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Performance comparison of microchannel evaporators with refrigerant R-22

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Abstract

An experimental study on the performance comparison of microchannel evaporators with refrigerant R-22 was conducted. Six microchannel evaporators were designed and manufactured for a residential air-conditioner. They were tested with psychrometric calorimeter test facilities. The experiment was performed with both vapor compression system and refrigerant circulation system. Each evaporator was made up of two parallel flow heat exchangers connected with several return pipes. The parallel flow heat exchanger had 41 microchannel tubes inserted between inlet and outlet headers. The microchannel tube had 8 rectangular ports with the hydraulic diameter of 1.3 mm. For the vapor compression system, the flow area ratio and the number of return pipes had a great effect on the cooling capacity. Type 3 with a flow area ratio of 73% and 58% showed the best cooling capacity. It had 12 return pipes and 3 circuits. There is a merging manifold in it. The effect of the number of circuits and merging manifold on the cooling capacity was relatively small. For the refrigerant circulation system, the effect of the mass flow rate on the cooling capacity was slightly superior to that of inlet quality. The effect of the number of circuits on the cooling capacity was different from the result of the vapor compression system. The effect of merging manifold was negligible, which was consistent with the result of the vapor compression system. The cooling capacity proportionally increased as the vertical inclination angle of the evaporator increased due to gravity force.

Keywords: Microchannel; Microchannel evaporator; Cooling capacity; Refrigerant

1. Introduction

Some challenging researches on the application of microchannel heat exchangers for the residential air-conditioner have been conducted, with the increase in the use of microchannel heat exchangers because of their small size, weight and refrigerant charge. Microchannel condensers have been developed for the residential air-conditioner and on the market. Microchannel evaporators, however, are not to be utilized for the residential air-conditioner. There are two major problems for their application. The first one is condensate retention inside louvered fins. The retention of condensate prohibits air from passing through the evaporator, and then severely deteriorates cooling capacity. The second one is flow maldistribution in microchannel tubes. Part of the evaporator is filled with superheated vapor and heat transfer performance is very much decreased because of flow maldistribution in microchannel tubes.

Although condensate retention and flow maldistribution problems are very essential compared with the microchannel condenser application, few studies have been done for the microchannel evaporator application. According to a survey of the open literature, most studies were carried out for the flow maldistribution in microchannel tubes while the

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research on the condensate retention inside louvered fins was few. This is because it is very expensive to make a new louvered fin for the residential air-conditioner.

Many studies on the flow maldistribution in the area of heat exchangers were conducted under the simple geometry like T-junction [1-6] or multi-branch [7-8] with round tubes. For adiabatic conditions with microchannel tubes, Tompkins et al. [9] studied flow distribution and pressure drop in microchannel manifolds. Kim et al. [10] and Lee and Lee [11] investigated the effect of the intrusion depth of the tube into a header on the flow mal-distribution characteristics. Kim and Sin [12] presented the two-phase flow distribution of air-water annular flow for a round header to 30 microchannel tubes. They reported that the tube intrusion depth had a great effect on the water flow distribution under downward flow configuration, while it did not alter the flow distribution under upward flow configuration. On the other hand, Cho and Cho [13] experimentally investigated flow maldistribution and phase separation characteristics in microchannel tubes using the refrigerant of R-22. Koyama et al. [14] represented some experimental results for two-phase flow distribution in horizontal headers with downward 6 minichannel-branches using R-134a. They reported that the liquid was distributed easily at the first branch and the vapor flowed to the downstream of the header.

For non-adiabatic conditions, Kim and Groll [15] investigated the performance of a unitary split system using microchannel heat exchangers instead of the conventional fin-tube designs as the outdoor coil for air conditioning and heat pump applications. They reported that the highest EER and cooling capacity was achieved with the system that used the microchannel heat exchangers with 20FPI and a 15° windward inclination of the heat exchanger. Cho and Cho [16] investigated the two-phase flow distribution and pressure drop in microchannel tubes under non-heating and heating conditions using the refrigerant of R-22. Kim et al. [17] investigated the effect of inclination on the air-side performance of a brazed aluminum heat exchanger under dry and wet conditions. They showed that heat transfer performance for both dry and wet conditions was not influenced significantly by the inclination angle. Oh et al. [18] experimentally studied the effect of orientation of microchannel tubes, hydraulic coating and louver pitch on the heat transfer performance of the microchannel evaporator. They reported that the orientation of a microchannel tube has a great effect on the heat transfer performance.

The microchannel heat exchanger is one of the potential alternatives for the conventional fin-tube heat exchanger. Both condensate retention and flow maldistribution are very important compared with the microchannel condenser application. Few studies, however, have been conducted for the residential air-conditioner as an evaporator. Therefore, the aim of the present study was to develop several microchannel evaporators for the residential air-conditioner and evaluate them under wet condition and the effects of some design parameters on the performance of microchannel evaporators were presented as well.

2. Experimental apparatus

2.1 Experimental setup

The performance evaluation of prototype microchannel evaporators was conducted in two different ways: First is the system that replaced the fin-tube evaporator with the prototype microchannel evaporator for the refrigerant vapor compression system(VCS), so it is called drop-in test. The drop-in test is generally applied to evaluate the performance of newly developed heat exchangers. So, it was conducted to provide real experimental data for the residential air-conditioner. Second is the refrigerant circulation system(RCS) to investigate the effect of inlet mass flow rate, inlet quality and inclination angle on the cooling capacity. It is very difficult to change the dynamic parameters, like inlet mass flow rate and inlet quality under the VCS, because the VCS has a compressor with the nearly fixed capacity.

Fig. 1 shows a schematic diagram of the experimental setup for the VCS and the RCS. The key specification of the VCS is summarized in Table 1. The VCS had the same specification with one of residential air-conditioners except for the evaporator. An expansion valve and a series of needle valves were installed to control the degree of the superheat at the outlet of the evaporator. The RCS consisted of a refrigerant pump, a mass flowmeter, a pre-heater, a test section and a condenser. Mass flow rate was controlled at the inlet of the test section with a refrigerant pump. And refrigerant inlet quality of the test section was kept constant by providing a pre-heater with required heat to bring subcooled liquid at the inlet of the pre-heater to saturated liquid with the inlet quality. Required heat, therefore, was calculated by Eq. (1) from the energy balance of refrigerant between inlet and outlet of the pre-heater.

$$Q = \dot{m}_{ref} \times (h_{out} - h_{in}) = \dot{m}_{ref} \times (x_{in} \times h_{fg} - h_{in}) \quad (1)$$

where, h_{out}, enthalpy of saturated liquid with the inlet quality, was calculated by measuring refrigerant pressure and temperature at the outlet of the pre-heater and h_{in} was the enthalpy of subcooled liquid at the inlet of the pre-heater. The condition of inlet air and evaporation temperature and pressure of the refrigerant were kept constant by controlling the temperature of indoor and outdoor chambers, respectively. The psychrometric calorimeter test facility consisted of indoor and outdoor chambers. The cooling capacity of the test section was measured by using the air sampling units at the indoor chamber. The air-side pressure drop through the test section was measured with a differential pressure transducer and the air flow rate was determined from the nozzle pressure difference. The absolute pressure was measured by a pressure transducer (10 bar range, ± 0.1 % resolution) at the inlet of the test section. And the refrigerant-side pressure drop through the evaporator was measured by a differential pressure gauge (1 bar range, ± 0.1% resolution).

2.2 Test conditions and parameters

Table 2 shows the experimental test conditions and parameters for the VCS and the RCS. The dry and web bulb temperatures were 27 °C and 19.5 °C, respectively.

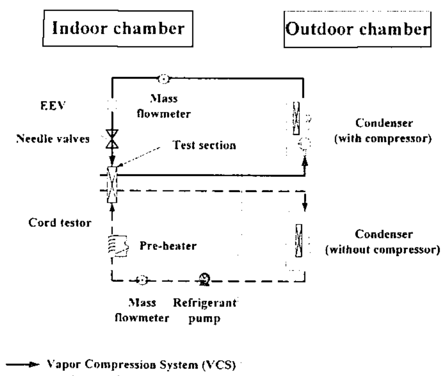


Fig. 1. Schematic of the experimental apparatus.

The air volume flowrate was constant at 17 CMM. The degree of superheat was controlled within 1 °C for the VCS while the evaporation temperature and pressure were 10 °C and 6.8 bar, respectively. The key parameters for the VCS were flow area ratio, number of return pipes, number of circuits and merging manifold. Mass flow rate, inlet quality and inclination angle were those for the RCS.

2.3 Prototype microchannel evaporators

Fig. 2 shows the typical features of the type 5 prototype evaporator. The type 5 consists of two parallel flow heat exchangers with inlet and outlet headers, respectively. It has 6 circuits and a merging manifold for collecting refrigerant from every circuit. Return pipes are installed to put together two heat exchangers mechanically. The refrigerant flows up and down through the return pipes and is thermally crossed with air. 18 thermocouples installed at the outlet of the test section measure the outlet air temperatures leaving the evaporator. The details of the louvered fin are summarized in Table 3.

Fig. 3 shows the outlet air temperatures for the type

Table 1. Key specification of the VCS.

Components	Specification
Refrigerant / Oil	R-22 / Mineral oil
Compressor	Rotary (2HP)
Evaporator	Microchannel evaporator
Condenser	Fin-tube heat exchanger
Expansion Device	Expansion & needle valve

Table 2. Test conditions and parameters.

Test	VCS	RCS
Conditions	(1) Air side - Dry bulb temperature : 27°C - Web bulb temperature : 19.5°C - Air volume flowrate : 17 CMM	
	(2) Refrigerant side	
	- Degree of superheat : within 1°C	- Temperature : 10°C - Pressure : 6.8bar
Parameters	- Flow area ratio - No. of return pipes - No. of circuits - Merging manifold	- Mass flow rate - Inlet quality - Inclination angle

5 without and with hydrophilic coating. The outlet air temperatures were not even at whole circuits, while average outlet air temperatures were reduced for the case with hydrophilic coating due to the increased drainage. In other words, the wettability of the condensate on louvered fins is enhanced, because the contact angle, defined as the angle between the liquid-vapor interface and the solid surface, increases after hydrophilic coating. This leads to an increase in the cooling capacity. All prototype evaporators, therefore, were hydrophilic coated to enhance the drain performance. A set of needle valves were installed in order to control the degree of the superheat in detail and minimize the effect of flow maldistribution resulting from a distributor placed at the inlet of all circuits. Fig. 4 shows the change of cooling capacity with hydrophilic coating and without hydrophilic coating as refrigerant change increases. It is found that the cooling capacity increased from 4.4% to 1.3% after hydrophilic coating.

The specification of all prototype evaporators is summarized in Table 4. Every evaporator consisted of two parallel flow heat exchangers connected with several return pipes as described for the type 5. The parallel flow heat exchanger consisted of inlet and outlet headers with the inner diameter of 19.4 mm and 41 parallel microchannel tubes. Each microchannel tube had 8 rectangular ports with a hydraulic diameter of 1.3 mm. Tube length and tube width are all 624 mm and 17 mm. The louvered fin had louver angle of 27°, louver pitch of 1.4 mm and flow depth of 18.8 mm. All prototype evaporators were hydrophilic-coated by the same procedures used in the automobile industry.

Flow area ratios for types 1, 2 and 3 were 100%, 61/66% and 73/58%, respectively. The flow area ratio represents the ratio of inlet flow area to outlet flow area of refrigerant. The inlet flow area equals the total cross section area of tubes at the inlet header of each circuit. The outlet flow area is for tubes at the outlet header of each circuit. For example, type 1 has 100% flow area ratio because the number of tubes for inlet header or second row at every circuit equals that for outlet header as shown in Fig. 2(b). Therefore, the number of microchannel tubes is constant for both first and second rows at each circuit. However, types 2 and 3 have 61/66% and 73/58% flow area ratio, which means that the outlet flow area is larger than the inlet flow area so that refrigerant can flow easily

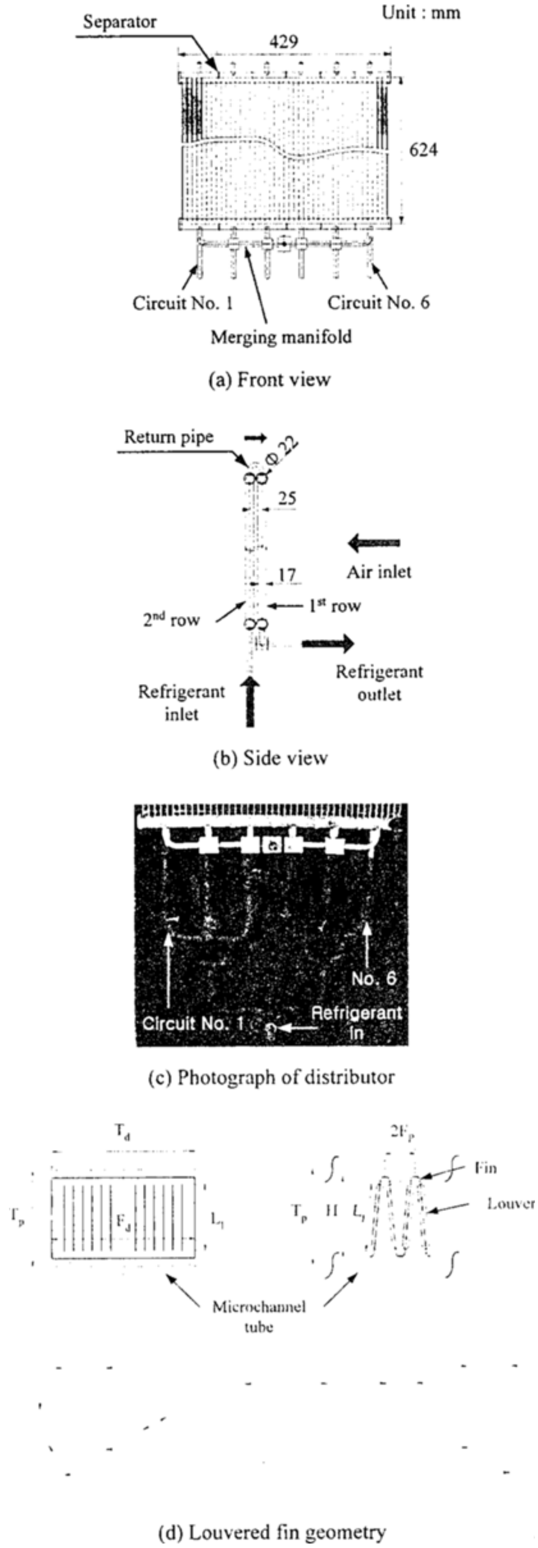


Fig. 2. Typical feature of the type 5.

Table 3. Details of the louvered fin

Item	Dimension	Item	Dimension
T_p	9.9 mm	L_l	6.6 mm
T_d	17 mm	H	8.02 mm
F_p	1.5 mm	θ	28 mm
F_d	18.8 mm	S_1	2.5 mm
L_p	1.4 mm	S_2	3.0 mm

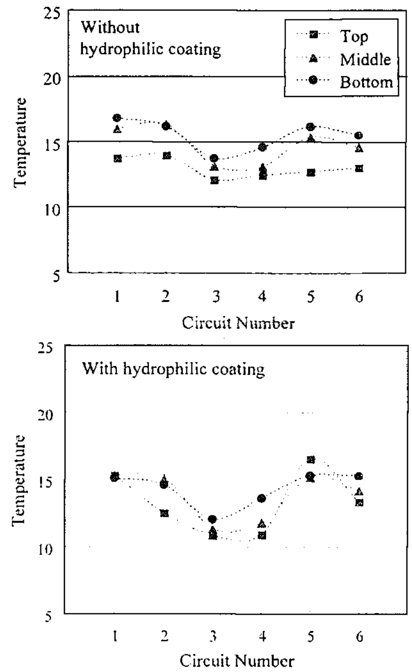


Fig. 3. Outlet air temperatures for hydrophilic coating.

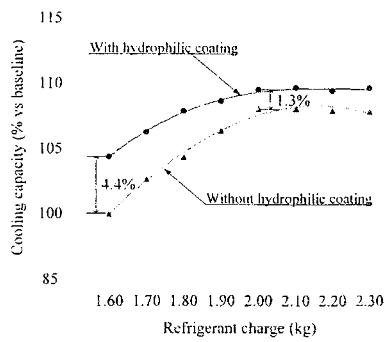


Fig. 4. Cooling capacity for hydrophilic coating.

through the tubes as it evaporates. This leads to a decrease in refrigerant pressure drop and the increase in mass flow rate and, consequently, the cooling capacity is improved.

Types 1 and 4 had the same geometry except for the number of return pipes. Types 4 and 5 also had the same geometry except for the number of circuits. Type 5 had a merging manifold at the outlet header for merging refrigerant from every circuit as shown in Fig. 2 (a), while type 6 did not have a merging manifold.

3. Results and discussion

3.1 Vapor compression system

Fig. 5 shows the various performances of the prototype microchannel evaporators for the vapor compression system(VCS). The cooling capacities of microchannel evaporators were calculated by the baseline of the cooling capacity of type 1 as shown in Fig. 5 (a). The cooling capacity is composed of two contributions of sensible and latent heat. The sensible heat ratios for all the evaporators ranged from 0.7 to 0.8, similar to that of conventional fin-tube heat exchangers. Both sensible and latent heats showed similar trends with the cooling capacity.

Types 1, 2 and 3 were tested in order to investigate the influence of the flow area ratio on the cooling capacity. In the case of an evaporator, generally, the cooling capacity can be improved by reducing the flow area ratio to a certain level but deteriorated in case of going beyond the limit. The cooling capacities of types 2 and 3 were larger by 6% to 15% than type 1, respectively. This result is consistent with the general observations and statements described in well known patents of the automobile industry. It leads that the flow area ratio is a very important design factor for applying the microchannel heat exchanger to the residential air conditioner as well as the automobile air conditioner. As shown in Fig. 2(b), refrigerant flows into the upper header of the first row through return pipes and then is redistributed into microchannel tubes of each circuit. In this case, flow redistribution characteristics in the first row strongly depend on the configuration of return pipes. The number of return pipes and their locations are very important for the flow distribution. In a distributing system that has one inlet at the header, the flow distribution characteristics are improved by increasing the number of the inlet as long as the flow rates are even at each inlet while the total flow rate through the system is constant. If the return pipes are regarded as several inlets for the upper header of the first row, they will work well for the flow distribution characteristics. However, the cooling capacity of type

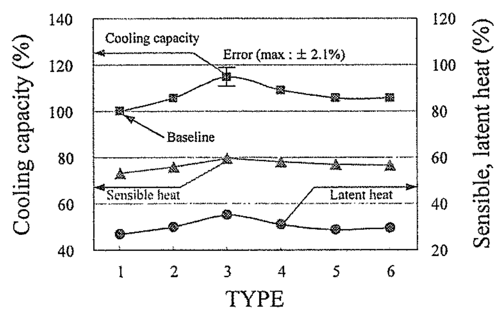
Table 4. Specification of the prototypes.

Parameters	Prototype					
	1	2	3	4	5	6
Flow Area Ratio	**100/100%**	**61/ 66%**	**73/ 58%**	100%	100%	100%
Number of return pipes	**12**	12	12	**6**	6	6
Number of circuits	3	4	3	**3**	**6**	6
Merging manifold	o	o	o	o	**o**	**x**

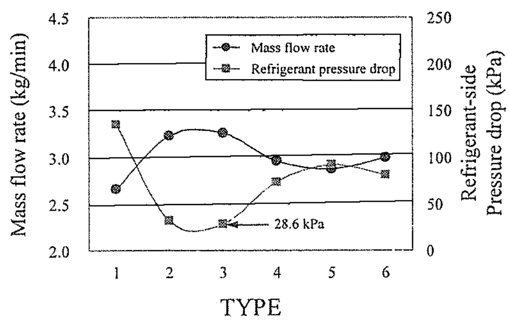
4 with 6 return pipes was larger by 8% than that of type 1 with 12 return pipes as shown Fig. 5(a). It can be explained that second redistribution characteristics from the upper header of the first row to micro-channel tubes were deteriorated by increasing the number of return pipes and changing the location of return pipes as well.

Obviously, the pressure drop of the type 3 with 12 return pipes is lower than any other type with smaller number of return pipes because the type 3 has much larger space or volume for refrigerant, resulting in little friction than that of types 4, 5 and 6 despite the fact that the mass flow rate of type 3 is larger than anything else as shown in Fig. 5(b). Moreover, the flow area ratio of type 3 is lower than that for other types, 4, 5 and 6. Lower flow area ratio gives a space or volume to the refrigerant during evaporation. It means that there is a little friction when refrigerant flows through a circuit with a lower flow area ratio. On the other hand, the pressure drop of the type 1 with 12 return pipes is larger than type 4 with 6 return pipes. The flow redistribution characteristics in the upper header of the first row are one of the important factors affecting the pressure drop due to the flow maldistribution. Therefore, we should consider the effect of flow area ratio and the number of return pipes simultaneously when comparing type 3 to types 4, 5 and 6 directly.

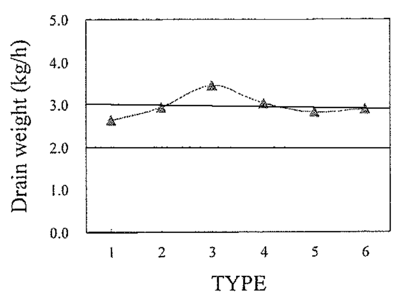
The effect of the number of circuits and merging manifold on the cooling capacity can be clearly understood by comparing the result between types 4, 5 and 6. The difference in the cooling capacity with respect to the number of circuits was small about 3% when comparing type 4 to type 5. Although the number of microchannel tubes per one circuit for type 5 was twice as large as that of type 4, there was little difference in the cooling capacity. It indicates that both type 4 and 5 have similar flow distribution characteristics. The effect of merging manifold on the cooling capacity was negligible within 1% when comparing type 5 to type 6. It was revealed that the



(a) Cooling capacity



(b) Mass flow rate and pressure drop



(c) Drain weight

Fig. 5. Performance of prototype evaporators for the VCS.

flow distribution characteristics were slightly affected by the pressure drop resulting from the configuration of the refrigerant flow at the exit of the evaporator.

The mass flow rate shows the opposite trend to the refrigerant side pressure drop as shown in Fig. 5(b).

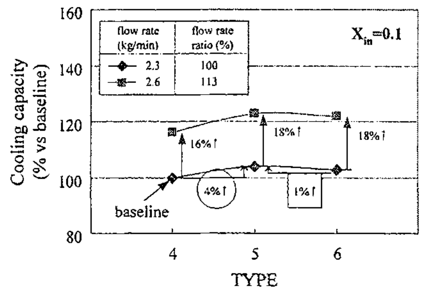
The mass flow rate through an evaporator can be changed due to flow maldistribution from its original configuration although the same compressor operates in the same system with tube length and diameter unchanged. In other words, the refrigerant gets much friction when the pressure drop resulting from the flow configuration of prototypes is large. It means that the smaller pressure drop, the larger mass flow rate. This leads to an increase in the cooling capacity.

It is clearly understood that the mass flow rate showed similar trends with the cooling capacity when comparing the cooling capacity of Fig. 5(a) to the mass flow rate of Fig. 5(b). Type 3 showed the maximum cooling capacity and the mass flow rate, while it showed the minimum pressure drop of 28.6 kPa. The pressure drop of type 3 is appropriate for residential air-conditioning application. As shown in Fig. 5(c), the drain weight also shows similar trends with the cooling capacity of Fig. 5(a). The cooling capacity strongly depends the drain weight of the evaporator under the wet condition.

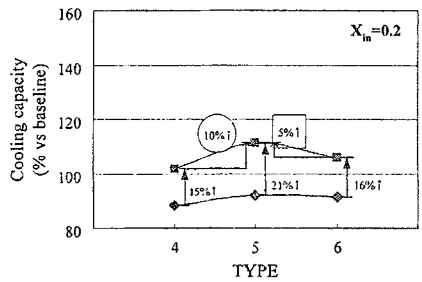
3.2 Refrigerant circulation system

Fig. 6 shows the effect of the mass flow rate and inlet quality on the cooling capacity with respect to types 4, 5 and 6. The cooling capacities increased by 16-18% at the inlet quality of 0.1 and 15-21% at the inlet quality of 0.2 as the mass flow rate increased by 13% from 2.3 to 2.6 kg/min. The real increase in the cooling capacity was about 5-8% based on simple arithmetic calculation considering the increase in the mass flow rate. Similarly, the real increase in the cooling capacity was 1-5% for the change of inlet quality. Consequently, it was found that the effect of the mass flow rate on the cooling capacity was slightly superior to that of the inlet quality.

In view of the number of circuits, the cooling capacity of type 5 was larger by 4-10% than type 4. The distributor for type 5 had 6 circuits as shown in Fig. 2 (c) and the distributor for type 4 was almost the same with the half of the distributor for type 5. Type 5 with 6 circuits showed better flow distribution characteristics than type 4 with 3 circuits due to the distributors as well as the mass flow rate and inlet quality. It means that flow distribution characteristics can be improved by increasing the number of circuits under the RCS. This result was quite different from the result of the VCS. This is because that there were some control devices, an EEV and six needle valves, for both making an equal distribution at each circuit

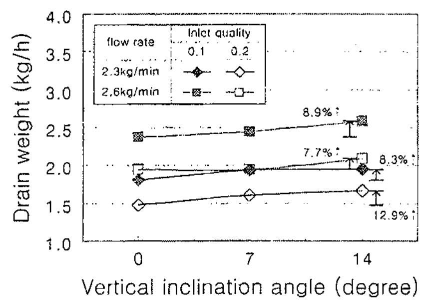


(a) Inlet quality of 0.1

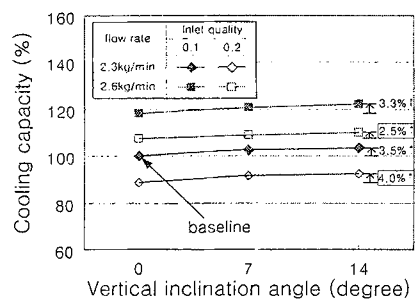


(b) Inlet quality of 0.2

Fig. 6. Effect of mass flow rate and inlet quality.



(a) Drain weight



(b) Cooling capacity

Fig. 7. Effect of vertical inclination angle.

in order to eliminate the influence of the distributor and controlling the degree of the superheat within 1℃. The effect of the merging manifold on the cooling capacity was negligible when comparing type 5 to type 6. This result was consistent with that of the VCS.

Fig. 7 shows the drain weight and the cooling capacity with respect to the inclination angle of type 5 under the RCS. The drain weight proportionally increased as the inclination angle increased due to the gravity force on the condensate inside louvered fins. The cooling capacity also increased as the inclinational angle increased. The cooling capacity increased from 2.5% to 4.0% while the drain weight increased from 8.3% to 12.9%. In general, the more drain weight leads to the increase in the portion of latent heat to the cooling capacity. The ratio of latent heat to the cooling capacity for the RCS is about 30% since multiplying 8.3% and 12.9% by 0.3 is equal to 2.5% and 3.9%, respectively. This ratio is compatible with the typical result of the VCS.

4. Conclusions

Two different methods were applied to evaluate the performance of prototype microchannel evaporators for the residential air-conditioner. For the vapor compression system, the flow area ratio and the number of return pipes had a great effect on the cooling capacity. Type 3 with a flow area ratio of 73 and 58% showed the best cooling capacity. The effect of the number of circuits and merging manifold on the cooling capacity was relatively small. For the refrigerant circulation system, the effect of the mass flow rate on the cooling capacity was slightly superior to that of inlet quality. The effect of the number of circuits on the cooling capacity was different with the result of the vapor compression system. The effect of merging manifold was negligible, which was consistent with the result of the vapor compression system. The cooling capacity proportionally increased as the vertical inclination angle of the evaporator increased due to gravity force.

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Nomenclature

F_p : Fin pitch, mm
F_d : Flow depth, mm
H : Fin height, mm
L_p : Louver pitch, mm
L_l : Louver length, mm
S_1 : Length of flow inlet, mm
S_2 : Length of flow outlet, mm
T_p : Tube pitch, mm
T_d : Tube depth, mm
θ : Louver angle, °

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