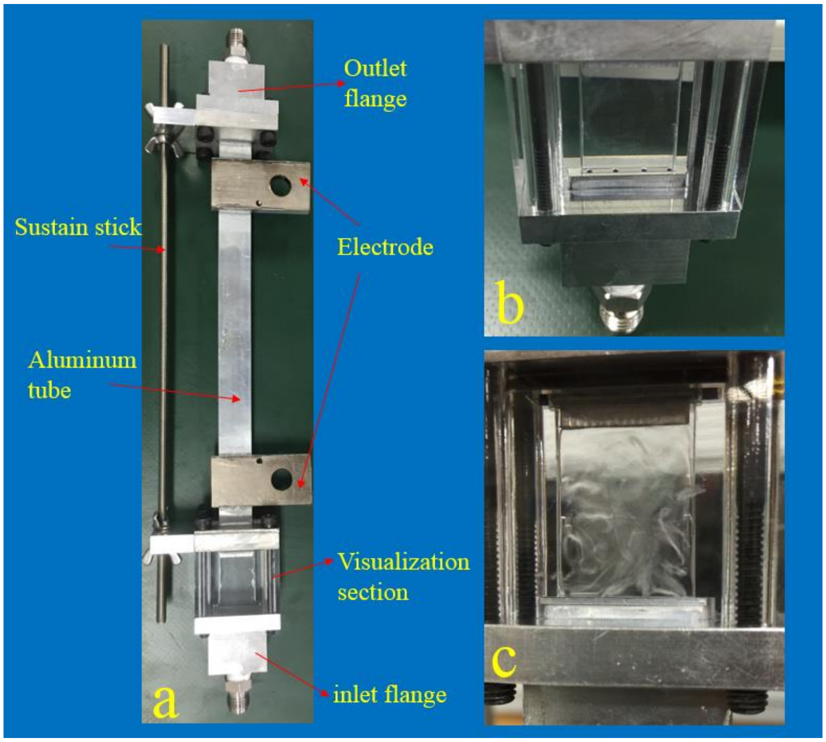

Figure 2: Test section

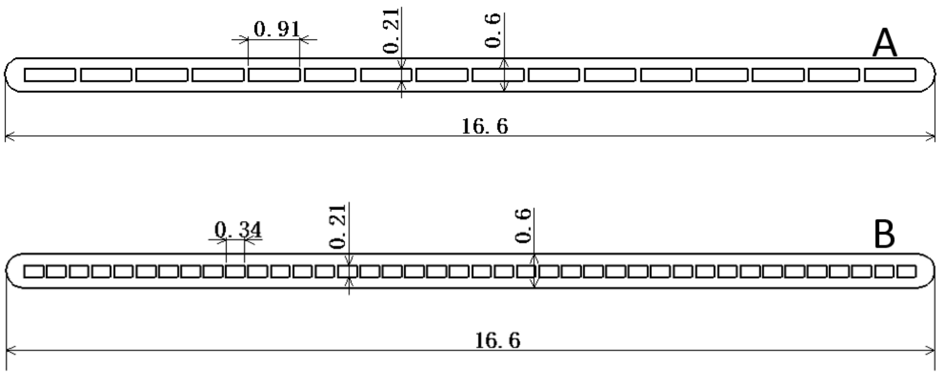

Figure 3: Tube dimensions

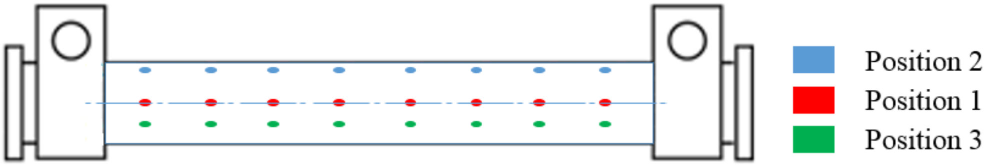

Figure 4: Thermocouple arrangement

The heat flux and mass flux ranges were 3 - 16 kW/m^2 and 60 - 240 kg/m^{2}s, respectively. The heat flux was controlled by adjusting the output voltage of the DC power supply, and the mass flux was regulated by the speed of the gear pump. The boiling temperature was set to 15 ℃ at the inlet of test section. Before flowing into the test section, the vapor and liquid were required to achieve equilibrium, which was verified through temperature and pressure measurements. If the difference between the inlet temperature and the saturated temperature calculated from the inlet pressure was less than 0.1 K, it indicated that a state of equilibrium had been achieved. Because of this, the inlet vapor quality in all experiments was approximately 0.3, which is controlled by the preheater.

The assumptions made for the purpose of data reduction are listed below:

1. The pressure inside the tube was assumed to be linear along with the flow direction, so the local pressure was calculated from the inlet and outlet pressures. The local saturated temperature was calculated according to the estimated local pressure. The pressure drops in the flanges and visualization section were neglected.
2. The inner wall heat flux was uniform at all locations, and was calculated by dividing the given heat by the heat transfer area inside the tube.
3. The inner wall temperature was assumed to be equal to that of the outer wall. This assumption could be validated by numerical simulation, showing that for the low heat flux used in the experiments, the difference between the inner wall and outer wall was less than 0.03 K.

The local heat transfer coefficient h_{ref} [kW/m^{2}K], was calculated by

$$h_{ref} = \frac{q}{T_w - T_{ref}} \tag{1}$$

where q is the inner wall heat flux, and T_w and T_{ref} are the average local wall temperature and saturated temperature determined by the local pressure, respectively. The average local wall temperature was defined as follows:

$$T_w = \frac{1}{5}(T_1 + 2T_2 + 2T_3) \tag{2}$$

where T_1, T_2, and T_3 are outer wall temperatures of positions 1, 2 and 3 shown in Figure 4. Position 1 located on the symmetric axis of the flat tube, while position 2 was set nearby the edge and position 3 located in the middle between the symmetric axis and the edge.

3. RESULTS AND DISCUSSION

3.1 Heat transfer characteristics of tubes A and B

Figures 5 and 6 show the experimental heat transfer performance of tubes A and B while the mass flux varied from 60 to 240 kg/m^{2}s and the heat flux varied from 3 to 16 kW/m^2. In order to facilitate visualization of trends, the experimental data points are connected by lines of the same color for each condition. The data of both tubes clearly show that the heat transfer coefficient decreased as the vapor quality increased under all experimental conditions. For tube B, all data show an approximately linearly decreasing trend. For tube A, the heat transfer coefficient demonstrated similar trends for mass fluxes exceeding 120 kg/m^{2}s, while the heat transfer coefficient for a mass flux of 60 kg/m^{2}s linearly decreased from 7 to 1 kW/m^2 K as the vapor quality increases from 0.3 to 0.7, and then leveled off at a low value. Similar trends were also presented by Tanaka et al. (2014) and Enoki et al. (2012). In Tanaka's research, two types of rectangular multi-port tubes with hydraulic diameters of 1.5 mm and 0.9 mm were used, and the heat transfer coefficient decreased with the increasing vapor quality for a low mass flux of 15 kW/m^2. Enoki et al. reported similar results as flow boiling occurred in a rectangular transparent tube for a low mass flux of 30 kW/m^2.

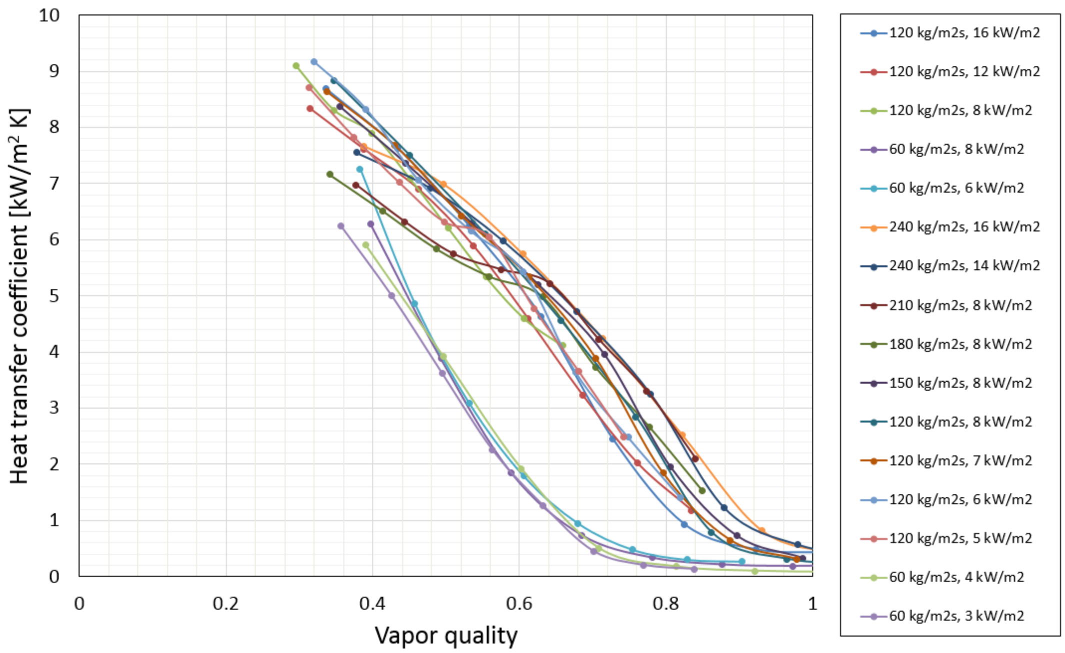

Figure 5: Heat transfer performance of tube A

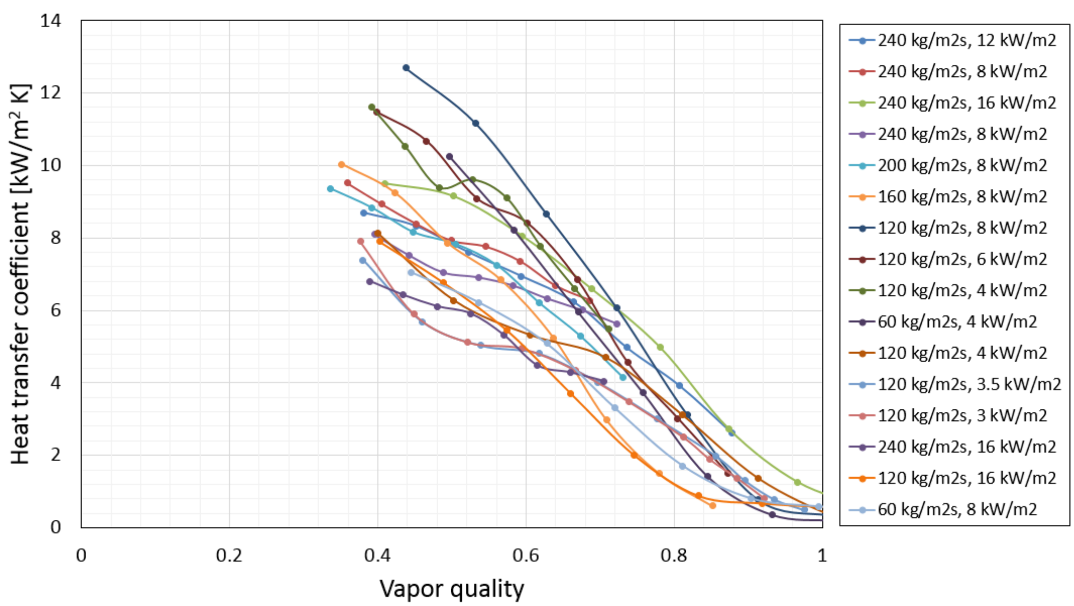

Figure 6: Heat transfer performance of tube B

It can be inferred that this trend is mainly induced by the typical flow pattern of a rectangular channel, in which most of the liquid refrigerant accumulates in the four corners based on the principle of minimum energy. As shown in Figure 7C, for relatively higher mass fluxes, thin liquid films (blue region) cover all surfaces of the rectangular channel, which provides excellent heat transfer performance (Tanaka et al., 2014). However, when the mass fluxes become lower, this liquid film shows reduced stability, which in turn induces intermittent dry patches and deteriorates the heat transfer performance. The channel shapes of tubes A and B are not square, thus the liquid accumulates in the four corners, which implies that dry patches will easily be generated on the wider surfaces, as shown in Figure 7A and 7B. As the vapor quality increases, the mass of liquid decreases, thereby making it more difficult for the liquid film to be formed on the surface. Additionally, the increase in the vapor velocity may cause turbulence and disturb the formation of liquid film. Therefore, the dry patches will occupy increasingly more area of the tube surface, resulting in increasingly worse heat transfer performance.

Generally, mass and heat flux strongly influence heat transfer performance through a complex mechanism while flow boiling occurs within a channel, a phenomenon that has been discussed in numerous research works. For tube B of the current study, although the mass and heat flux influence the heat transfer performance, their effects are substantially less dominant than that of vapor quality, thereby making it difficult to fully elucidate their mechanism of action. For tube A, when the mass flux was reduced to 60 kg/m^{2}s, a dramatic deterioration of heat transfer performance was clearly observed. The heat transfer coefficient decreased to a value of 2000 kW/m^{2}K while the vapor quality was only 0.6, and it may be speculated that dry-out occurred everywhere except in the corners when the vapor quality reached 0.7. Although this phenomenon was not observed in tube A, it may be speculated that if the mass flux continued to decrease from 60 kW/m^2, a dramatic drop in heat transfer performance would occur.

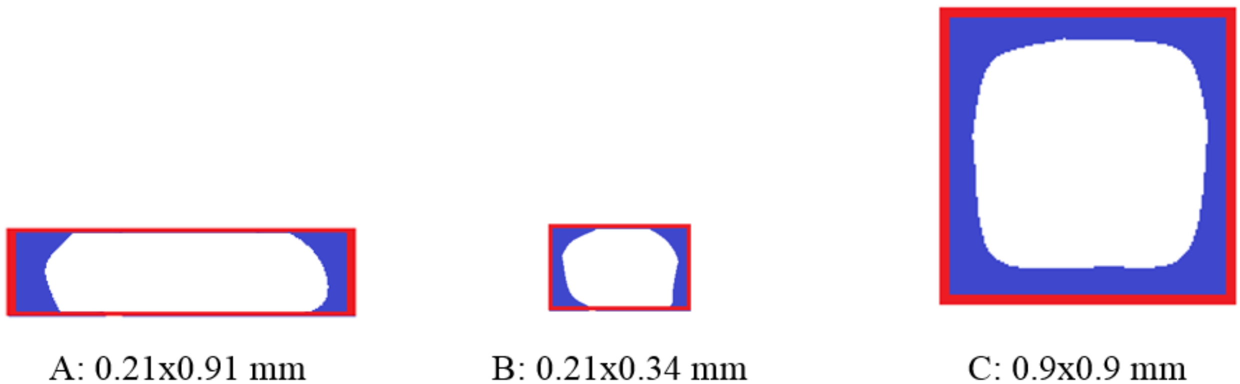

Figure 7: Flow patterns in three different tubes

3.2 Effect of cross section dimension on heat transfer

At the same heat and mass flux conditions, the heat transfer performances in tube A and tube B were compared to that of Tanaka's tube with a cross-section of 0.9×0.9 mm, which is termed tube C in this paper. Figure 8 shows the comparison. It can clearly be seen that at the relatively lower mass flux of 60 kg/m^{2}s, tube C demonstrates the best performance over the entire measured vapor quality range, while tube A and B demonstrate a substantially worse performance due to their structural limitations discussed above. When the mass flux increased to 120 kg/m^{2}s, the dry patch was alleviated in tubes A and B. In the low vapor quality region, because of the smaller channel size, a thinner film was likely established in tubes A and B, which led to superior heat transfer performance over that of tube C. However, as the vapor quality increased, the dry patch occupied more of the tube surface, thereby causing the heat transfer performance to drop dramatically. Figures 5 and 6 show that for most cases, the heat transfer performance of tube B was better than that of tube A.

3.3 Pressure drop of tubes A and B

The two phase pressure drop within the vertical tube can be expressed as the sum of frictional, gravitational, acceleration, inlet contraction, and outlet extraction components. The pressure drop in the visualization section is quite small compared to that within the tubes, so it may be neglected. The frictional pressure drop plays a dominant role amongst the five components; therefore, numerous researchers focus on it and many correlations are found within the literature. In this study, three correlations from Lee and Lee (2001), Lockhart and Martinelli (1949), and Mishima and Hibiki (1996) were adopted for comparison with the experimental results of tubes A and B. The total pressure drop calculation method was the same as Anwar's (2015). Fifteen data points of tube A and 21 data points of tube B were compared with the calculation results, as shown in Figure 9. It can be seen that the historical Lockhart and Martinelli correlation shows the best accuracy of the three correlations, in which 72.2% of the data were located in the $\pm$30% range.

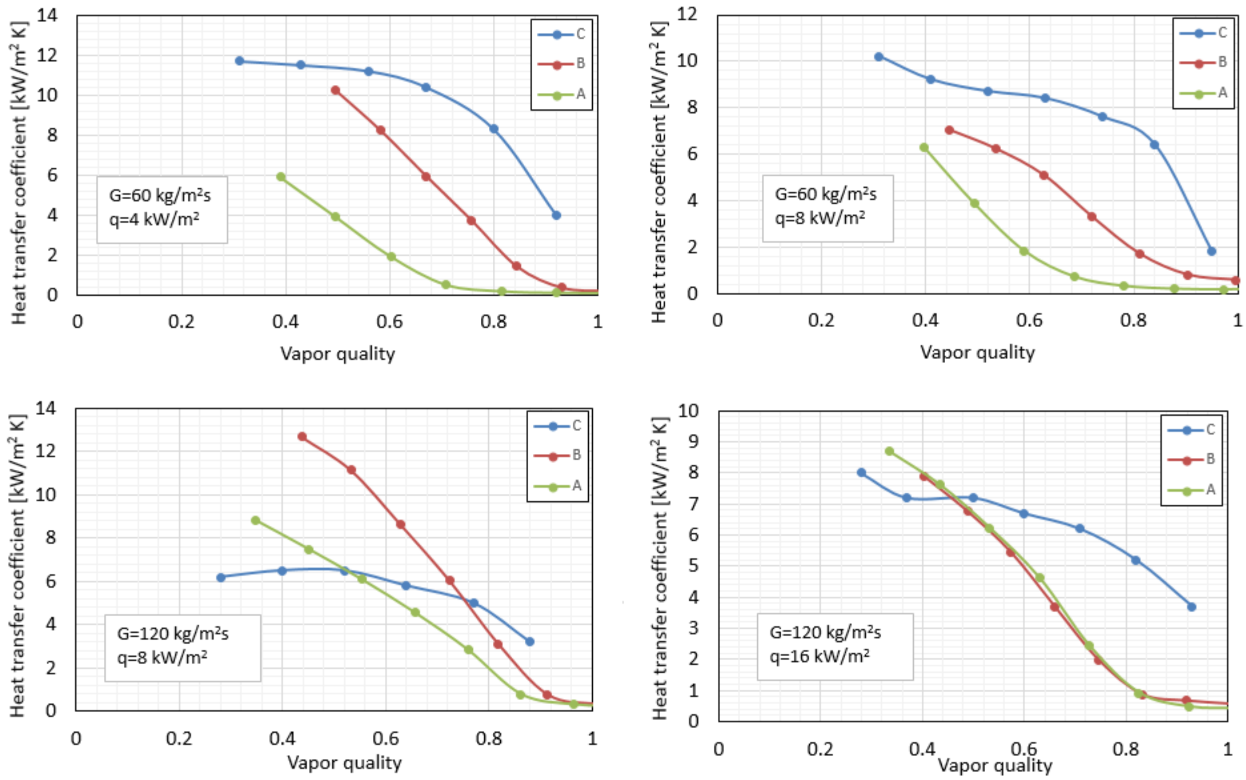

Figure 8: Comparisons of heat transfer performance in tubes A, B and C

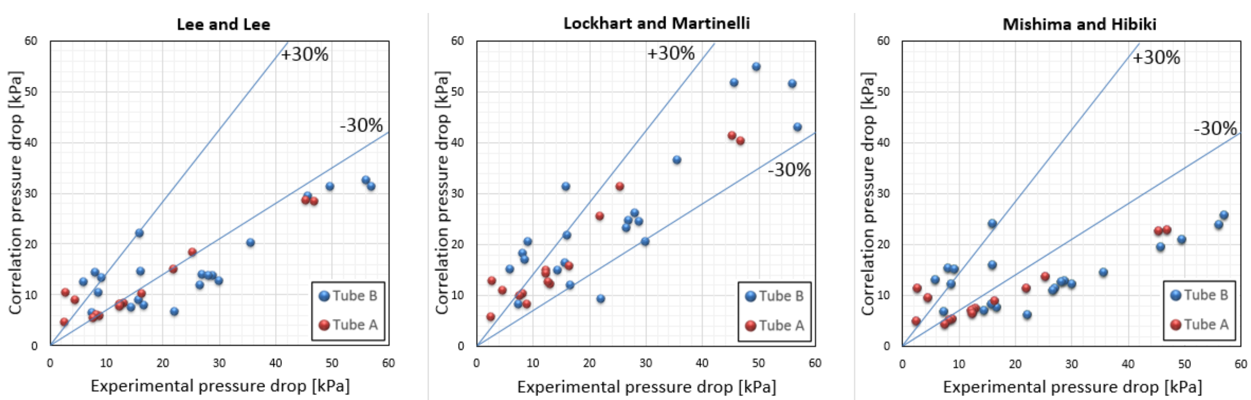

Figure 9: Comparison between experimental and correlation results of pressure drop

4. CONCLUSIONS

The characteristics of local heat transfer and pressure drop were investigated experimentally for the vertical up-flow boiling of refrigerant R1234yf in two types of aluminum multi-port extruded tubes having 16 channels with a cross-section of 0.91 × 0.21 mm (tube A) and 40 channels with a cross-section of 0.34 × 0.21 mm (tube B). The heat transfer performance was compared with that of multi-port extruded tubes having 16 channels with a cross-section of 0.9 × 0.9 mm (tube C), and the pressure drop was compared with calculation results of three different correlations. From the results, the following conclusions were drawn.

(1) Under low heat and mass flux conditions, the heat transfer coefficient almost linearly decreased as a function of vapor quality in tubes A and B. As the vapor quality increased, the area of dry patches became lager within these rectangular channels, which deteriorated the heat transfer performance dramatically.

(2) The channel dimensions significantly influenced the heat transfer performance. At a mass flux of 60 kg/m^{2}s, tube C showed better heat transfer performance than tubes A and B for all measured vapor quality regions because the shape of tubes A and B is not square, so the dry-out occurs more easily on the wider surfaces, thereby compromising the heat transfer performance. When the mass flux increased to 120 kg/m^{2}s, the dry-out was alleviated, enabling better performance in low vapor quality regions. In high vapor quality regions, the dry patches reoccupied the heat transfer area, leading to reduced heat transfer performance in tubes A and B. Moreover, tube A was worse than tube B because of its lower aspect ratio.

(3) The correlations of Lee and Lee (2001), Lockhart and Martinelli (1949), and Mishima and Hibiki (1996) did not precisely predict the experimental pressure drops of tube A and B. Relatively, the Lockhart and Martinelli correlation showed the best prediction.

NOMENCLATURE

h	heat transfer coefficient	(kW/m^{2}K)
q	heat flux	(kW/m^2)
T	temperature	(K)

Subscript

ref	refrigerant
w	outer wall of flat tubes
1	Position located on the symmetric axis of the flat tube
2	Position located nearby the edge of the flat tube
3	position located in the middle between the symmetric axis and the edge

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