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ABSTRACT

Flow boiling heat transfer characteristics of R744 with and without oil were investigated experimentally in horizontal smooth tubes with an inner diameter of 11.2 mm. In order to investigate the effect of the miscible oil on the heat transfer of R744, POE (polyolester) RENSIO C85E oil is added to give an oil circulation rate (OCR) between 0.5% and 4%. Results are compared with those of pure R744. The experimental conditions include evaporation temperatures of -15 and -30 °C, mass fluxes from 40 to 200 kg/m^2 s, heat fluxes from 2 to 10 kW/m^2, and vapor qualities from 0.1 to 0.8. The heat transfer coefficients of pure R744 show strong dependence on heat flux due to the dominance of the nucleate boiling. However, heat transfer also depends on mass flux and evaporation temperature due to enhanced convective boiling contribution at higher mass flux, 200 kg/m^{2}s, and change in thermo-physical properties**.** The pressure of oil generally deteriorates the heat transfer coefficient of pure R744. The reduction in heat transfer coefficient is most apparent at low qualities, 0.1 to 0.4, and at low mass fluxes, 40 kg/m^2. It is caused by the suppression of nucleate boiling due to increased surface tension. At conditions where the convective boiling contribution is dominant, qualities above 0.5 and mass fluxes above 100 kg/m^{2}s, oil increases heat transfer coefficients.

1. INTRODUCTION

With the growing concerns of the ozone depletion and global warming, R744 has been receiving great attention as an alternate environmentally-friendly refrigerant. Refrigeration systems using R744 as a working fluid are already commercialized on the market. Recent literature reviews by Thome and Ribatski (2005) and Zhao and Bansal (2006) reveal that many researchers experimentally investigated the flow boiling heat transfer of pure R744, especially at high evaporation temperatures from -4 to 25 °C. However, very limited studies have been conducted on oil-R744 mixtures. In typical refrigeration or air-conditioning systems, a small amount of the lubricating oil migrates from the compressor and through the system, which considerably affect the heat transfer performance of the components. Therefore, it is necessary to measure the flow boiling heat transfer coefficient of both pure R744 and oil-R744 mixtures and to understand the effect of oil on the heat transfer, especially at low temperature.

Zhao and Bansal (2006) also saw that most studies of heat transfer in oil-R744 mixtures were conducted at high evaporation temperature ranges, save one study by Hassan (2004). Comparing the previous studies on oil-R744 mixtures, the effect of oil concentration on the boiling heat transfer coefficient is inconsistent. Generally, the heat transfer coefficient of R744 was shown to decrease as OCR increases. The rate of deterioration varies according to the test conditions such as tube diameter, heat flux, mass flux, and evaporation temperature. The deterioration of heat transfer coefficient due to the presence of oil gets worse as evaporation temperature decreases. At low evaporation temperature of -30 ~ -10 °C by Hassan (2004) saw that the heat transfer coefficient decreases continuously as oil concentration increases from 0 to 4 %. However, the mass flux was limited to a small range (90~125 kg/m^{2}s), which makes it difficult to use the data for various system design.

The literature review reveals that the experimental studies on the flow boiling heat transfer of oil-R744 mixture are very limited and showed inconsistent results. In this paper, the flow boiling heat transfer coefficient of R744 with and without oil under various heat and mass fluxes at low evaporation temperatures is investigated.

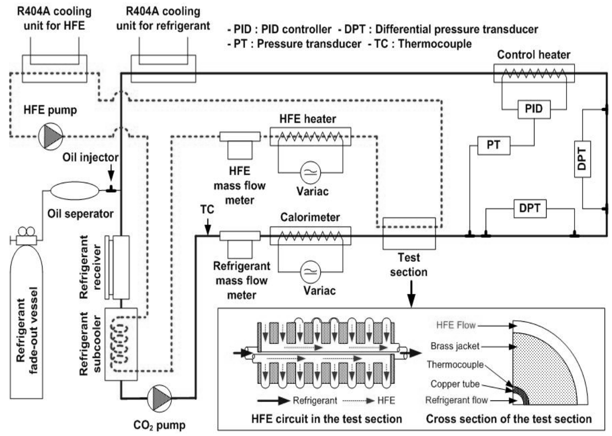

Fig. 1. Simplified schematic of the test facility

2. EXPERIMENTAL FACILITY AND TEST CONDITIONS

Fig. 1 shows a schematic of the experiment facility. The test facility consists of a refrigerant loop for testing, and a secondary coolant loop with HFE 7100 for controlling the test conditions. In the test loop, liquid refrigerant is pumped by the gear pump to the calorimeter, which heats it to the quality desired at the inlet of the test section. and visualization section is located after the test section. The control heater adds heat to maintain a desired saturation temperature in the test section. The refrigerant leaves the test section and condenses in a plate heat exchanger connected to the R404A cooling unit. The single–phase, secondary HFE 7100 loop subcools the refrigerant in the subcooler and then adds heat in the test section, as shown in Fig. 1.

Polyolester (POE) oil, RENISO C85E (density at 15 ^{o}C is 1004 kg/m^3 and viscosity at 40 ^{o}C is 80 mm^2/s), is used as the lubricating oil. It is completely miscible with R744 under operating evaporation temperatures and is widely used in industry. The mass of liquid R744 in the system is calculated by each test, and enough oil is added to achieve the desired OCR. The OCR is held constant during each experiment, and oil separator keeps oil from entering the fade-out vessel.

The test section consists of the test tube, brass jacket, and tube circuit for HFE as presented at Fig 1. The inside and outside diameters of the test tube are 11.2 and 15.8 mm, respectively. The copper tube has a heated length of 150 mm which is surrounded by a two-piece, cylindrical brass jacket. The secondary fluid, HFE, flows around this jacket, in order to provide a uniform temperature. The brass jacket unifies normally low temperature glide of the secondary fluid, HFE, used here for heating. All gaps between the two brass pieces and the test tube are filled with high thermal conductivity paste and they are held tight with metal clamps. to reduce the contact thermal resistance. Thermocouples are placed at the top, bottom, and sides at 3 locations along test section. The thermocouples were attached with solder in grooves carved on the tube surface. The remaining portion of the groves is filled with high thermal conductivity paste. The thermocouples are equally spaced along the axis of the test section at an 50 mm intervals starting 25 mm from the inlet of the heated section. As a result, the temperatures at 12 points on the tube surface are measured and their average values are used to calculate the heat transfer coefficient.

T-type thermocouples with a calibrated accuracy of ±0.10 °C are used to measure the refrigerant temperature and wall temperature of the test tube. The absolute pressure of R744 is determined by a pressure transducer with an uncertainty of ±3.4 kPa and pressure drops are evaluated by differential pressure transducers with the accuracy of ±0.086 kPa. The refrigerant mass flow rate is measured by a mass flow meter with an accuracy of ±0.1% of the reading. Electrical power inputs to the calorimeter and HFE heater are measured with watt transducers which have 0.2% reading accuracy.

Flow boiling heat transfer coefficients for R744 with and without oil are measured at a saturation temperature of -15 and -30 °C, mass flux of 40, 100, and 200 kg/m^2 , heat flux of 0.5,1, 2, 5, and 10kW/m^2 and qualities between 0.1 and 0.8.

3. DATA REDUCTION

In order to obtain an average heat transfer coefficient, the heat transfer rate to the refrigerant which is R744 in this study, $\dot{Q}_{ref}$, is determined as shown in the following equation.

$$\dot{Q}_{ref} = (\dot{m}C_P)_{HFE} \cdot (T_{HFE,i} - T_{HFE,o}) + \dot{Q}_{Amb} - \dot{Q}_{Cond} \tag{1}$$

$$h = \frac{\dot{Q}_{ref}}{A_{surf} \cdot (T_W - T_{CO_2})} \tag{2}$$

From Eq. (1) the heat transfer rate from secondary fluid was determined from the HFE specific heat, mass flow rate, and temperature difference between the inlet and outlet of the test section. The heat exchange rate with the environment, $\dot{Q}_{Amb}$, was obtained in a calibration experiment where an electrical heater was inserted in the test section while the power was carefully measured and presented as a function of the overall heat transfer coefficient of the test section and the log mean temperature difference between the HFE and the ambient air. The axially transferred conduction heat loss through the pipe, $\dot{Q}_{Cond}$, is estimated by a finite element code and experimentally checked. As presented in Eq. (2), the average heat transfer coefficient, *h*, is determined from the calculated heat transfer rate to refrigerant, $\dot{Q}_{ref}$, the measured average tube wall temperature, the test tube geometry, and the refrigerant saturation temperature calculated from the measured saturation pressure. Data regression and determination of refrigerant properties are performed using Engineering Equation Solver (2005).

The uncertainty of the heat transfer coefficient occurs due to the uncertainties of the independent measured parameters: temperature, pressure, mass flow, and electrical power input as presented earlier. The uncertainty propagation of heat transfer coefficient is evaluated based on Moffat (1988). The uncertainty is within the range of 8–20% of the measured heat transfer coefficients, and is shown as a vertical error bar in the figures.

4. RESULTS AND DISCUSSION

4.1 Flow Boiling Heat Transfer Coefficient of Pure R744

Fig. 2 and Fig. 3 represent the flow boiling heat transfer coefficient of pure R744 with respect to the varying heat and mass fluxes at the evaporation temperature of -15 ^{o}C and -30 ^{o}C, respectively. The effects of heat flux, mass flux, quality and the evaporation temperature are studied in the following paragraphs.

4.1.1 Effect of heat flux on pure R744: Fig. 2 and Fig. 3 show that the heat transfer coefficient increases at all qualities as heat flux increases from 0.5 kW/m^2 to 10 kW/m^2 at a given mass flux for both evaporation temperatures. Fig. 4 represents the averaged heat transfer coefficient over the quality range 0.1 to 0.8 according to the variation of the heat flux. In general, the flow boiling heat transfer coefficients are enhanced with a rise of heat flux due to the augmentation in the nucleate boiling. The results follow the general flow boiling heat transfer trend of R744 which has a strong dependence on the nucleate boiling.

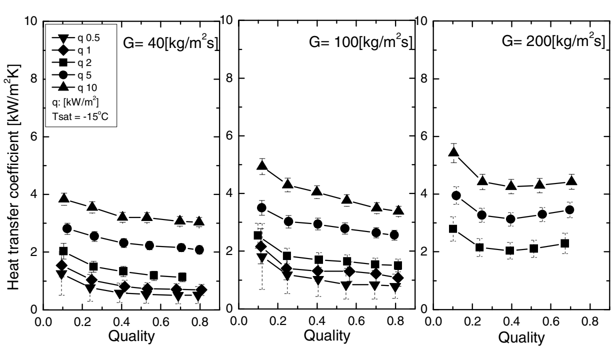

Fig. 2. Heat transfer coefficients with the change of mass and heat fluxes at an evaporation temperature of –15 °C

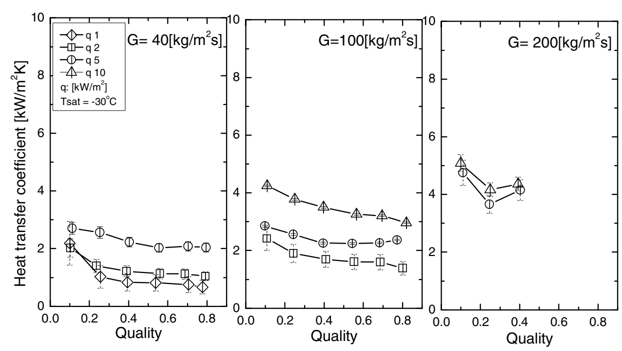

Fig. 3. Heat transfer coefficients with the change of mass and heat fluxes at an evaporation temperature of –30 °C

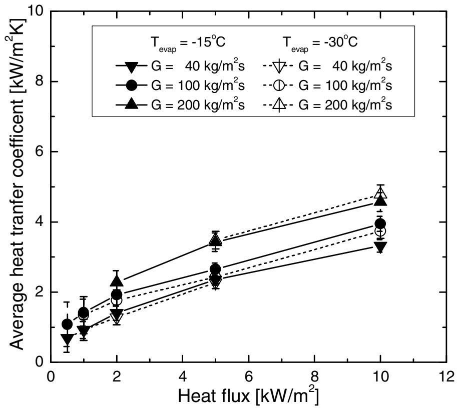

Fig. 4. Average heat transfer coefficients with respect to heat flux

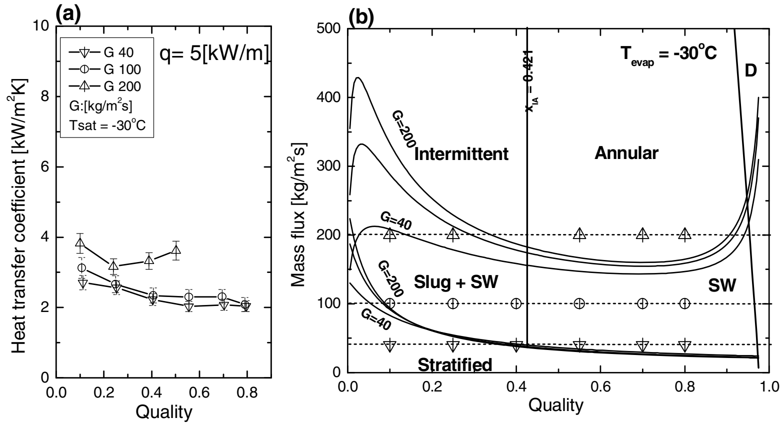

Fig. 5. (a) Heat transfer coefficients with respect to mass flux and quality and (b) Wojtan et al. flow pattern map

4.1.2 Effect of mass flux and quality on pure R744: Fig. 5(a) presents the flow boiling heat transfer coefficients with respect to the mass flux at the evaporation temperature of -30 ^{o}C. In general, the heat transfer coefficient increases with mass flux because of the increase I convective boiling. In addition, the relationship between the heat transfer coefficient and quality varies with respect to the mass flux. At lower mass fluxes, 40 and 100 kg/m^{2}s, the heat transfer coefficient decreases as the quality increases. However, at a higher mass flux, 200 kg/m^{2}s, the heat transfer coefficients decrease at low quality, but begin to increase when the quality reaches a critical value. This is because the convective boiling factor becomes more important with the increase in mass flux. This trend was also shown in a previous study presented by Park and Hrnjak (2005). Understanding the two-phase flow patterns is essential to analyzing the effect of mass flux on the heat transfer coefficients. In a previous study by Park and Hrnjak (2005) using the same facility, flow patterns were observed in 6.1 mm tubes, and results agreed with the Wojtan et al. (2005) flow map. It is believed that the flow patterns in the 11.2 mm tube are predicted by the flow pattern map corresponding to the current test conditions.

According to the flow pattern map at the evaporation temperature of -30 ^{o}C shown in Fig. 5(b), at low mass fluxes, 40 and 100 kg/m^{2}s, the flow pattern remains stratified throughout the experimental range of qualities. When the flow is stratified, the wetted area of the tube decreases at high qualities due to partial dryout, which results in a decrease in the heat transfer coefficient. As the nucleate boiling cannot be initiated on the top of the inner wall, where the liquid film is partially dry-out, it results in the decrease in the flow boiling heat transfer as the quality increases at low mass flux region. However, when the mass flux is increased to 200 kg/m^{2}s, the flow pattern becomes annular at high qualities, nucleate boiling occurs actively on the entire tube surface, convective boiling coefficients increase and the heat transfer coefficient increases as well.

These trends can be confirmed by examining the local heat transfer coefficient at each section of the tube, top, sides, and bottom. Fig. 6(a) and (b) show that the heat transfer coefficient at the evaporation temperature of -30 ^{o}C for the mass flux of 100 kg/m^{2}s and 200 kg/m^{2}s, respectively, and Fig. 8(a) and (b) show local one. It is shown in Fig. 8(a) that the local heat transfer coefficients around the periphery of the tube are distributed in the order of h_{bottom} > h_{side}> h_{top}, and the gap between them grows as the quality increases. It verifies the decrease in the heat transfer coefficients due to dryout on the top of the tube as the quality increases. At the mass flux of 200 kg/m^{2}s, however, the gap between the local heat transfer coefficients, Fig. 8(b), becomes negligible, since the entire inner wall is wetted by the liquid film and the flow boiling heat transfer occurs equally at each section of the wall.

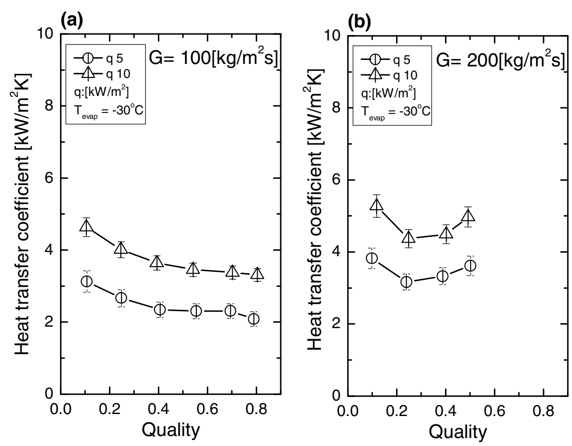

Fig. 6. Heat transfer coefficients at (a) G= 100 kg/m^{2}s and (b) G= 200 kg/m^{2}s

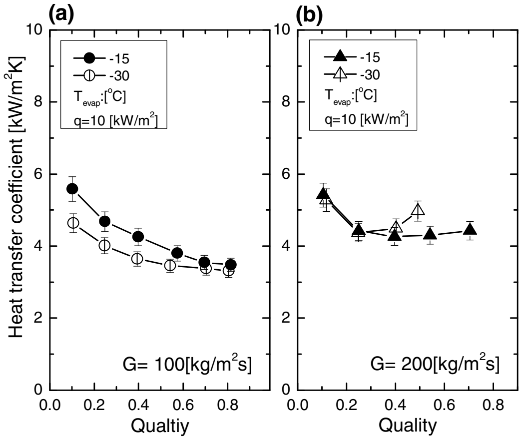

Fig. 7. Effect of evaporation temperature on heat transfer coefficients

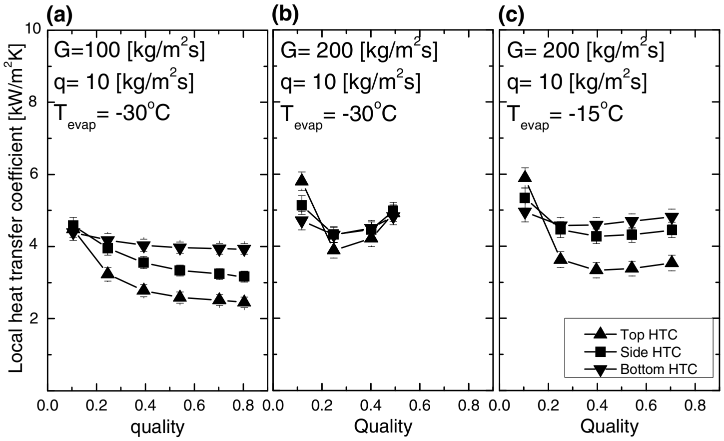

Fig. 8. Local heat transfer coefficients at top, sides, and bottom of the tube

4.1.3 Effect of evaporation temperature on pure R744: Fig. 7 shows the variation of the heat transfer coefficient due to evaporation temperatures -15 °C and -30 °C at various mass and heat fluxes. The overall values of the heat transfer coefficient at -30 °C are lower than those at -15 °C, because the nucleate boiling heat transfer is enhanced with increasing evaporation temperature due to an increase of reduced pressure by Gorenflo (1993) and a decrease of surface tension. The reduced pressures of R744 are 0.310 and 0.194 and the surface tensions are 7.10 and 10.08 at -15 °C and -30 °C, respectively. These trends were also observed in the previous studies by Park and Hrnjak (2005) and Cho et al. (2000). However, when the convective boiling heat transfer becomes important with an increase in the mass flux and the quality, the trend becomes reversed and the heat transfer coefficients at -15 °C becomes lower than those at -30 °C. The wall dryout at higher evaporation temperatures and high qualities causes the reduction of the heat transfer coefficient as described in the previous paragraph. This can be examined by comparing Fig. 8(b) and (c), which are the local heat transfer coefficients at -15 °C and -30 °C for the mass flux of 200 kg/m^{2}s. At the higher evaporation temperature of -15 °C, the local heat transfer coefficients at the top are much lower than those at the sides and bottom, which means that there is insufficient fluid on the top of the tube to initiate the nucleate boiling. The reduction of the wetted area is related to the decrease in the surface tension at higher temperature, since a lower surface tension enhances the nucleate boiling in the liquid film, and the entrainment of liquid droplets occurs actively through the mechanism of bubble bursting. This results in the increase in the partial dry region on the inner wall of the tube, as explained by Fujita and Ueda (1978).

4.2 Flow Boiling Heat Transfer Coefficient of Oil-R744 Mixtures

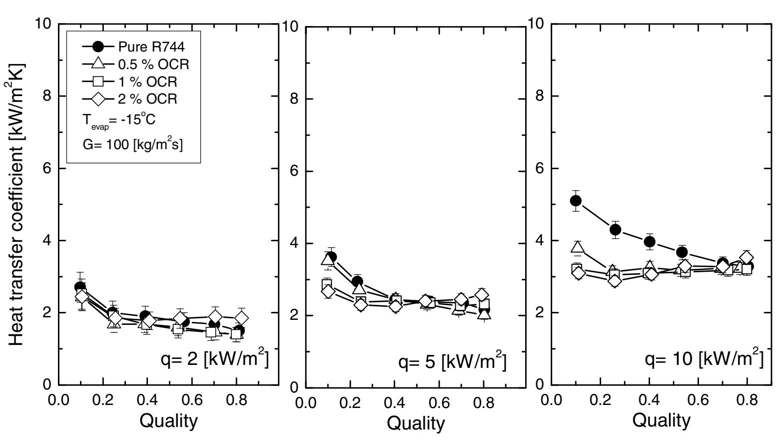

Fig. 9. Effects of oil on the heat transfer coefficient

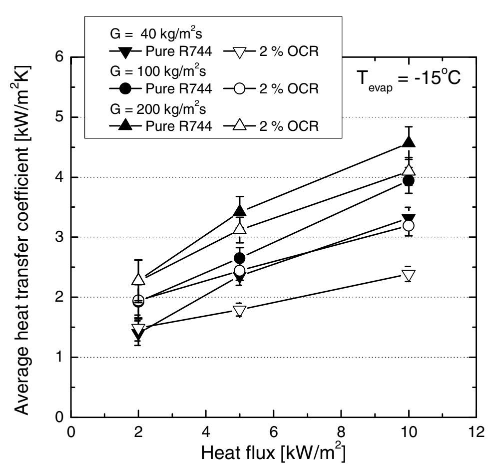

Fig. 10. Average heat transfer coefficients of pure R744 and oil-R744 mixture of OCR = 2 % with respect to heat flux

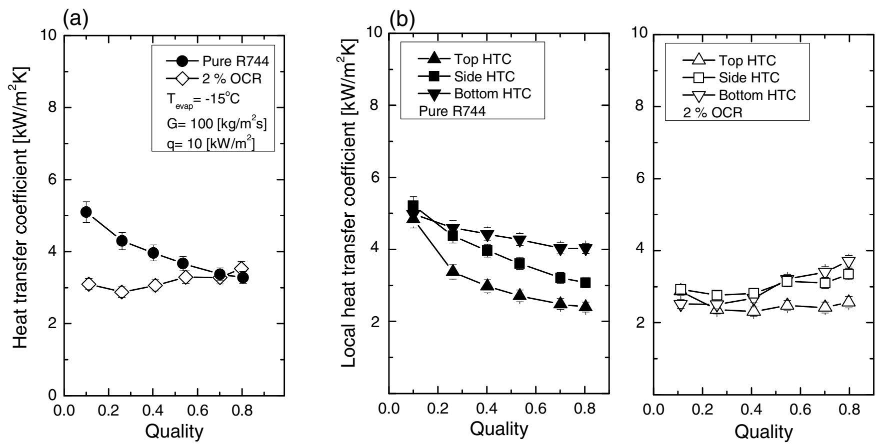

Fig. 11. (a) Heat transfer coefficients and (b) local heat transfer coefficient of pure R744 (left) and oil-R744 mixture of OCR = 2 % (right) at evaporation temperature of -15 °C for G = 100 kg/m2s, and q= 10 kW /m2

4.2.1 Negative effects of oil addition on the flow boiling heat transfer coefficient: Fig. 9 presents the flow boiling heat transfer coefficient of oil-R744 mixtures over the heat flux ranged from 2 to 10 kW/m^2 for a given mass flux of 100 kg/m^{2}s and the evaporation temperature of -15 ^{o}C. The results indicate that the flow boiling heat transfer coefficient decreases as the OCR increases from 0.5 % to 2 %. When the lubricating oil is added, the nucleate boiling is suppressed due to an increase of the surface tension by Seeton and Hrnjak (2006). That's why the tendency of the heat transfer deterioration with oil addition is more severe in the range where the nucleate boiling heat transfer is dominant, low qualities and high heat fluxes. In Fig. 10, the averaged flow boiling heat transfer coefficients of pure R744 and oil-R744 mixtures over the quality range are plotted with respect to the heat flux at various mass fluxes. All cases have the general trends that higher heat fluxes result in better heat transfer performance. However, the addition of oil significantly reduces this effect, especially at low mass fluxes, because nucleate boiling is dominant.

4.2.2 Positive effects of oil addition on the flow boiling heat transfer coefficient: As explained in the previous paragraph, the deterioration in the heat transfer due to oil addition is more significant at low qualities due to the suppression of the nucleate boiling. As shown in Fig. 11, however, the decrease in the flow boiling heat transfer coefficient is almost negligible at high qualities and even enhanced at the quality above 0.7. At the mass flux of 100 kg/m^{2}s, the flow boiling heat transfer coefficient of pure R744 decreases as quality increases, because the dry-out region on the top of the tube grows in the stratified flow as mention in section 4.1. When the oil is added, it is speculated that the higher surface tensions of oil-R744 mixtures tend to increase the wetted fraction of the tube wall. Therefore the decrease of the boiling heat transfer coefficient with increasing quality becomes less significant with oil addition. This hypothesis can be examined with the local heat transfer coefficients, as shown in Fig. 11. The local heat transfer coefficient of pure R744 at the top and sides is less than the bottom, which indicates dryout on the top of the tube. However, at 2 % OCR, the local heat transfer coefficient at the sides and the bottom are very close, while only the top has lower value. This agrees with the hypothesis that the wetted portion of the tube increases with oil addition.

5. SUMMARY AND CONCLUSIONS

The flow boiling heat transfer coefficients is experimentally investigated in the horizontal smooth tube of 11.2 mm inner diameter for R744 with and without oil at evaporation temperatures of -15 and -30 °C, mass fluxes from 40 to 200 kg /m^{2}s, and heat fluxes from 2 to 10 kW/m^2, and for vapor qualities ranging from 0.1 to 0.8. The effects of heat flux, mass flux, quality, and evaporation temperature on the flow boiling of heat transfer are presented and the changes in the heat transfer performance with oil addition are analyzed. The flow boiling heat transfer trend of pure R744 agree with previous studies. The strong dependence of nucleate boiling and the effect of mass flux are important to determining the flow pattern and corresponding heat transfer performance. The effect of lubricating oil is also investigated with an OCR of 0.5 to 2 %. When the nucleate boiling is dominant, oil addition degrades the heat transfer of two phase flow, due to the suppression of nucleate boiling. However, the increase in surface tension due to the oil was shown to increase heat transfer coefficient in certain cases, by reducing wall dryout. This shows that it is critical to examine all factors of oil addition before making generalized statements.

NOMENCLATURE

A	area (m^2)		*Subscripts*
C_P	specific heat (J/kg K)	Amb	ambience
G	mass flux (kg/m^2 s)	$Cond$	conduction
h	heat transfer coefficient (W/m^2 K)	HFE	secondary fluid, HFE
$\dot{m}$	mass flow rate (kg/s)	i	inlet
P_{red}	reduced pressure	l	liquid
q	heat flux (W/m^2)	o	outlet
$\dot{Q}$	heat transfer rate (W)	ref	refrigerant
T	temperature (°C or K)	sat	saturation
x	quality	W	wall

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