MEASUREMENT OF FLOW BOILING HEAT TRANSFER COEFFICIENT OF CO_2-OIL MIXTURES INSIDE A COPPER TUBE USING ELECTRICAL RESISTANCE METHOD

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Abstract: The flow boiling heat transfer coefficients of pure CO_2 and CO_2-oil mixtures inside a micro-fin copper tube were measured at saturation temperatures of 0 and 10 ˚C using electrical resistance method. The micro-fin tube is 5.43 mm in mean inner diameter and is heated using a film heater. In the case of pure CO_2, the results showed that the flow boiling heat transfer coefficients at the saturation temperature of 10 ˚C are slightly higher than those of 0 ˚C. The flow boiling heat transfer coefficients increase with the increase in heat flux, but are almost independent of mass velocity. These results imply that the flow boiling heat transfer in the case of pure CO_2 is dominated by nucleate boiling. In the case of CO_2-oil mixtures, the heat transfer coefficients depend on mass velocity, but are independent of heat flux. So the flow boiling heat transfer is considered to be dominated by forced convective evaporation. From the flow visualization results, the foaming phenomenon was not observed, but it is found that an oil film was formed on the inner wall surface, and a lot of oil droplets are observed in the liquid phase of the refrigerant.

Key Words: experiment, CO_2 flow boiling, flow visualization, micro-fin tube, PAG oil

1   INTRODUCTION

Due to the global warming issue, carbon dioxide (CO_2) is expected to be used as a working fluid for automobile air-conditioners and heat pump systems. In Japan, in order to reduce CO_2 emissions, a CO_2 heat pump system for the residential hot water heater was developed. The improvement in the performance of the CO_2 heat pump system is an immediate research task. On the other hand, micro-fin tubes are used extensively for refrigeration and air-conditioning equipment of fluorocarbon refrigerant, and have made a huge contribution to the improvement of heat exchangers for fluorocarbon refrigerants. The use of micro-fin tubes in the CO_2 heat pump system is expected as one of the effective measures of performance improvement.

Several researchers have reported about the flow boiling of CO_2 in micro-fin tubes. Schael and Kind (2005) reported flow pattern and boiling heat transfer in a helical grooved micro-fin tube of 9.52 mm O.D. Their flow patterns were compared with those proposed by Thome and El Hajal (2003). They also showed that the heat transfer coefficients of the micro-fin tube are very much higher than those of smooth tubes. Cho et al. (2006) carried out experiments on flow boiling heat transfer for CO_2 in horizontal smooth and micro-fin tubes of 5 and 9.52 mm

O.D. They reported that the heat transfer coefficients of micro-fin tubes on heat transfer are 1.5–2.1 times higher than those of smooth tubes.

Since PAG oil is used as the lubricating oil in the CO_2 heat pump system, it is another important task to clarify the influence of the PAG oil on CO_2 flow boiling heat transfer. Tanaka et al. (2001) gave an experimental report on the flow boiling heat transfer and dryout of CO_2 in stainless steel smooth tubes, and pointed out that the evaporation heat transfer characteristics of CO_2 were different from those of fluorocarbon refrigerants. They also showed that the heat transfer coefficients decrease remarkably at oil concentration larger than 0.7mass%. Katsuta et al. (2002) investigated the flow boiling heat transfer characteristics of CO_2 in a stainless steel smooth tube of 4.59 mm I.D. at the conditions of 1 and 5mass% oil concentrations. Gao and Honda (2002) reported the flow boiling heat transfer characteristics of CO_2 and the influence of the PAG oil inside a copper smooth tube of 5 mm in inner diameter using the Wilson plot. Koyama et al. (2005) reported the flow boiling heat transfer and the influence of PAG oil in copper smooth and micro-fin tubes. The mentioned research showed that the flow boiling heat transfer coefficients decrease remarkably in the case of CO_2-oil mixtures. However, there is a large difference between each research about how much amount of the oil will give a remarkable decrease on the heat transfer coefficient.

Gao et al. (2007) experimentally investigated the flow boiling heat transfer of CO_2–PAG oil mixtures in horizontal smooth and micro-fin tubes of about 3 mm in inner diameter. Their test section was heated by a film heater. They measured the real local heat transfer coefficient by means of measuring the wall temperatures using thermocouples which are mounted under the film heater. They found that the heat transfer coefficient decreased by about 50% compared with those of pure CO_2 when oil circulation ratio was more than 0.11mass%. They showed that the enhancement effect of the micro-fin tube on local heat transfer coefficients appears significantly at higher vapor quality, higher mass velocity and lower oil circulation ratio conditions. They also showed that the dryout quality decreases significantly with increase in mass velocity for the smooth tube, but is nearly independent of mass velocity for the micro-fin tube. The effect of the oil on dryout quality is negligible for both smooth and micro-fin tubes. However, their test section of micro-fin tube is hard to be manufactured, and many techniques are required when manufacturing their test sections.

As mentioned above, the research on heat transfer characteristics of micro-fin tubes for CO_2 is still insufficient. The enhancement effects of micro-fin tube on the flow boiling heat transfer are not clarified yet especially in the case of CO_2-oil mixtures. And development of a simple measuring method with high measurement accuracy for copper tubes is expected. In this study, experiments on the flow boiling heat transfer of pure CO_2 and CO_2-oil mixtures inside a micro-fin copper tube were conducted at saturation temperatures of 0 and 10 ˚C using electrical resistance method. And observation of flow situation of CO_2-oil mixtures was also conducted.

2   EXPERIMENTAL APPARATUS AND METHOD

The experimental apparatus used in the present research is shown in Figure 1. It is a heat pump system consisting mainly of a two-stage compressor(1), an oil-separator(2), two gas-coolers (4, 5), two expansion valves(12a, 12b), a pre-heater(13), a test section(15), an auxiliary evaporator(17), and an accumulator(18). In order to observe the flow situation of the oil mixtures, three sightglass(6a, 6b, 6c) was installed after the gascooler, before and after the test section. The mass flow rates were measured using two coriolis type mass flow meters(7a,7b), for measuring the total mass flow rate and that flowing through the test section. The precisions of the mass flow rate meters are ±0.25%full-scale(Full-scale:100kg/h). The pressures and bulk temperatures at the mixing chambers(11) prepared before the

expansion valve and the outlet of the test section were measured using two pressure transducers and two sheathed thermocouples (There is no mixing chamber at both ends of test section). The precisions of the pressure transducers and sheathed thermocouples are ±0.01 MPa and ±0.05 K, respectively. The pressure drop over the test section was measured using a pressure difference transducer(14) with a precision of ±0.2 kPa. The circulation ratio of the oil was adjusted by the oil-control valve(3).
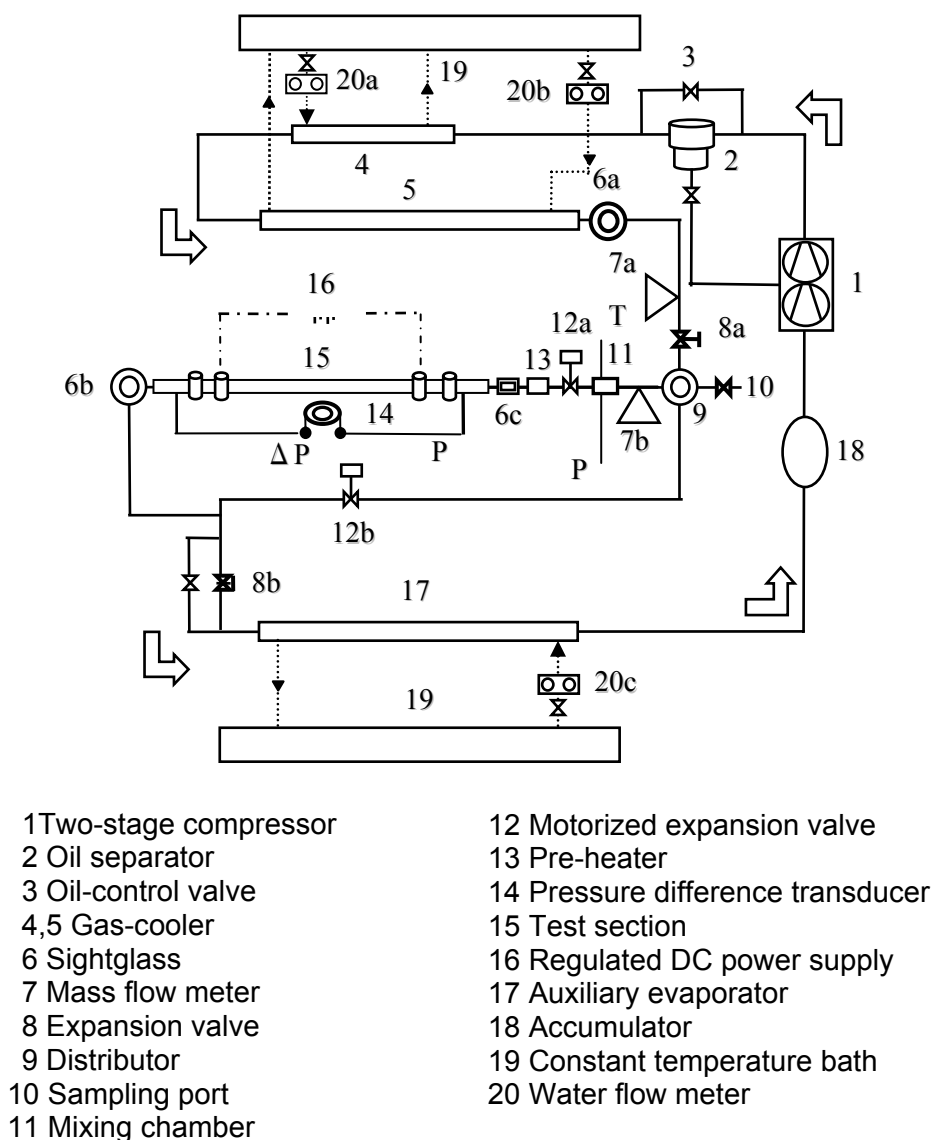

1 Two-stage compressor
2 Oil separator
3 Oil-control valve
4,5 Gas-cooler
6 Sightglass
7 Mass flow meter
8 Expansion valve
9 Distributor
10 Sampling port
11 Mixing chamber
12 Motorized expansion valve
13 Pre-heater
14 Pressure difference transducer
15 Test section
16 Regulated DC power supply
17 Auxiliary evaporator
18 Accumulator
19 Constant temperature bath
20 Water flow meter

Figure 1: Schematic diagram of experimental apparatus

The test section is a horizontal micro-fin copper tube, 5.43 mm in mean inner diameter and 1.78 m in total length, and it is heated by a film heater which is wound outward of the tube. The test section is divided into 4 subsections. Each subsection is 0.3 m in effective heating length. The voltage terminals are attached to both ends of each subsection, and the electric potential drops are measured. The tube wall temperatures of each subsection were measured using electrical resistance method: the temperatures are evaluated from calibrated relation between the wall temperature and electrical resistance by measuring the electric resistance of the tube. The relation between the wall temperature and electric resistance for each subsection was calibrated by measuring the saturation temperature (pressure) and the electric resistance at the conditions without heating the test section. In order to obtain a high

measurement accuracy, the electric current for measuring the wall temperature was set larger than 10 A. Accordingly, the change in voltage at each subsection can be obtained larger than 10mV. The heat flux owing to the electric current for measurement is less than 20 W/m^2. The temperature difference between the tube wall and refrigerant was calibrated less than 0.1 K. The uncertainty of heat transfer coefficient is estimated as less than 14% by uncertainty analysis. The electric currents for heating and measuring wall temperature were generated by two regulated DC power supply, and were measured using two 1 mΩ (±0.01%) standard resistors. Figure 2 shows the details of the micro-fin tube test section, and the dimensions of the micro-fin tube are shown in Table 1.

The oil circulation ratio (OCR) was measured using the oil-droplet methods proposed by Gao and Honda (2006). The CO_2 solubility and the density of the PAG oil were measured by Gao et al. (2008) at the pressure range of 3.5-5 MPa. For measuring the OCR, the CO_2 solubility in the oil droplets and the density of the oil were taken into consideration using the data of Gao et al. The thermophysical properties of CO_2 were calculated using software package PROPATH (2001). The purity of CO_2 used in the present research is 99.95mass%. The experimental conditions are shown in Table 2.
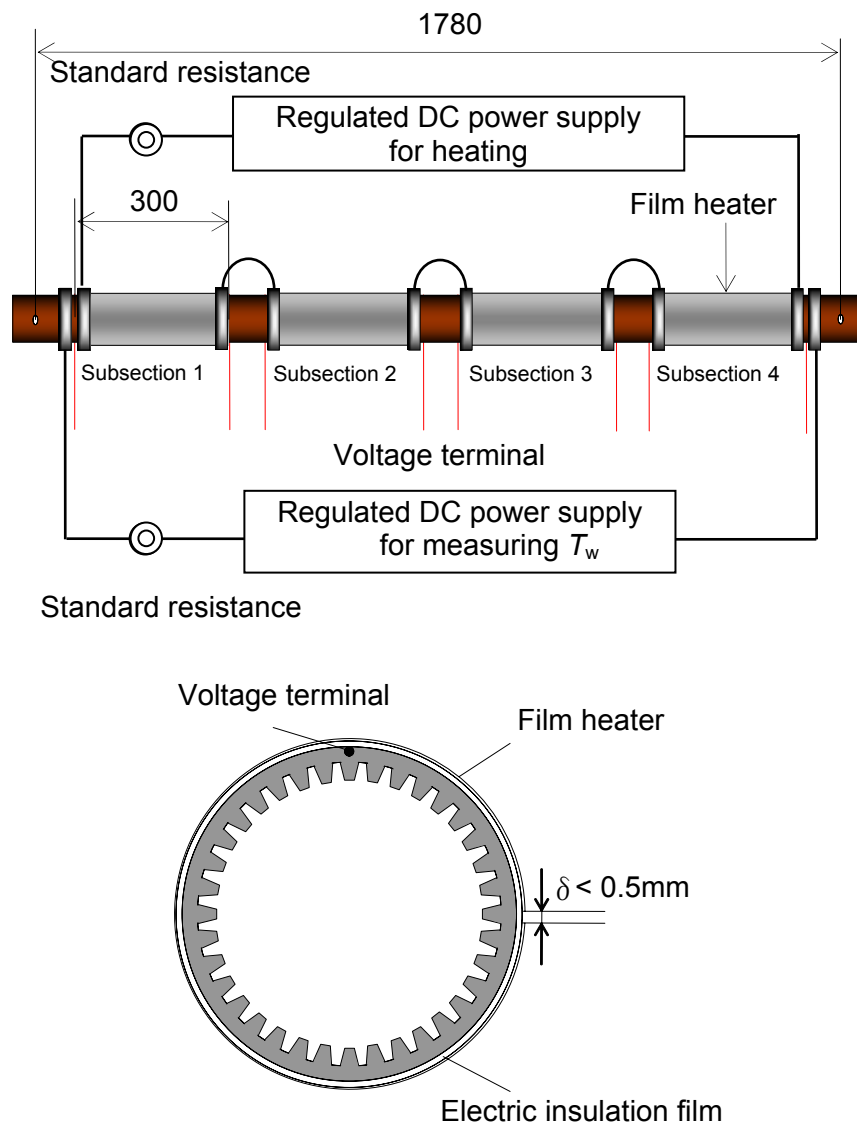

Figure 2: Details of the test section

Table 1: Dimensions of the micro-fin tube

Mean inner diameter, mm	5.43
Number of fins, -	55
Fin height, mm	0.161
Fin tip angle, deg	11.9
Helix angle, deg	14.3
Area enhancement ratio, -	1.86

Table 2: Experimental conditions

Mass velocity, kg/(m^{2}s)	100, 200, 300
Saturation temperature, ˚C	0, 10
Heat flux, kW/m^2	10, 20, 30
Inlet vapor quality, -	0.127~0.90
Oil circulation ratio, mass%	0~1.02

3 DATA REDUCTION

The heat transfer coefficient (α) of each subsection is defined as follows:

$$\alpha = \frac{q}{T_w - T_s} \tag{1}$$

where q is the heat flux calculated with the mean inner diameter of the micro-fin tube, and T_w and T_s are the wall temperature and the saturation temperature of the refrigerant, respectively. Regardless of the existence of PAG oil, no remarkable changes in saturation temperature (bubble point temperature) due to the existence of the oil have been measured. Therefore, the saturation temperature of pure CO_2 is used in equation (1) even in the case of CO_2-oil mixtures.

4 EXPERIMENTAL RESULTS

4.1 Results of pure CO_2

Figures 3(a) and (b) show the effect of mass velocity and heat flux on heat transfer coefficients at the saturation temperature of 0°C in the case of pure CO_2. As shown in Figure 3(a), the heat transfer coefficients are almost independent of mass velocity, although the heat transfer coefficients for mass velocity G of 300 kg/m^{2}s are higher than those in other conditions when quality x >0.7. On the other hand, as shown in Figure 3(b), the heat transfer coefficients increase significantly with the increase in heat flux. And the heat transfer coefficients are also almost independent of vapor quality. From the observations above, it is considered that the flow boiling heat transfer is dominated by nucleate boiling. And it is found that the dryout inception quality increases with the increase in mass velocity, while it is independent of heat flux.

Figures 4(a) and (b) show the effect of mass velocity and heat flux on heat transfer coefficients at the saturation temperature of 10 °C in the case of pure CO_2. As shown in Figure 4(a), except for the conditions when G = 100 kg/m^{2}s and x > 0.5, there is almost no influence of mass velocity on heat transfer coefficient. When G = 100 kg/ m^{2}s and x > 0.5, the heat transfer coefficients are lower than those at other conditions. The decrease is considered due to the change in flow pattern. That is, it is considered that this decrease is due to the partial dryout at the top of the tube. And the heat transfer coefficients decrease with the increase in vapor quality, while they increase when G = 300 kg/ m^{2}s and x > 0.8. This is considered due to the influence of forced-convection evaporation. On the other hand, as shown in Figure 4(b), the heat transfer coefficients decrease with the increase in vapor quality, and they increase with the increase in heat flux at x < 0.7. However, the heat transfer coefficients are independent of heat flux at x > 0.7. This is considered due to nucleate boiling being suppressed by the oil film accumulated in the tube.
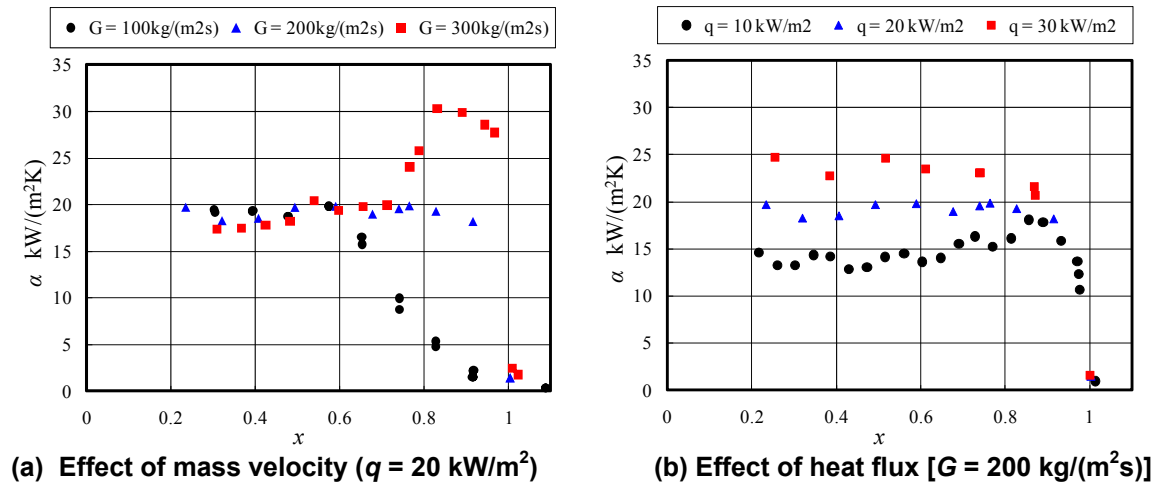

(a) Effect of mass velocity (q = 20 kW/m^2)   (b) Effect of heat flux [G = 200 kg/(m^{2}s)]

Figure 3: Effect of mass velocity and heat flux on heat transfer coefficients
(T_s = 0 °C, OCR = 0 mass%)

(a) Effect of mass velocity (q = 20 kW/m^2)   (b) Effect of heat flux [G = 200 kg/(m^{2}s)]
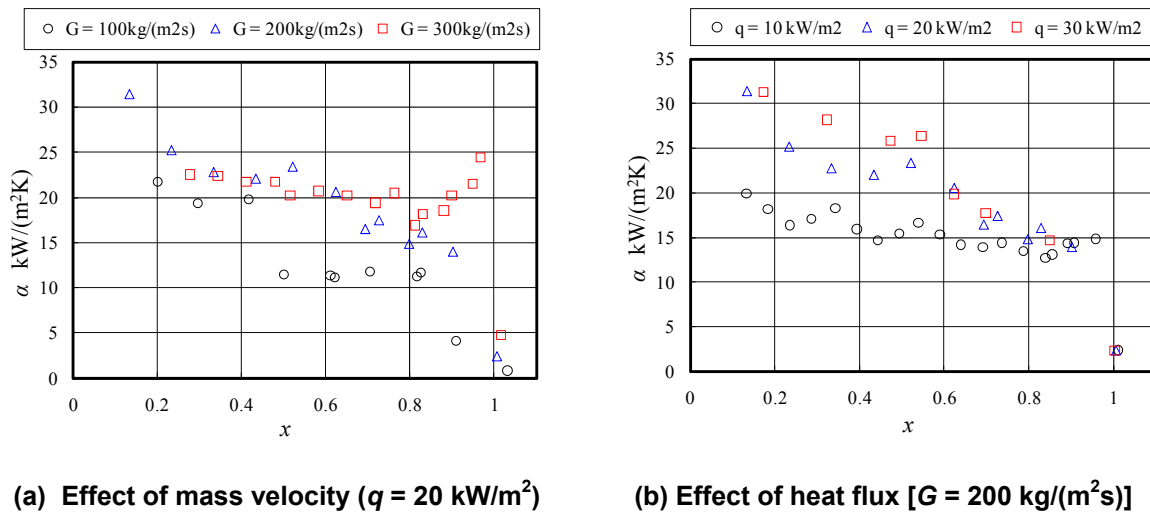
Figure 4: Effect of mass velocity and heat flux on heat transfer coefficients
(T_s = 10 °C, OCR = 0 mass%)

From the above results, at the saturation temperature of 10 ˚C, it is considered that the flow boiling heat transfer in the pre-dryout region is dominated by nucleate boiling. And it is found

that the dryout inception quality increases with the increase in mass velocity, while it is independent of heat flux.

By comparing Figures 4 and 3, it is found that the heat transfer coefficients increase slightly with increasing the saturation temperature. This is considered to be due to the number density of active cavities for nucleate boiling increasing with the decrease in surface tension, caused by increase in saturation temperature. It is also found that the dryout inception quality is almost independent of the saturation temperature.

4.2   Results of CO_2-oil mixtures

Figures 5(a) and (b) show the effect of mass velocity and heat flux on heat transfer coefficients at the saturation temperature of 10°C in the case of CO_2-oil mixtures. As shown in Figure 5(a), the heat transfer coefficients increase with mass velocity. And the heat transfer coefficients increase slightly with the increase in vapor quality, while they are almost independent of vapor quality at mass velocity G of 100 kg/m²s. On the other hand, as shown in Figure 3(b), the heat transfer coefficients are almost independent of heat flux, although the heat transfer coefficients for heat flux q of 30 kW/m² are higher than those of other conditions in the pre-dryout region. From the observations above, it is considered that the flow boiling heat transfer is dominated by forced convective boiling. And it is found that the dryout inception quality increases with the increase in mass velocity, while it is independent of heat flux, as same as with pure CO_2.

Figure 6(a) shows the flow visualization results of pure CO_2, and figures 6(b), (c) and (d) show those of CO_2-oil mixtures for vapor quality x = 0.3, 0.5 and 0.7. These photos were taken using a high-speed video camera with a flame speed of 500fps and a shutter speed of 1/20000. As shown in Figure 6(a) for pure CO_2, the CO_2 liquid is a colorless and transparent liquid, and the foaming phenomenon is not observed. On the other hand, as shown in Figures 6(b), (c) ad (d) for CO_2-oil mixtures, the CO_2 liquid shows cloudy with light yellow color, and the foaming phenomenon is hardly observed. But it is found that an oil film formed on the inner wall surface of the tube, and a lot of oil droplets are observed in the refrigerant liquid phase. With the increase in vapor quality, it is observed that the amount of the refrigerant liquid and the numbers of the oil droplets in the refrigerant liquid phase decrease, while the thickness of the oil film increases gradually. This result implies that the thermal resistance caused by the oil film should increase gradually with vapor quality if the oil film flows as a laminar flow. For this reason, the influence of the forced-convection evaporation
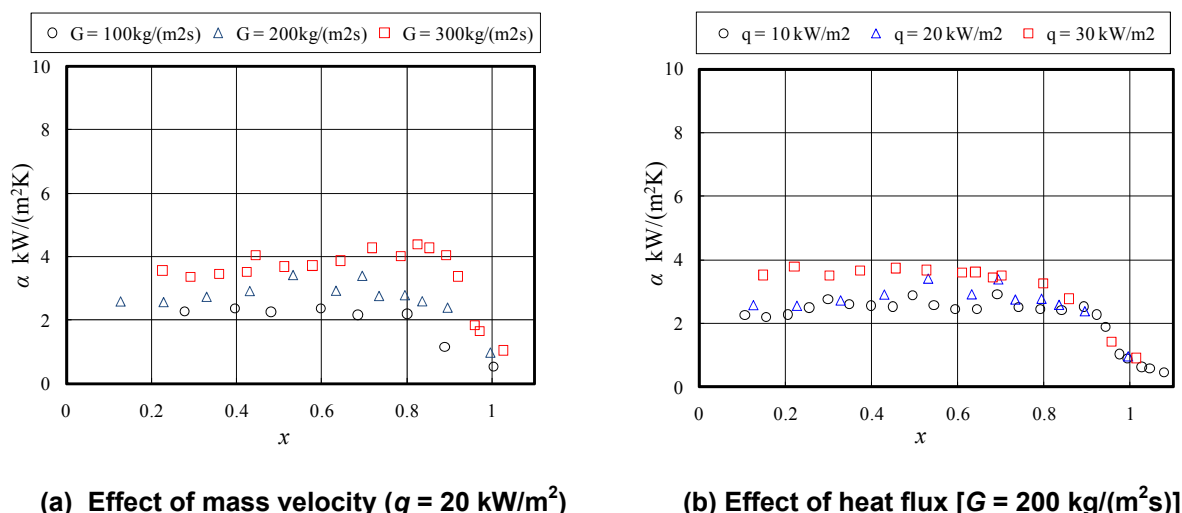

(a) Effect of mass velocity (q = 20 kW/m²)   (b) Effect of heat flux [G = 200 kg/(m²s)]

Figure 5: Effect of mass velocity and heat flux on heat transfer coefficients
(T_s = 10 °C, OCR = 1.0 mass%)

(a) x = 0.4, Pure CO_2 (OCR < 0.01 mass%)

(b) x = 0.3, OCR = 0.9 mass%

(c) x = 0.5, OCR = 0.9 mass%

(d) x = 0.7, OCR = 0.9 mass%

**Figure 6: The flow visualization results in the case of pure CO_2 and CO_2-oil mixtures
[T_S = 10°C, G = 200 kg/(m^{2}s)]**

by the increase in vapor quality on flow boiling heat transfer is reduced by the oil film, and then the heat transfer coefficient varies slightly with the increase in vapor quality.

5   CONCLUSIONS

Experiments on the flow boiling heat transfer coefficients of pure CO_2 and CO_2-oil mixtures inside a micro-fin copper tube were conducted at saturation temperatures of 0 and 10 ˚C using electrical resistance method, and the following results were obtained.
1. In the case of pure CO_2, regardless of the saturation temperature of 0 or 10˚C, the heat transfer coefficients are slightly influenced by mass velocity, while they are greatly dependent on heat flux. For that reason, it is considered that the flow boiling heat transfer in this case is dominant by nucleate boiling.
2. With increasing saturation temperature in the case of pure CO_2, heat transfer coefficients increase slightly. This is considered due to that the number density of active cavities for nucleate boiling increases with the decrease in the surface tension, which is caused by increasing the saturation temperature.
3. In the case of CO_2-oil mixtures, the heat transfer coefficients are slightly influenced by heat flux, while they are greatly dependent on mass velocity. It is considered that the flow boiling heat transfer in this case is dominant by forced convective evaporation.
4. Regardless the existence of the oil at both saturation temperatures, the dryout inception quality increases with the increase in mass velocity, while it is independent of heat flux.
5. From the flow visualization results, the foaming phenomenon was not observed, but it is found that an oil film was formed on the inner wall surface of the tube, and a lot of oil droplets are observed in the refrigerant liquid phase.
6. With the increase in vapor quality, it is observed that the amount of the refrigerant liquid and the numbers of the oil droplets in the refrigerant liquid phase decrease, while the thickness of the oil film increases gradually.

6 NOMENCLATURE

G	mass velocity	kg/(m^{2}s)
OCR	oil circulation ratio	mass%
T_s	saturation temperature	°C
T_w	wall temperature	°C
ΔT	temperature of super heated	°C
x	vapor quality	-
α	heat transfer coefficient	W/(m^{2}K)

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