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Effect of Flow Direction of Heating Medium on Boiling Heat Transfer Characteristics in a Single-Channel Plate-Fin Heat Exchanger

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ABSTRACT

This study deals with evaporative heat transfer characteristics of vertically upward flows in single-channel plate-fin heat exchangers. R-134a was used as the refrigerant. The refrigerant channel was sandwiched by two water channels with the same channel shape. The refrigerant was heated by hot water from the both side. In this experiment, pressure of the working fluid was maintained at 0.665MPa (saturation temperature 25 °C). Subcooled liquid with the subcooling of 5 and 10 K, or wet vapor with the quality of 0.1 was supplied to evaporator. Two kind of offset fin, Fin A and Fin B, whose fin pitch were 1.95 and 3.175 mm, respectively, were used. The channel hydraulic diameter with Fin A and Fin B were 1.78 and 2.89 mm, respectively. Effects of the fin pitch and flow direction of the heating water, such as counter and parallel flow arrangements, on the heat transfer rate, pressure drop, and wall temperature distribution on the outside of the water channel were examined. The wall temperature was visualized by an IR camera. As the results, it was shown that the effects of the water and the fin pitch on the heat transfer rate were a little due to a dryout in the refrigerant flows. However, a large difference in the wall temperature distribution between the counter and parallel flow was observed. The difference might be caused by the phase distribution of refrigerant, and lead to the increase in the pressure drop for the parallel flow. For the parallel flow arrangement, the temperature on the right side with the water inlet port was quite higher than the right side. The temperature distribution was kept over the heat exchanger. The tendency was weaker for Fin B with the larger fin pitch due to phase separation caused by gravity.

1. INTRODUCTION

Technological advance on heat exchangers is required for effective use of thermal energy. Plate-fin heat exchangers can contribute to size reduction of heat exchangers and reduction in the temperatures difference between fluids because of the quite large heat transfer area per volume and small surface area to the surroundings. Small surface area leads to small heat loss. The other advantage is on the high flexibility in channel design. Channel thickness and fin configuration can be designed for each fluid independently. On the other hand, since wide and thin channels are formed between plates and fins are inserted in each channel, pressure loss for a mass flux becomes larger. Therefore plate-fin heat exchangers have many parallel channels. When a heat exchanger is used as evaporator or condenser, since the working fluid is fluid flowing as gas-liquid two-phase flow, the understanding of flow characteristics in such wide and thin channels is important. Because heat transfer and pressure loss characteristics should be strongly affected by the flow behaviors, especially phase distributions.

Previously, Arima, et al. [1], Okamoto, et al. [2] investigated boiling flow characteristics of ammonia or ammonia/water mixture in plate type evaporators without fin insertion used for an ocean thermal conversion system. Baba, et al. [3] visualized boiling flow behaviors in a brazing plate heat exchanger by neutron radiography, and

reported that downward refrigerant flow produced higher heat transfer performance than upward flow due to the difference in phase distribution. Kushibe, et al. [4] examined three kinds of plate heat exchanger with different groove on the heat transfer surface. The heat exchangers were used as evaporator using ammonia as the refrigerant. Correlating equations for boiling heat transfer coefficient in refrigerant flow and pressure loss and heat transfer coefficient in water flow were proposed. Asano, et al. [5] reported the effect of water flow direction in a vertical plate fin heat exchanger with refrigerant upward flow on heat transfer performance. They concluded that the parallel flow arrangement produced higher heat transfer performance than the counter flow arrangement.

In an evaporator, since refrigerant temperature can be estimated to be constant at the saturation temperature, the logarithmic mean temperature difference becomes the same independent of the water flow direction, parallel flow or counter flow. However, the local temperature difference should be different depending on the water flow direction. At the inlet section of refrigerant, the temperature difference for the parallel flow will be larger than the value for the counter flow. The larger temperature difference is expected to enhance boiling of subcooling liquid or low quality wet vapor. The difference in boiling behavior at the inlet section may lead to a difference in phase distribution in heat exchanger. In this study, the effect of flow direction of heating medium on heat transfer performance and flow characteristics was experimentally examined for a single-path plate-fin evaporator.

2. EXPERIMENTAL APPARATUS AND METHOD

Schematic diagram of experimental setup is shown in Fig. 1. A hydrofluorocarbon R134a was used as the refrigerant. The experimental loop was composed of a gear pump, a pre-heater, the test section, a sub-cooler, and a condenser. An accumulator was connected to a branch at the upstream of the pump in order to stabilize the system pressure. The system was maintained by the temperature of the accumulator. The refrigerant was circulated by the gear pump, and was supplied to the test section after setting the inlet condition by a heat input at the preheater. The refrigerant flow vertically upward, and hot water flow upward or downward to form a parallel or counter flow heat exchange. The plate fin heat exchanger is formed by sandwiching a single refrigerant path between two water paths.

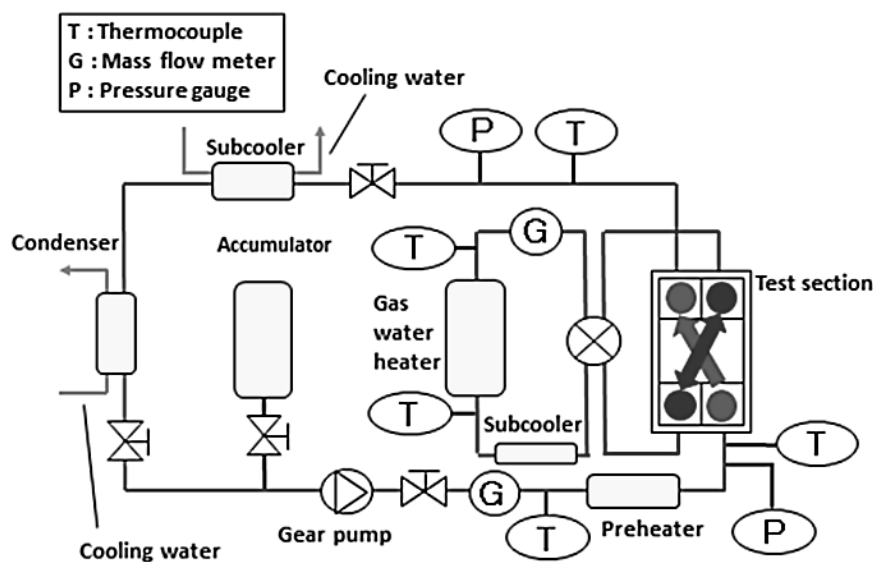


Fig. 1 Schematic diagram of experimental apparatus.

The configuration of the tested plate-fin heat exchanger is shown in Fig. 2. The heat exchanger was made of aluminum. Aluminum offset fins with the same configuration were installed into each channel. At the inlet and exit section, a clearance between the fin and the bottom or top wall was set to enhance the lateral flow distribution. In the case that there was no clearance at the inlet section, liquid stagnation was observed in our previous air-water two-phase flow experiments. It was confirmed in the experiments that the clearance solved the problem on the liquid stagnation. The same offset fins were installed in the refrigerant and water paths. Two types of the test section as

shown in Fig. 3 and Table 1 were used. The average hydraulic diameter defined from the cross-sectional area and wetted perimeter of the refrigerant path were 1.75 mm and 2.82 mm. Hot water flew upward or downward to form a parallel or counter flow heat exchange, respectively.

In order to estimate the distribution of heat transfer rate, the outside wall temperature of the water path was visualized by an IR camera (NIPPON AVIONICS Co., Ltd., Model number: R300S-R02, Measuring range: -40 to 500 °C, Sensitivity: 0.03 °C, Accuracy: ±1 °C). On the measured surface, thermal insulator was removed, and black paint with the emissivity of 0.94 was painted. In a preliminary experiment, radiation heat was evaluated under the condition where the refrigerant path was evacuated. It was confirmed that there was no significant decrease in the water temperature through the heat exchanger. Since the heat diffusion in the water flow was quite high due to the fin insertion, it could be said that the wall temperature showed water temperature.

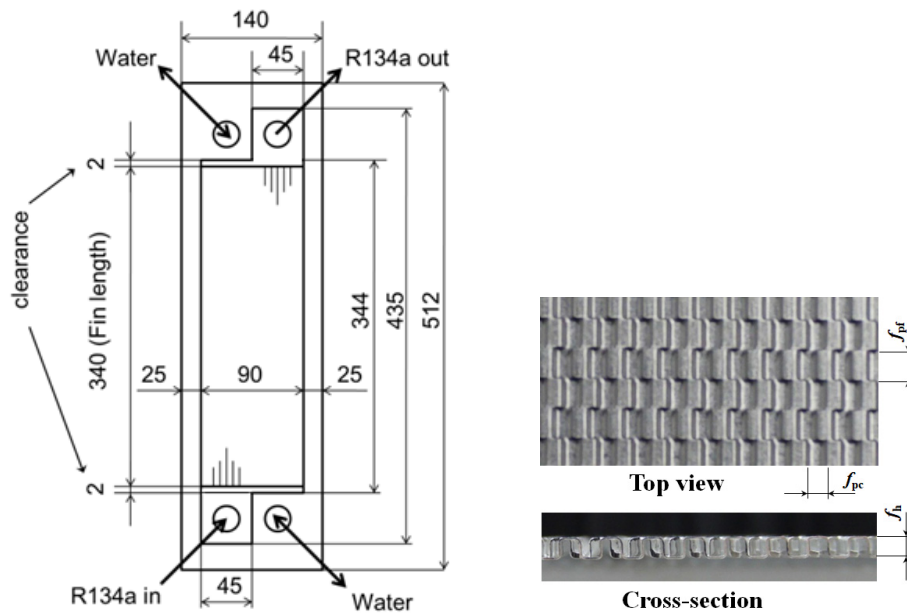


Fig. 2 Plate-fin evaporator. Fig. 3 Offset fin.

Table 1 Specifications of offset fins and channels.

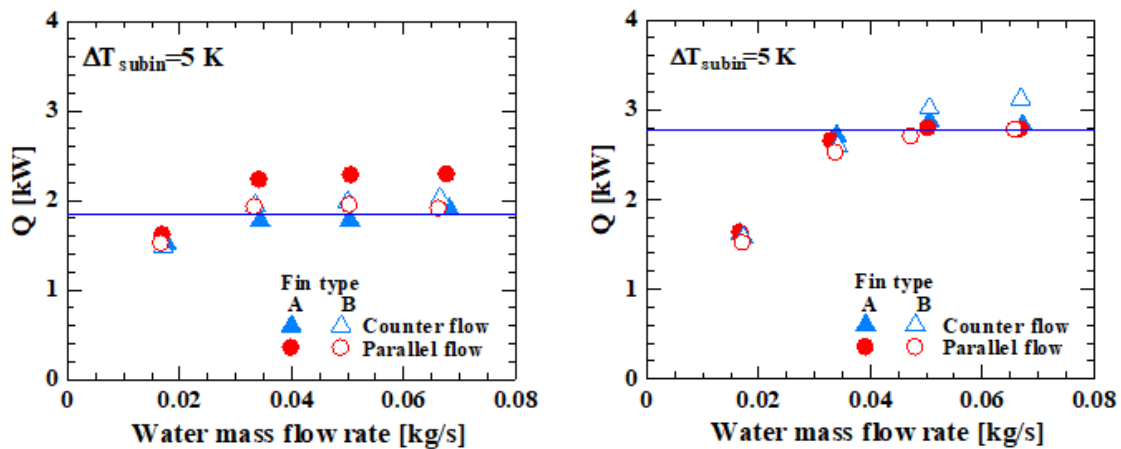
		Fin A	Fin B
Offset fin	Material	Aluminum	Aluminum
	Thickness	0.2 mm	0.3 mm
	Height f_h	2.0 mm	3.2 mm
	Number of fins	13 fins/inch	8 fins/inch
	Fin pitch		
	in cross-section f_{pc}	1.95 mm	3.175 mm
	in flow direction f_{pf}	3 mm	6 mm
Channel	Cross-sectional area	145.4 mm ²	236.3 mm ²
	Wetted perimeter	327.2 mm	327.4 mm
	Hydraulic diameter	1.778 mm	2.887 mm

The experiments were carried out at the constant refrigerant pressure at 0.665 MPa (saturation temperature 25 °C) and for three inlet conditions of inlet subcooling liquid with the subcooling degree $\Delta T_{in} = 5, 10$ K and inlet wet vapor with the quality $x_{in} = 0.1$, and for three mass fluxes $Gr = 70, 105, 140$ kg/(m²s). Water mass flow rate was varied in the range of 0.017 to 0.067 kg/s with the constant inlet temperature at 50 °C.

3. EXPERIMENTAL RESULTS AND DISCUSSION

The measured results of the heat transfer rate are plotted against the water flow rate and is shown in Fig. 4. The heat transfer rate was measured from the enthalpy difference of water. Figure (a) and (b) show the results for the inlet subcooling degree T_{subin} of 5 K and the refrigerant mass flow rate m_r of 0.010 kg/s and 0.015 kg/s, respectively. The results for Fin A and Fin B are plotted by bold and open symbols, respectively. The results for the counter and parallel flow arrangements are plotted by blue triangle and red circle symbols, respectively. The horizontal solid line indicates the required heat transfer rate to generate saturated vapor.

The heat transfer rate, Q , increased with increasing in water flow rate and seemed to become saturated for each case. Since the temperature difference between water and refrigerant was enough large at the outlet of the water, the saturation in the heat transfer rate should be caused not by the increase in temperature effectiveness but by a deterioration in the heat transfer coefficient. The heat transfer coefficient of water flow should be enhanced with the increase in the mass flow rate. And, under the condition with the water flow rate larger than 0.03, the heat transfer rate reached the value to generate saturated vapor. Therefore, it could be said that the deterioration in heat transfer was caused by dryout in the refrigerant flow. The effect of the fin pitch and water flow direction was a little. For the lowest water flow rate of 0.017 kg/s, there were little difference in the heat transfer rate. In this case, the outlet temperature of water became close to the refrigerant temperature. The cause of the little difference in the heat transfer rate was high temperature effectiveness.



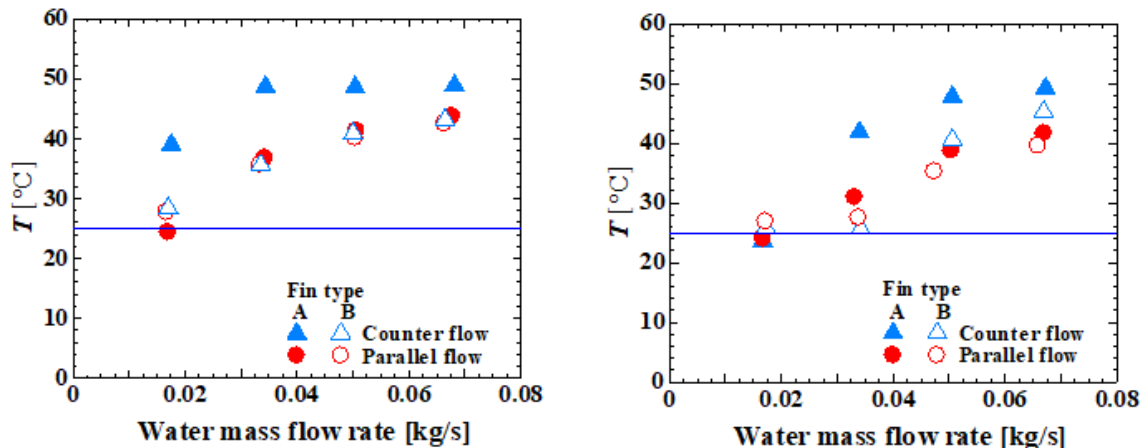
(a) Refrigerant mass flow rate : 0.010 kg/s
(mass flux : Fin A 70 kg/(m²s), Fin B 43 kg/(m²s))

(b) Refrigerant mass flow rate : 0.015 kg/s
(mass flux : Fin A 105 kg/(m²s), Fin B 64 kg/(m²s))

Fig. 4 Heat transfer rate measured from the enthalpy difference of the heating water between the inlet and outlet of the test section. The horizontal line shows the required heat to evaporate all of the liquid.

The outlet temperatures of refrigerant, $T_{r,out}$, are shown in Fig. 5 (a) and (b). The plot symbols are the same with those in Fig. 4. The horizontal line shows the saturation temperature according to the inlet refrigerant pressure. In both cases, the temperature rose with the increase in the water flow rate, and even for the lowest water flow rate, the temperature became superheated values in some cases. For the counter flow in Fin A with the minimum water flow rate, although the heat transfer rate was lower than the value to generate the saturated vapor, the outlet temperature of refrigerant showed superheated vapor. The temperature was the measured value, so the refrigerant might be under the thermal non-equilibrium condition at the measure section. The inner wall of the refrigerant channel around the outlet should be dryout. In the comparison between counter and parallel flows, the temperatures for the counter flow

became higher due to the larger temperature difference at the refrigerant outlet. While the effect of fin shape was little for the lower refrigerant mass flow rate in Fig. (a), the effect of the fin shape was observed for the counter flow with the higher refrigerant mass flow rate in Fig. (b). The temperature for Fin A with the smaller channel diameter became higher. The cause might be higher inner wall temperature of the refrigerant channel due to the higher heat transfer coefficient of water flow.



(a) Refrigerant mass flow rate : 0.010 kg/s
(mass flux : Fin A 70 kg/(m²s), Fin B 43 kg/(m²s))

(b) Refrigerant mass flow rate : 0.015 kg/s
(mass flux : Fin A 105 kg/(m²s), Fin B 64 kg/(m²s))

Fig. 5 Refrigerant temperature at the outlet of the test section.

The temperature distributions on the external surface of the water channel measured by the IR camera for Fin A and Fin B are shown in Fig. 6 (a) (b) and Fig. 7 (a) (b), respectively. The temperature distributions are shown by the color scale of 20 to 55 °C. The refrigerant was supplied as subcooled liquid with the subcooling degree, ΔT_{subin} , of 5 K, and the hot water flow rate, m_w , was 0.033 kg/s. The flow rate was the second lowest condition in Figs. 4 and 5. Figure (i) and (ii) show the results for the different refrigerant flow rate of 0.010 and 0.015 kg/s, respectively. In each figure, the left figure shows the results for the counter flow, and the right figure shows the results for the parallel flow. The inlet and outlet ports for water and refrigerant are indicated by red and blue arrows, respectively. As described above, the temperature distribution on the external surface of water channel could be treated as water temperature, because the outside wall could be approximated to be adiabatic. Therefore, the area with large temperature change would be considered as the place with high heat flux.

For the counter flow arrangement of Fin A, a red area spreading from the water inlet port was observed. In this area, heat flux should be quite low due to dryout in the refrigerant flow. The area became narrower with increasing the mass flow rate of refrigerant due to decreasing the dryout area. While the red area was horizontally symmetrical for the lower mass flow rate of refrigerant, the area was observed only on the left side for the higher mass flow rate of refrigerant. The reason might be not on the water side but on the refrigerant side. The vapor quality might be higher on the left side with the inlet port of refrigerant, therefore, dryout might easily occur on the left side due to high heat transfer coefficient of refrigerant vaporization. In the comparison between Fin A and Fin B, the red area for Fin B was narrower than the case for Fin A. It could be said that heat flux in the lower part of the heat exchanger was higher for Fin A than Fin B.

On the other hand, for the parallel flow arrangement, the temperature distribution was quite different. Since the incoming water was cooled by the boiling heat transfer of the refrigerant, no red area was observed near the water inlet. For Fin A, there was a large temperature difference in the horizontal direction. Temperature on the right side was higher than the left side. The distribution was seemed to be kept along the flow direction. Even for the parallel flow arrangement, hot water at around 40 °C reached to the outlet port of the refrigerant, therefore superheat heat temperature was observed. The tendency became weaker for Fin B, especially the higher mass flow rate in Fig. (ii). It might be that the heat transfer characteristics might be dominated by the heat transfer in water flow for Fin B.

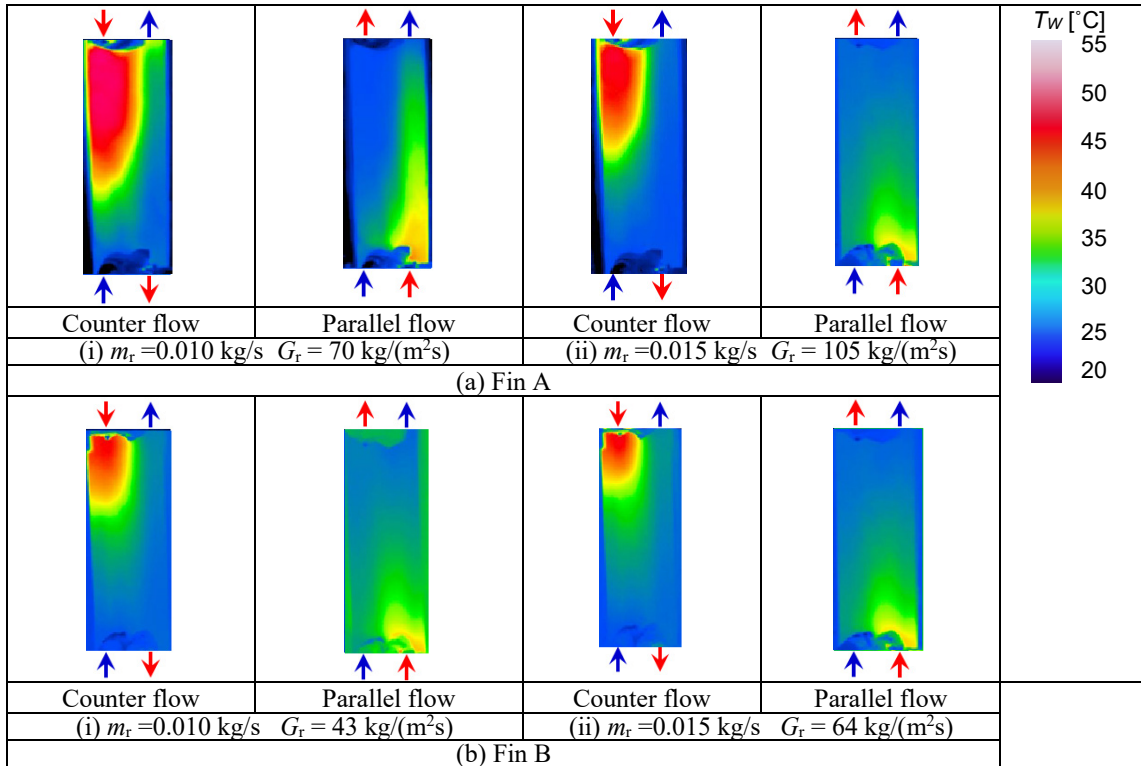


Fig. 6 Wall temperature profile on the outside of water path ($m_w = 0.017$ kg/s, $\Delta T_{\text{subin}} = 5$ K).

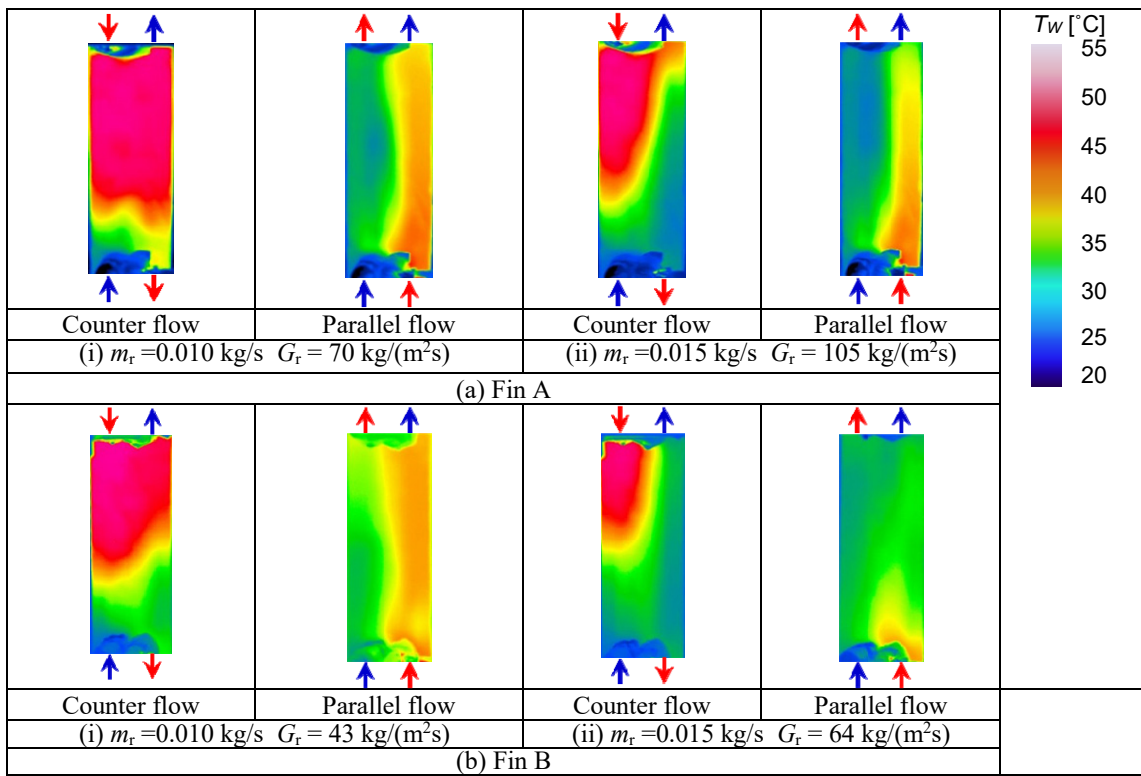
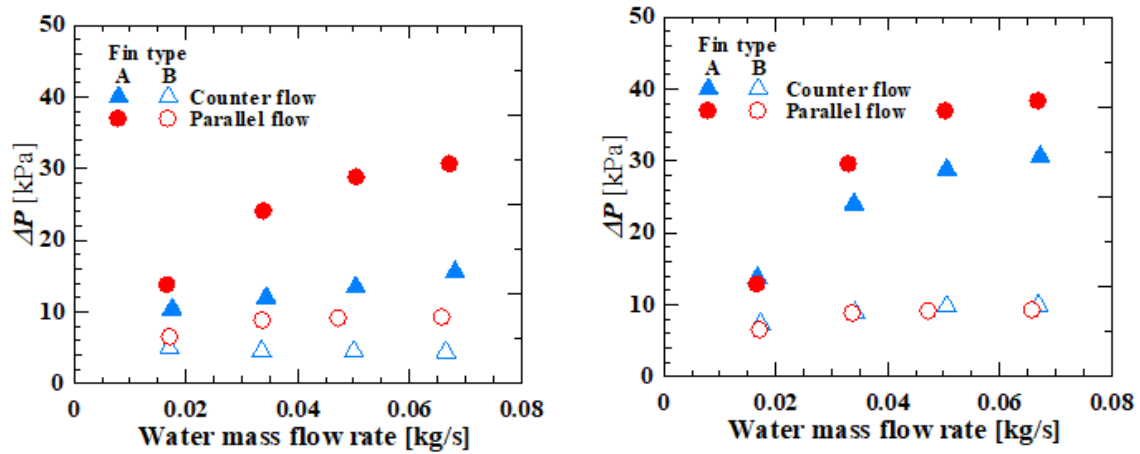


Fig. 7 Wall temperature profile on the outside of water path ($m_w = 0.033$ kg/s, $\Delta T_{\text{subin}} = 5$ K).

The measured pressure loss are shown in Fig. 8 (a), (b). The pressure loss was measured as the difference between the inlet and outlet pressure measured by the pressure transducers. The experimental conditions and plotted symbols are the same with those in Figs. 4 and 5. It was clearly shown that the pressure losses for Fin A were quite higher than those for Fin B. For Fin A, the values of the parallel flow arrangement were higher than those for the counter flow arrangement. The difference might be caused by the maldistribution of refrigerant as described on Fig. 6. The refrigerant might flow as a high quality wet vapor flow on the left side. On the other hand, for Fin B, such maldistribution might be reduced by a strong heating at around the water inlet on the lower right side.



(a) Refrigerant mass flow rate : 0.010 kg/s
(mass flux : Fin A 70 kg/(m²s), Fin B 43 kg/(m²s))

(b) Refrigerant mass flow rate : 0.015 kg/s
(mass flux : Fin A 105 kg/(m²s), Fin B 64 kg/(m²s))

Fig. 8 Pressure drop of refrigerant flow in the test section.

The heat transfer quantity Q shown in Fig.4 can be divided into three parts according to the refrigerant flow condition, such as the heat transfer quantity Q_{liq} in the liquid phase region, Q_{two} in the two-phase flow region, and Q_{gas} in the vapor flow region as shown in Eq. (1).

$$Q = Q_{liq} + Q_{two} + Q_{gas} \quad (1)$$

It is assumed that Q_{liq} can be calculated by multiplying the enthalpy difference between the subcooled and saturated liquid by the refrigerant mass flow rate. Similarly, it is assumed that Q_{gas} can also be calculated from heat exchanger outlet condition. As the result, Q_{two} is calculated from the equation (1). The heat transfer coefficient K_{two} in the two-phase flow region is calculated by the Eq. (2) using the logarithmic average temperature difference.

$$Q_{two} = K_{two} \cdot LMTD_{two} \cdot A_{two} \quad (2)$$

where A is the heat transfer area of the two-phase flow region, and is calculated based on the proportion of the heat transfer rate. The calculated results of K_{two} is summarized in Table 2. Under every flow condition in Table 2, the value of K_{two} increased with increasing the water mass flow rate. It is known that the evaporative heat transfer coefficient increases with increasing vapor quality by liquid film vaporization near dryout. The increase in the values of K_{two} are considered to be due to the film vaporization.

Table 2 Heat transfer coefficient, K_{two} , in the two-phase flow region under each flow condition.
 ((i) $m_r = 0.010$ kg/s, (ii) $m_r = 0.015$ kg/s)

			K_{two} [kW/(m ² ·K)]			
			$m_w=0.017$ kg/s	0.033 kg/s	0.050 kg/s	0.067 kg/s
FinA	(i)	Counter	7.29	10.58	12.68	15.05
		Parallel	7.18	13.11	15.56	17.73
	(ii)	Counter	7.22	12.36	17.04	20.05
		Parallel	7.15	13.69	18.68	22.39
FinB	(i)	Counter	6.96	11.21	14.22	16.23
		Parallel	7.16	11.60	17.12	20.02
	(ii)	Counter	6.94	13.02	14.70	16.40
		Parallel	6.98	13.21	16.62	19.91

4. CONCLUSIONS

Boiling heat transfer characteristics of vertically upward flows in single channel plate-fin heat exchangers with offset fins were examined using R-134a. The effect of the water flow direction on the heat transfer rate was a little, because the heat transfer rate became saturated by dryout in the refrigerant flow or the increase in the temperature effectiveness. From the measured external surface temperature, a large temperature difference in the horizontal direction caused by maldistribution of the refrigerant flow was observed. The fact might lead to larger pressure loss of the refrigerant for Fin A with smaller hydraulic diameter. As for the heat transfer coefficient in the two-phase flow region, the heat transfer coefficient increased with increasing the water flow rate.

NOMENCLATURE

A	Heat transfer area	(m ²)
G_r	Refrigerant mass flux	(kg·m ⁻² ·s ⁻¹)
K	Heat transfer rate	(W·m ⁻² ·s ⁻¹)
m_r	Mass flow rate of refrigerant	(kg·s ⁻¹)
m_w	Mass flow rate of water	(kg·s ⁻¹)
ΔP	Pressure loss	(kPa)
Q	Heat transfer quantity	(kW)
T_r	Refrigerant temperature	(°C)
T_w	Water temperature	(°C)
ΔT_{sub}	Degree of subcooling	(K)
x	Quality	(-)
ε	Emissivity	(-)

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