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EFFECT OF GRAVITY ON THE DISTRIBUTION OF CO_2 IN A PARALLEL FLOW EVAPORATOR

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ABSTRACT

In this study, the distribution characteristics of CO_2, a promising natural refrigerant, were experimentally investigated for a parallel flow evaporator with 10 flat aluminum tubes. The tubes have the length of 1000 mm and each tube has six micro-channels with a diameter of 0.8 mm. To detect the flow distribution into the tubes, the wall temperatures were measured along the tubes which are heated by flexible electric resistance heating wires. If the mass flow rate in a tube is lower than others, the superheating of the refrigerant would occur earlier in that tube and the corresponding wall temperature become higher. To investigate the effect of the gravity, three installation positions of the evaporator were considered; vertical headers/ horizontal tubes (case 1), horizontal headers/ horizontal tubes (case 2), and horizontal headers/ vertical tubes (case 3). Experimental results showed that case 1 was most greatly influenced by the gravity. The refrigerant vapor was supposed to be collected at the upper part of the inlet header because of the gravity. The refrigerant distribution becomes better with the increase of the evaporating temperature and mass flux.

INTRODUCTION

Studies on natural refrigerants are gaining more interest because they have much lower GWP(Global Warming Potential) than most of the artificial refrigerants. Among the natural refrigerants, CO_2 is expected to be one of the promising refrigerants since it has good thermodynamic characteristics and is not toxic or flammable. However, its operating pressure is very high, and heat exchangers are required to endure high pressure in CO_2 refrigeration system. Parallel flow heat exchangers which are widely used in conventional refrigeration systems are thought to be suitable for CO_2 systems. In a parallel flow evaporator, the heat transfer performance can be seriously deteriorated if the refrigerant in a header is unevenly distributed into each tube. The distribution characteristics are dependent on the shape of the evaporator, the position of the inlet and outlet pipe, and the installation position. They are also influenced by the properties of the refrigerant. Some studies have been carried out on refrigerant distribution in parallel flow heat exchangers.

Azzopardi and Whalley[1] tried to find the mechanism of flow inside by using T-junction. Vist and Petterson[2] performed experiments on the distribution of in a compact parallel flow heat exchanger with R134. The evaporator they used had circular tubes with inner diameter of 4mm. They installed the valves at each tube and investigated distribution characteristics of the refrigerant. Kim and Sin[3] made a PVC header pipe, and observed the 2-phase flow in it. Since the apparatus could not endure high pressure, they used water and air for the experiments.

NOMENCLATURE

a	[m^2]	Cross section area of a channel
A	[m^2]	Inner wall area of a tube
c_{ij}	[m^2K/W]	Constants for temperature modification (Eq. 6)
D	[m]	Diameter of a channel
$\dot{m}$	[kg/s]	Mass flow rate
m''	[kg/m^2s]	Mass flux
n_{ch}		Number of channels (n_{ch} =6)
n_{tube}		Number of tubes (n_{tube} =10)
$\dot{Q}$	[W]	Heat flow rate for a tube
q''	[W/m^2]	Heat flux
R	[Ω]	Electric resistance
T	[°C]	Temperature
V	[V]	Electric voltage applied to heating wires

Subscripts

avg	Average
cal	Calibration
dif	Difference
eva	Evaporation
i	Tube number
j	Location number
mea	Measured
mod	Modified
sv	Saturated vapour

Marchitto et al. [4] used transparent header pipes and branching tubes. They also used water and air to simulate the 2-phase flow. Therefore, it was impossible to verify how the refrigerant really distributed into each tube under the real operation condition.

Since CO_2 evaporators should endure a high pressure, micro channel tubes with smaller diameters are used compared with conventional parallel flow evaporators. Also, the header should be designed to have enough strength. Furthermore, the property of CO_2 in liquid-vapor equilibrium is much different from air-water mixture or conventional refrigerants. Therefore, the results of previous studies might not be directly applied to CO_2. From this background, it was decided to investigate the refrigerant distribution in a parallel flow evaporator with CO_2 in different operation conditions including the direction of the gravity.

EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 shows the schematic diagram of the experimental apparatus. The test section is a parallel flow evaporator with 10 tubes. CO_2 is evaporated by the heat from the heating wire wrapping the tubes. The CO_2 vapor is condensed in a heat exchanger which is cooled by a thermostat. Sub-cooled liquid state CO_2 is separated in an accumulator, and it is circulated by a magnetic gear pump. The mass flow rate is measured by a Coriolis mass flow meter. An absolute pressure transmitter (A.P.T.) detects the inlet pressure; and a differential pressure transmitter (D.P.T.) measures the pressure difference between the inlet and outlet of the test section. The refrigerant flow is observed at two sight glasses which are installed at the entrance and the exit of the test section.

The test section (Figure 2) consists of ten parallel extruded flat tubes, two headers, and connecting pipes which are all made of aluminium. To be able to select the position of the incoming or leaving refrigerant, three connecting pipes are attached at the inlet and exit header, respectively. At each header only one connecting tube is open and other two tubes are closed for a given experiment. T-type thermocouples were attached on the outer wall of the tube to estimate the distribution of CO_2. For each tube which has the length of 1000mm, temperatures were measured at five locations with an interval of 240mm. Location 1 to 5 correspond to 10, 250, 490, 730, and 970 mm from the inlet header.

To investigate the effect of the gravity, three installation positions of the evaporator were considered; vertical headers/ horizontal tubes (case 1), horizontal headers/ horizontal tubes (case 2), and horizontal headers/ vertical tubes (case 3).

Figure 3 shows the cross section of the micro-channel tube. It has six circular channels with a diameter of 0.8mm, and the thickness of the tube is 1.6mm.

Figure 4 shows the cross section of header pipe and the tube arrangement. The inner diameter of header is 15mm and its thickness is 5mm. Each tube is installed with an interval of 15mm and inserted 5mm into the header.

As shown in Figure 5, the tube is closely wound with a flexible dielectric heating wire. To avoid the direct contact with the heating wire, a small gap is maintained between the thermocouples and the heating wire.

The whole system is thermally insulated by EPDM(Ethylene Propylene Terpolymers). Before charging the system with CO_2 refrigerant, the system was evacuated by a vacuum pump. The charging amount was measured with an electronic scale. For each experimental condition, the data were collected for 2 minutes when the system reached steady state.

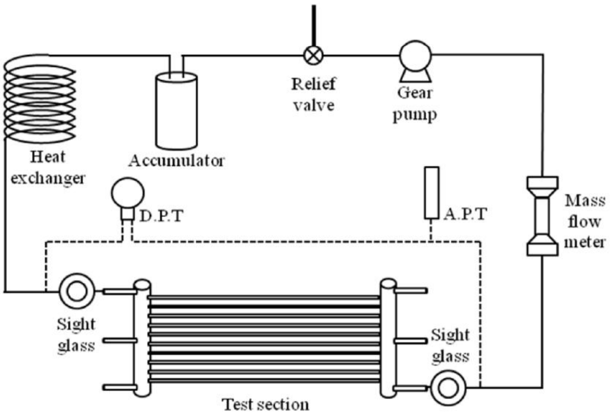

Figure 1 Schematics of experimental apparatus

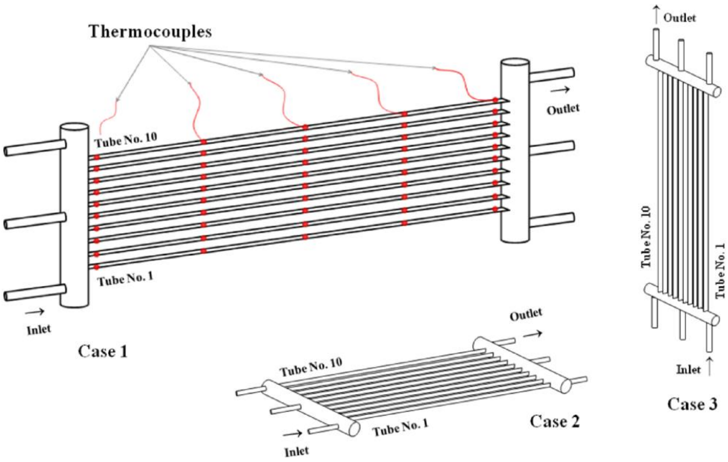

Figure 2 Test section and installation position

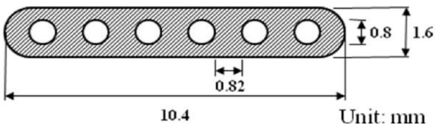

Figure 3 Cross section of the micro-channel tube

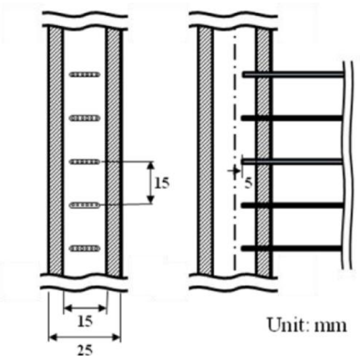

Figure 4 Details of the header

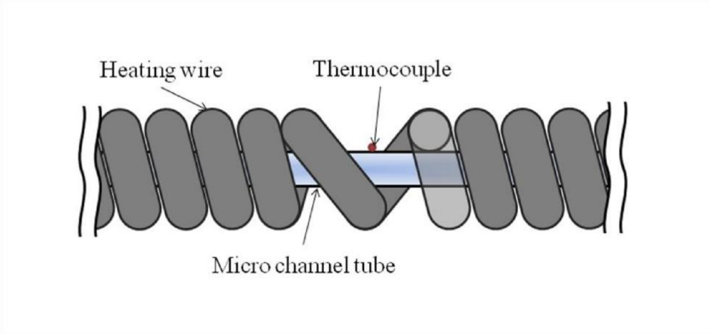

Figure 5 Heating wire and thermocouples

DATA REDUCTION

The heat transfer rates are the same for all tubes, because the heating wires have the same electric resistance. For each tube, the heat transfer rate is calculated from the electric resistance of a heating wire and the applied voltage.

$$\dot{Q} = \frac{V^2}{R} \tag{1}$$

The mass flux and heat flux at the test section are calculated as follows.

$$q'' = \frac{\dot{Q}}{A} = \frac{\dot{Q}}{n_{ch}\pi DL} \tag{2}$$

$$m'' = \frac{\dot{m}}{n_{tube}n_{ch}a} = \frac{4\dot{m}}{n_{tube}n_{ch}\pi D^2} \tag{3}$$

Because the thermocouples attached to the outer tube wall are influenced by the heating wire, cautions were made to give gap distance as uniformly as posssible for all measuring points. However, after some preliminary runs, it was found that a calibration was necessary to correct the eliminate the effect of the uneven gaps.

Therefore, the experiments for the calibration were performed with the case 3 in which the refrigerant is expected to be distributed into each channel most evenly. In these experiments, the mass flux of refrigerant and evaporation temperature were fixed, while the heat flux was changed.

Figure 6 shows the temperatures measured for 10 tubes at the location 1 (10mm from the inlet). For other locations, the temperatures showed similar deviations. The average temperature of the location j is defined as follows:

$$T_{j,avg} = \frac{1}{10}\sum_{i=1}^{10} T_{ij,mea,cal} \tag{4}$$

The difference between the measured and the average temperature is defined as the temperature difference.

$$T_{ij,dif} = T_{j,avg} - T_{ij,mea,cal} \tag{5}$$

This temperature differences are almost proportional to the heat flux, which is also similar for other locations.

$$T_{ij,dif} = c_{ij} \times q'' \tag{6}$$

Where c_{ij} is a proportionality constant. After deciding 50 c_{ij} values from the experimental data for the calibration, all the measured temperatures in the main experiments could be modified as follows.

$$T_{ij,\mathrm{mod}} = T_{ij,mea} + T_{ij,dif} \tag{7}$$

Figure 7 shows an example of the temperature modification (case1, 120kg/m^{2}s, 10^{o}C). Measured temperatures are shown in a), and modified temperatures in b). It can be seen that the deviations of the temperatures are corrected, if modified temperatures are used.

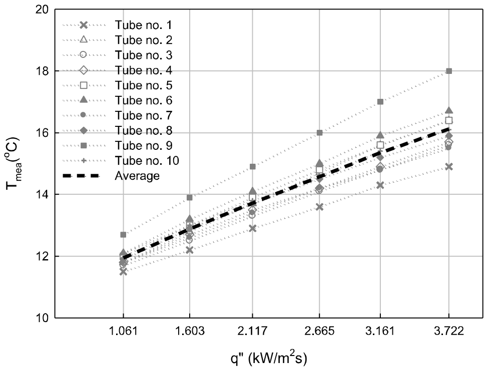

Figure 6 Deviation of temperatures for location 1

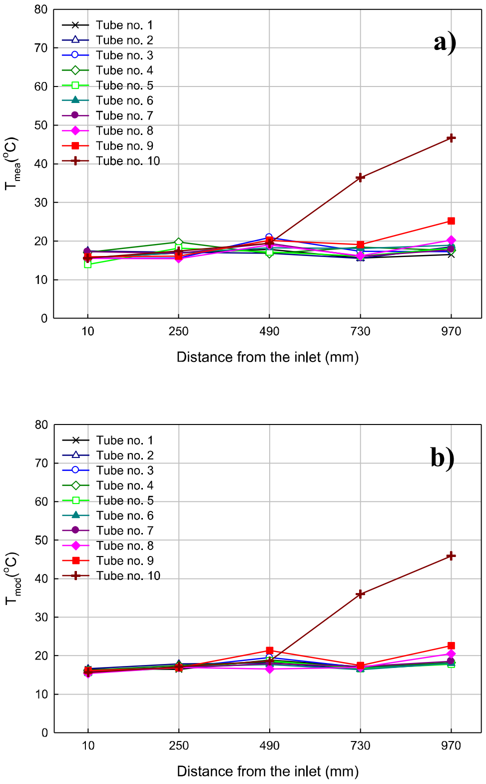

Figure 7 An example of temperature modification (Case1, m"=120kg/m^{2}s, T_{sat}=10°C, $q''=q''_{sv}$)

RESULT AND DISCUSSION

Figure 8 shows the effect of the direction of gravity for the heat flux equal to the saturation heat flux ($q''=q''_{sv}$). In this condition, the refrigerant would be in saturated vapour state at the tube exit if the refrigerant is evenly distributed into each channel. However, if some tubes have smaller mass flow rates, superheating of the vapor would occur and their wall temperature would increase. In Figure 8 a), the top tube (#10) showed a temperature increase in location 4 and 5. This indicates the mass flow rate in tube #10 is smaller than that of other tubes. It seems that the bubbles generated in the micro-channel flow back to the inlet header by instability and are gathered in the upper part of the header by gravity, as shown in Figure 9 a).

In Figure 8 b), no tube showed superheating for case 2, which indicates even distribution of the refrigerant. If some vapour exists in the header, this would not influence the distribution of the refrigerant as shown in Figure 9 b). For case 3 (Figure 8 c)), tube #1 which is most close to the refrigerant inlet showed a slight superheating. This result is similar to the work of Kim and Sin[3]. In the same gravity direction, they observed that the tubes near the refrigerant inlet had slightly smaller mass flow rates than other tubes.

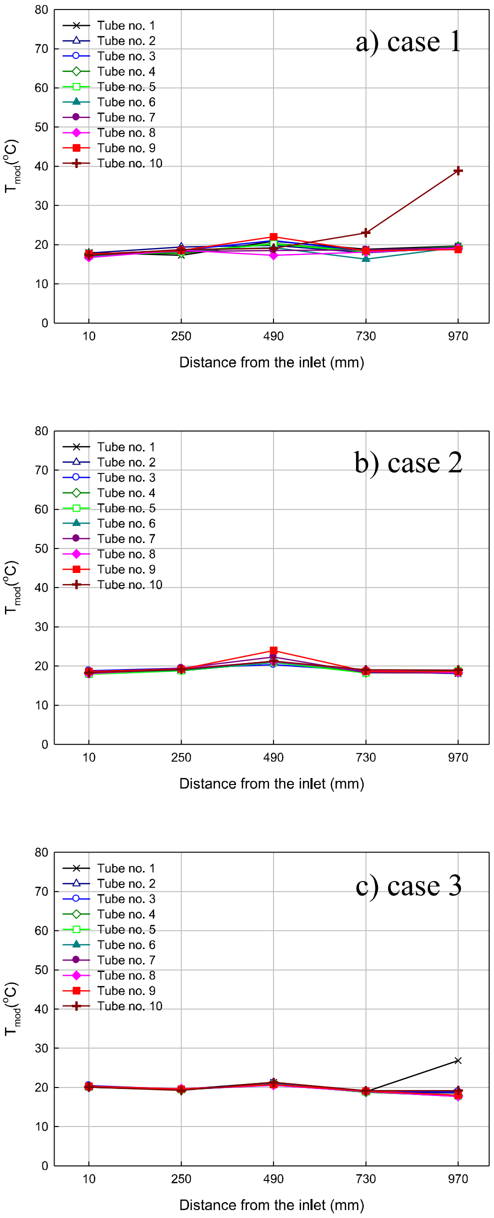

Figure 8 Effect of gravity on the refrigerant distribution (m"=160kg/m^{2}s, T_{eva}=10°C, $q''=q''_{sv}$)

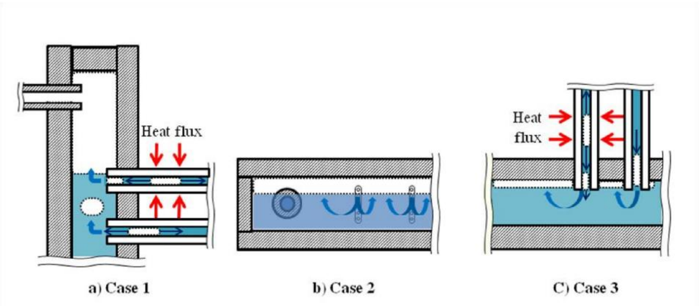

Figure 9 Expected liquid-vapor distribution in a header

Figure 10 shows the effect of the direction of gravity if the heat flux is 110% of the saturation heat flux ($q''=1.1\times q''_{sv}$). In this condition, the refrigerant would be superheated by the equal amount if the refrigerant is evenly distributed into each channel. However, for case 1, the degree of superheating becomes bigger as the tube number increases. This indicates the mass flow rate becomes smaller at upper tubes by the same reason described above. For case 2, the degree of superheating was almost the same for every tube and showed no special trend. For case 3, the degree of superheating was slightly bigger at tube #1.

The effect of the evaporating temperature on the refrigerant distribution is shown in Figure 11 for case1. As the evaporating temperature is increased, it is found that the refrigerant distribution becomes better. In CO$_2$ system, the difference of the property, especially the density, between the liquid and vapour phase becomes smaller, as the evaporating temperature approaches the critical one (31.0°C). Therefore, the effect of gravity seems to exhibit smaller effect on the distribution at higher evaporating temperature.

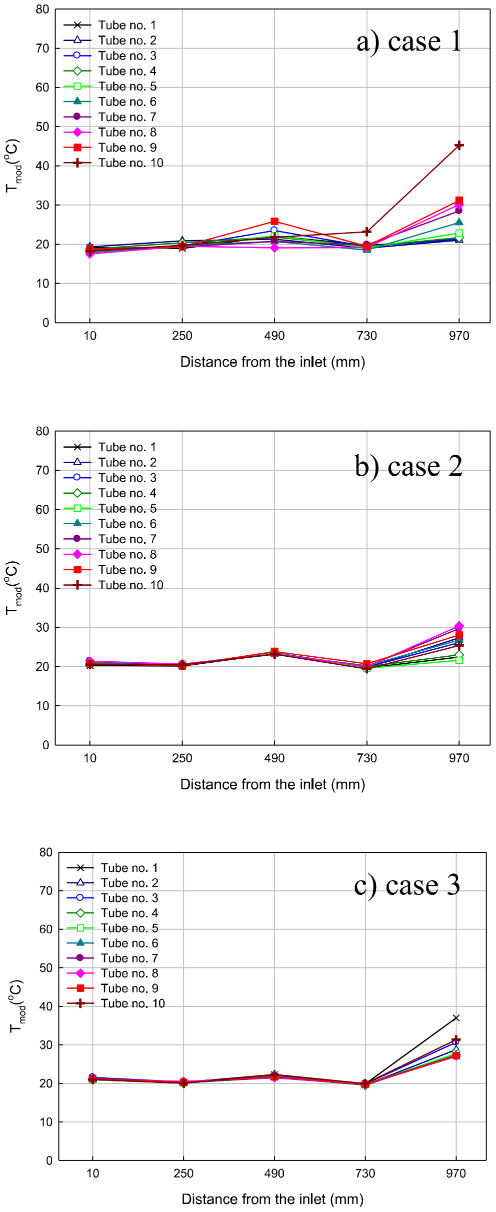

Figure 10 Effect of gravity on the refrigerant distribution (m”=160kg/m^{2}s, T$_{eva}$=10°C, $q''=1.1\times q''_{sv}$)

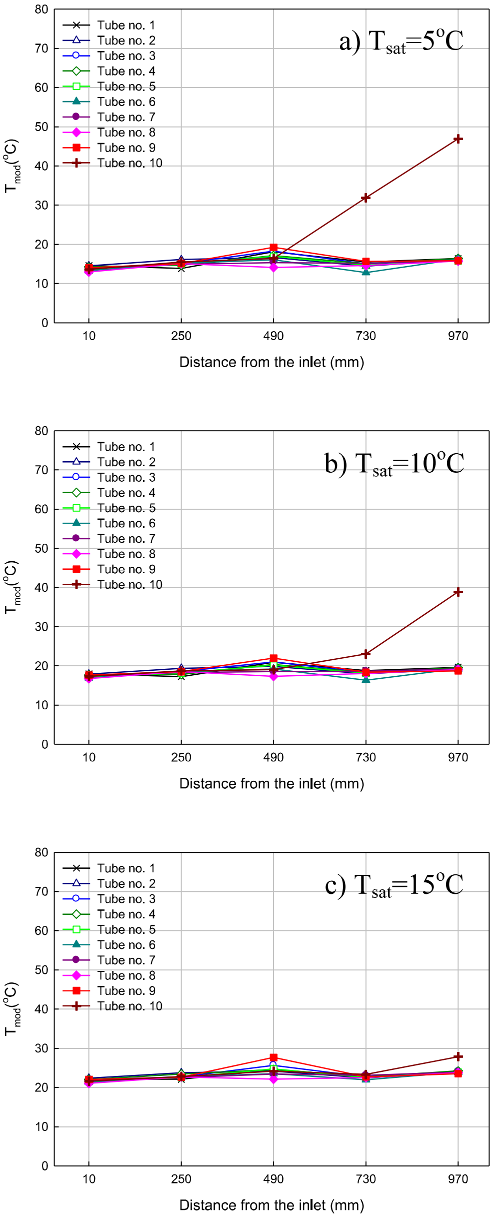

Figure 11 Effect of evaporating temperature on the refrigerant distribution (Case 1, m”=160kg/m^{2}s, $q''=q''_{sv}$)

Figure 12 shows the effect of mass flux on refrigerant distribution for case1. As the mass flux is increased the refrigerant distribution gets better. It seems that fewer bubbles return to the inlet header and less vapour is collected at the upper part of the header at higher mass flow rate.

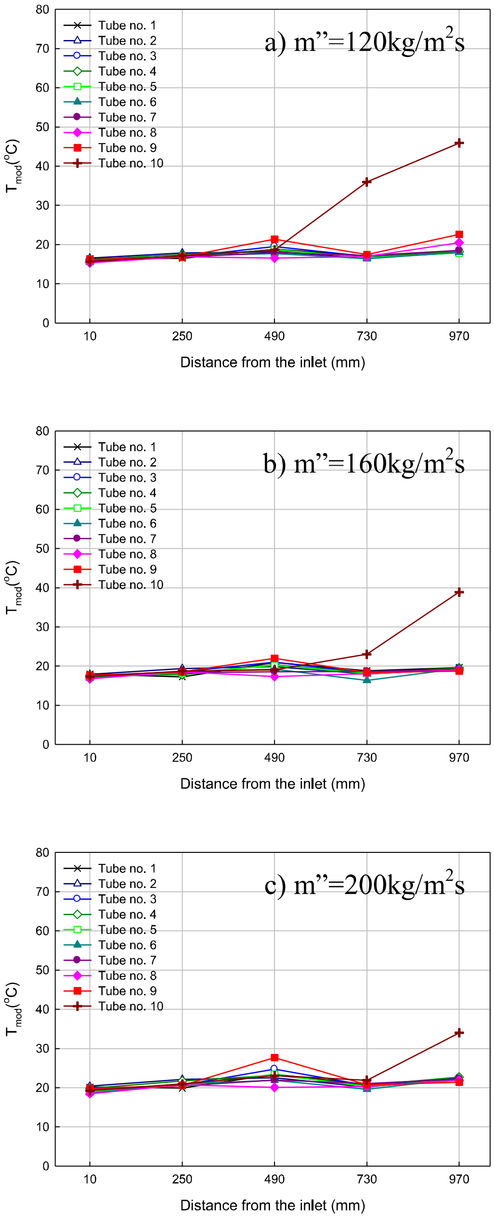

Figure 12 Effect of mass flux on the refrigerant distribution (Case 1, T_{eva}=10°C, $q''=q''_{sv}$)

CONCLUSION

In this study, the distribution characteristics of CO_2 were experimentally investigated for a parallel flow evaporator with 10 flat aluminum tubes. The tubes have the length of 1000 mm and each tube has six micro-channels with a diameter of 0.8 mm. To investigate the effect of the gravity, three installation positions of the evaporator were considered; vertical headers/ horizontal tubes (case 1), horizontal headers/ horizontal tubes (case 2), and horizontal headers/ vertical tubes (case 3). Major findings in this study are summarized as follows.

- Refrigerant distribution was most greatly influenced by the gravity in case 1. The refrigerant vapor was supposed to be gathered at the upper part of the inlet header because of the gravity. The top tube (#10) showed a temperature increase at location 4 and 5 indicating that its mass flow rate is smaller than that of other tubes.
- As the evaporating temperature is increased, it is found that the refrigerant distribution becomes better. The effect of gravity seems to exhibit smaller effect on the distribution at higher evaporating temperature, because the difference of the property, especially the density, between the liquid and vapour phase becomes smaller, as the evaporating temperature approaches the critical one (31.0°C).
- As the mass flux is increased the refrigerant distribution gets better. It seems that fewer bubbles return to the inlet header and less vapour is collected at the upper part of the header at higher mass flow rate.

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