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Heat Transfer Enhancement of Falling Film Evaporation of HFO-1233zd(E) and HFC-134a on a Horizontal Tube by Thermal Spray Coating

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ABSTRACT

A falling film evaporator can reduce the amount of refrigerant compared with a flooded evaporator. Required functions for the heat transfer surface in falling liquid film evaporation are thin liquid film formation without breaking at low heat flux, nucleate boiling promotion in liquid film, and suppression of liquid entrainment at high heat flux. In this study, a porous thermal spray coating using copper as the coating material was made on a copper cylinder. The heat transfer performance of falling film evaporation and pool boiling was evaluated using HFO-1233zd(E) as the refrigerant, and the obtained results were compared with those for HFC-134a. The test cylinder was heated by a cartridge heater inserted at the center. Falling film evaporation experiments had been conducted with a film mass flow rate of 3.3×10^{-2} kg/(m·s), heat flux of 10 to 85 kW/m², and a saturation temperature at 20 ºC. The effects of the thermal spray coating, heat flux and thermo-physical properties of the refrigerants on heat transfer performance were investigated. The heat transfer coefficient increased with increasing heat flux. For the thermal spray coating, a large hysteresis effect according to the heating procedure with increasing or decreasing heat flux was observed in the characteristics of the heat transfer coefficient. The heat transfer enhancement factor by the thermal spray coating was up to 4.8. The value was higher than that for HFC-134a, especially under high heat flux condition. In the comparison between pool boiling and falling film evaporation heat transfer, falling film produced higher heat transfer coefficients for the thermal spray coating while the heat transfer on the smooth surface deteriorated due to partial dryout. The fine porous structure enhanced liquid spreading by the capillary force and evaporation from the liquid film surface by vapor bubble agitation.

1. INTRODUCTION

As most of widely used HFC refrigerants have a high global warming potential (GWP), reduction of the charged amount and a shift to low GWP refrigerants are required. Falling film evaporators are expected as the substitute of typical flooded evaporators because they can work with less refrigerant. An important issue for high evaporation heat transfer performance is to form and keep a thin liquid film on the heat transfer surface under low heat flux conditions. In addition, thicker liquid film is required to avoid dryout under high heat flux conditions. The thicker liquid film leads to a rise of wall superheat which causes nucleate boiling in the liquid film. Nucleate boiling may lead to a decrease of the liquid film flow rate due to entrainment and then dry patches will appear. One of the solutions for these problems is to construct fine structures on the heat transfer surface. Grooves and micro channels had been applied to promote the spreading of the liquid film and suppress the entrainment. In addition, the cavities will activate nucleation sites. Several methods are applied to create such a structured surface, for example machined process, sintered metal, and thermal spraying. This study focused on thermal spraying because it is a relatively simple process and can be easily adapted to hard machining materials such as stainless steel. Ubara *et al.* (2020) investigated the heat transfer enhancement effect of thermal spraying surface on falling film evaporation using HFC-

134a at a saturation temperature of 20 ºC. The results showed that the thermal spraying surface resulted in heat transfer up to 5.2 times higher than that of a smooth surface.

Heat transfer performance depends not only on the structure of the heat transfer surface but also on thermophysical properties of refrigerants. Zhao *et al.* (2017) experimentally evaluated the falling film evaporation heat transfer coefficient using HFC-134a and HCFC-123 at a saturation temperature of 6 ºC on mechanically processed heat transfer tubes. It was shown that HFC-134a provided around 2 to 3 times of heat transfer coefficients of HCFC-123 for each tube because of a less surface tension and higher working pressure. HFO-1233zd(E) is one of the expected new generation refrigerants with quite low GWP and its heat transfer performance is attracting attention. Nagata *et al.* (2016) investigated pool boiling and condensation heat transfer around a horizontal copper tube using several low-GWP refrigerants including HFO-1233zd(E). Pool boiling heat transfer coefficients of HFO-1233zd(E) were lower than HFC-134a at the same saturation temperature of 10 ºC and 40 ºC. However, there are few studies about the falling film evaporation on a horizontal tube, targeting the condition in a shell and tube evaporator.

In this study, a porous coating processed by thermal spraying of copper was applied to the heat transfer surface. The heat transfer performance of falling film evaporation and pool boiling was evaluated using two refrigerants; HFO-1233zd(E) and HFC-134a. The pool boiling experiments were also conducted to confirm the wall superheat for nucleate boiling. The heat transfer coefficients were evaluated at a film mass flow rate of 3.3×10^{-2} kg/(m·s), a heat flux range of 10 to 85 kW/m^2 and a saturation temperature at 20 ºC. The effects of the configuration of the heating surface, heat flux and refrigerant properties on heat transfer performance are discussed.

2. EXPERIMENTAL APPARATUS AND METHOD

2.1 Experimental apparatus

Figure 1 shows a schematic diagram of the experimental apparatus. The apparatus consisted of a pressure vessel, a refrigerants circulation loop, a condenser, and a test section. The pressure vessel was first well evacuated and then filled with the working fluid. Falling film evaporation and pool boiling experiences could be carried out by changing the liquid level. In the falling film experiment, the liquid level was set to below the test section. The refrigerant liquid was circulated by a gear pump and supplied to three needle nozzles with an inner diameter of 1.48 mm. The nozzles were fixed at the bottom of a horizontal copper tube above the test tube. The mass flow rate was measured by a Coriolis mass flow meter. To supply liquid to the test tube with an axially homogeneous distribution, liquid from the nozzles was supplied via a dummy tube with a mechanically processed surface. It was confirmed from the flow observation at the bottom of the dummy tube that liquid spread well in the axial direction and then flowed to the test section immediately below the dummy tube. The supplied liquid refrigerant temperature was measured by K-type thermocouples at the inlet of the nozzles. Refrigerant temperature in the pressure vessel was also measured by K-type thermocouples at 65 mm above and 35 mm below the test section. The refrigerant vapor was condensed in a double tube heat exchanger connected to the upper part of the pressure vessel, and the condensed liquid returned to the test section owing to gravity. The vapor pressure was maintained by the temperature and flow rate of the cooling water. In the pool boiling experiments, the nozzles and the dummy tube were removed, and the test section was submerged into the liquid pool.

Figure 2 shows details of the test section. The test section was a copper tube with an outer diameter of 19.05 mm and a length of 50 mm; it was heated by a cartridge heater inserted along the center axis. The heating length of the heater was equal to the length of the test section, and the input power was measured by a wattmeter. The outer wall temperature of the test tube was measured by T-type thermocouples placed at the bottom to avoid disturbances to the falling liquid film. A constantan wire with a diameter of 0.1 mm was soldered on the surface of the tube at each measurement point. In this study, a thermal spray coating was applied to the heat transfer surface for heat transfer enhancement. Copper (Cu 99.99 % Oxygen-free copper) was sprayed by an arc wire spraying process on the sandblasted surface of the test tube. Microphotographs of the thermal spray coating are shown in Figure 3. The top view showed the rough surface formed by the sprayed molten drops. The average surface roughness measured by a laser microscope was 20.6 μm. In the cross-sectional view, the bright orange area was the base material, and the brown area on the base material was the coating layer. Microscale cavities were observed on the inner and outer surfaces of the coating.
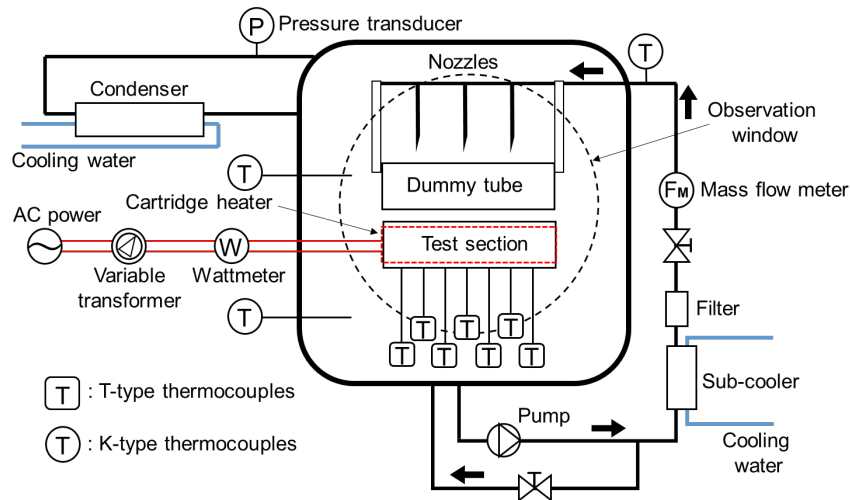
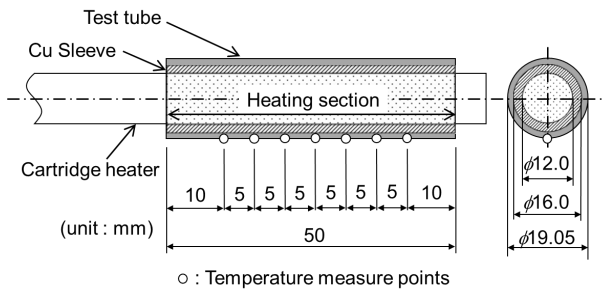**Figure 1:** Schematic diagram of the experimental apparatus.

Figure 2: Details of the test section.

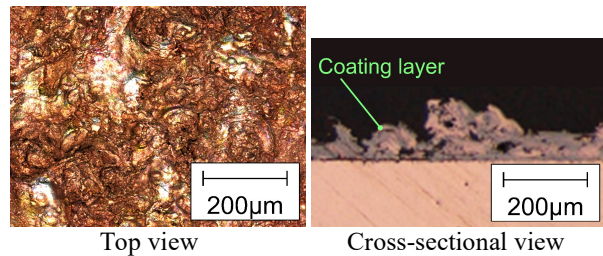
Top view   Cross-sectional view
Figure 3: Microphotographs of the sprayed surface.

The heat transfer coefficient was measured using HFO-1233zd(E) and HFC-134a as the working fluid at a saturation temperature of 20 °C with a set pressure of 0.11 MPa and 0.57 MPa, respectively. The liquid mass flow rate per unit length was varied from 8.6×10^{-3} to 6.3×10^{-2} kg/(m·s), and the heat flux was varied from 10 to 85 kW/m². The subcooling degree of the supplied liquid was maintained within 0.5 K in the falling film experiments. The evaporation and boiling behaviors on the test section were observed from the horizontal direction during the experiments.

2.2 Data reduction
Liquid mass flow rate on the one side per unit length, Γ, is defined by the following equation:

$$\Gamma = \frac{m}{2L} \tag{1}$$

where m is the measured mass flow rate, and L is the heating length. Heat flux, q, is calculated by the following equation:

$$q = \frac{Q}{A} = \frac{Q}{\pi D L} \tag{2}$$

where Q is the heat input, A is the heat transfer area, and D is the outer diameter of the test tube. For the coated tube, the heat flux was defined using the diameter at the bottom of the coating. The heat transfer coefficient, α, is defined by the following equation:

$$\alpha = \frac{q}{T_{wall} - T_{sat}} \qquad (3)$$

where T_{wall} is the measured surface temperature, and T_{sat} is the saturation temperature calculated from the measured pressure using REFPROP version 10.0 (Lemmon et al., 2018). T_{wall} is obtained by the trimmed average of the seven measured wall temperatures. The maximum and the minimum value of the heat transfer coefficient were plotted as the error bars.

3. RESULTS AND DISCUSSION

3.1 Effect of liquid flow rate
Figure 4 shows the heat transfer coefficient against liquid mass flow rate per unit length. Half-filled symbols are the results for HFO-1233zd(E) and filled symbols are those for HFC-134a. Open symbols are previous reported results for a smooth tube by Ribatski and Thome (2007) using HFC-134a.

For the smooth tube using HFO-1233zd(E), the heat transfer coefficient increased with increasing the liquid film flow rate mainly because of the expanding of wetted area. At the heat flux of 50.1 kW/m^2 and $\Gamma = 8.5 \times 10^{-3}$ kg/(m·s), effective data could not be obtained as the whole surface was dried out and the surface temperature rose extremely high. At the heat flux of 16.7 kW/m^2, the heat transfer coefficient increased with increasing the liquid film flow rate over the experimental range. As nucleate boiling was not active at low heat flux, the effect of forced convection might be dominant. At the higher heat fluxes of 33.4 and 50.1 kW/m^2, heat transfer coefficients seemed to be saturated to flow rate changes when $\Gamma > 3.0 \times 10^{-2}$ kg/(m·s). Nucleate boiling heat transfer in the liquid film was dominant. Besides, it is expected that bubbles increase the volumetric flow rate of the liquid film and promote spreading of the liquid film.

For the thermal spray coated tube, heat transfer coefficients were higher than those for the smooth tube. At the heat flux of 16.7 kW/m^2, the values were almost constant over the experimental range, because nucleation sites could be activated on the thermal spray coating, and the heat transfer was dominated by nucleate boiling. At 33.4 kW/m^2, the heat transfer coefficient increased below $\Gamma = 1.7 \times 10^{-2}$ kg/(m·s) and became constant above that. At 50.1 kW/m^2, the surface was fully dryout when $\Gamma = 8.5 \times 10^{-3}$ kg/(m·s), then the heat transfer coefficient increased rapidly at $\Gamma = 2.8 \times 10^{-2}$ kg/(m·s), and over that, it became saturated. For each heat flux, there was the minimum Γ keeping the heat transfer coefficient dominated by nucleate boiling in the liquid film. The coated surface made the minimum flow rate lower than the smooth surface. Microstructures on the surface enhanced liquid spreading and rewetting, so it prevented the formation of dry patches leading to heat transfer deterioration.
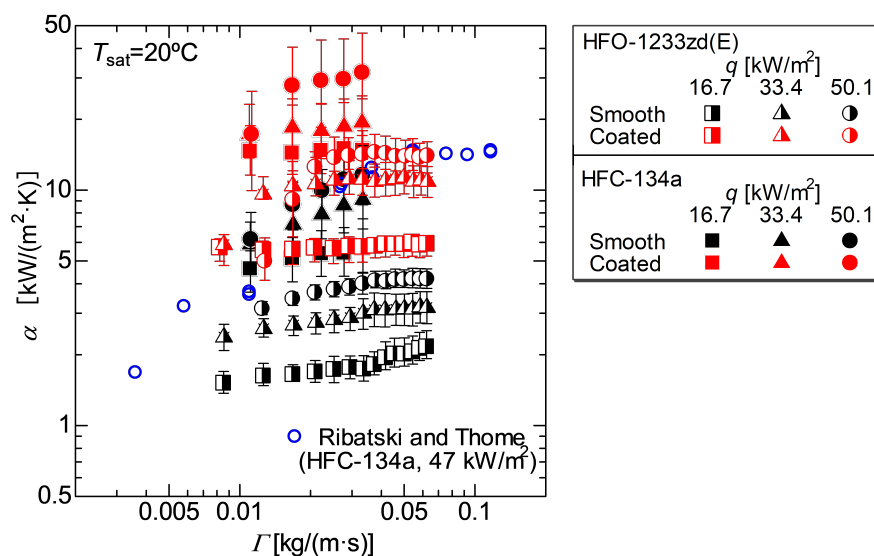

Figure 4: Heat transfer coefficient against film mass flow rate.

Compared with the results for HFC-134a, the heat transfer coefficient of HFO-1233zd(E) got lower under the same conditions. The main reason might be the difference in nucleate boiling heat transfer characteristics which strongly depend on refrigerant properties such as surface tension and vapor density. HFO-1233zd(E) would need higher wall superheat for the onset of nucleate boiling due to its larger surface tension. Several similar tendencies in the effect of liquid film flow rate on falling film evaporation heat transfer could be observed. The heat transfer coefficient increased up to a certain flow rate and then became saturated. As the heat flux increased, the gradient of the heat transfer coefficient against liquid film flow rate increased. In addition, on the thermal spray coated surface, the heat transfer coefficient became saturated under lower heat flux conditions than the smooth surface did.

3.2 Effect of heating procedure

Figure 5 shows falling film heat transfer coefficients against heat flux. Filled symbols are the results for the heat flux increasing process and open symbols are those for the decreasing process. Liquid film mass flow rate was set to 3.3×10^{-2} kg/(m·s). The broken and chain lines show the correlation of pool boiling heat transfer around a smooth horizontal tube proposed by Jung et al. (2003) for HFC-134a and HFO-1233zd(E), respectively.

It can be seen from the gradient of the curves that the heat transfer was dominated by nucleate boiling in the liquid film because the film flow rate was enough to cover the heat transfer surface and the gradient was well agreed with that of pool boiling correlation. However, for heat flux values above 50.0 kW/m^2, a deterioration in the heat transfer coefficient was observed only for HFO-1233zd(E). The deterioration might be caused by dry patches formation. In the heat flux decreasing process, the heat transfer coefficient was slightly lower than that of the increasing process. Since the advancing contact angle for rewetting is larger than the retreat contact angle, once the dry patch had formed in high heat flux condition, it remained and the heat transfer coefficient might deteriorate in the decreasing process.

The thermal spray coating showed higher heat transfer coefficients than the smooth tube over the experimental heat flux range. The heat transfer coefficient monotonically increased with increasing heat flux. The deterioration of the heat transfer coefficient at high heat flux as seen in the smooth tube did not appear. This might be because of the high wettability of the rough structure. In the heat flux decreasing process, the heat transfer coefficient was improved at heat flux of 66.2 kW/m^2, and then it decreased with the same gradient with the smooth tube. In contrast to the smooth tube, the heat transfer coefficients were higher in the decreasing process. Since many cavities were activated at the highest heat flux condition, nucleation site density was higher in the decreasing process.

Compared with the results for HFC-134a, for the smooth tube, deterioration in the heat transfer coefficient at high heat flux appeared only for HFO-1233zd(E). As the surface tension of HFO-1233zd(E) is 1.7 times larger than HFC-134a, the spreading of the liquid film of HFO-1233zd(E) might be weak. Therefore, HFC-134a produced higher heat transfer coefficients over the experimental range of heat flux values. The difference in heat transfer coefficients between two refrigerants were agreed with the Jung's correlation on the pool boiling heat transfer coefficient, so the difference in heat transfer performance might be caused by the difference in nucleation site density. For the thermal spray coated surface, HFC-134a also showed a higher heat transfer coefficient as shown for the smooth tube, but the tendency was different. HFO-1233zd(E) had a larger hysteresis effect in the heat transfer coefficient, and the gradient of the curve for HFC-134a was smaller than HFO-1233zd(E). It was considered that most of the cavities on the thermal spray coating were activated at low heat flux in HFC-134a, while HFO-1233zd(E) required a higher wall superheat due to the higher surface tension.

Figure 6 shows the heat transfer enhancement factor by the thermal spray coating; the ratio of the heat transfer coefficient of the thermal spray coated tube to the smooth tube. Under the high heat flux condition, larger enhancement factor could be obtained by using HFO-1233zd(E). The highest value was 4.8 at the heat flux of 10.0 and 66.2 kW/m^2. Larger enhancement factor at the low heat flux in the decreasing process might be due to the higher nucleation site density on the porous surface. In contrast, the enhancement at the high heat flux might be due to the suppression of dry patch formation. Because it was shown for the smooth surface that the deterioration in the heat transfer coefficient assuming to be due to dry patch formation was observed at the high heat flux only for HFO-1233zd(E), the promotion of the spreading of liquid film by the rough structure of the thermal spray coating might be more effective for HFO-1233zd(E) than HFC-134a.
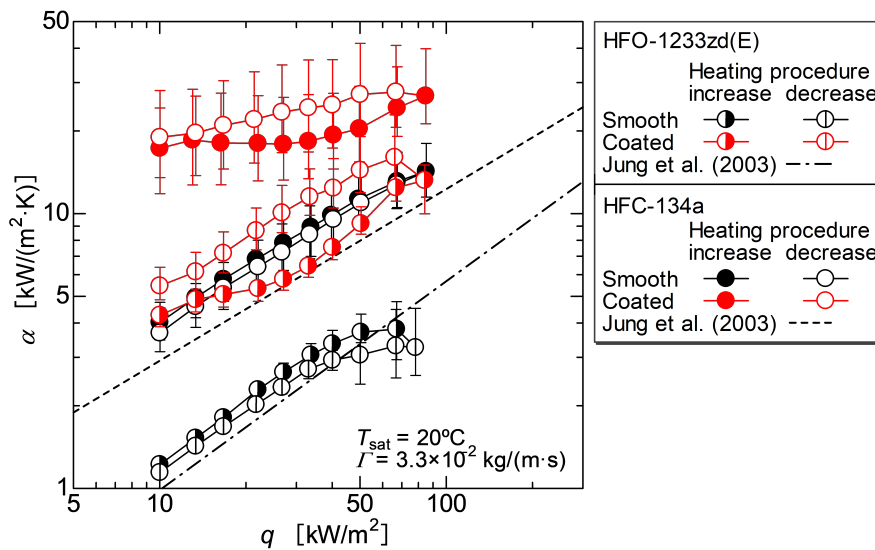
Figure 5: Falling film evaporation heat transfer coefficient of HFO-1233zd(E) and HFC-134a.
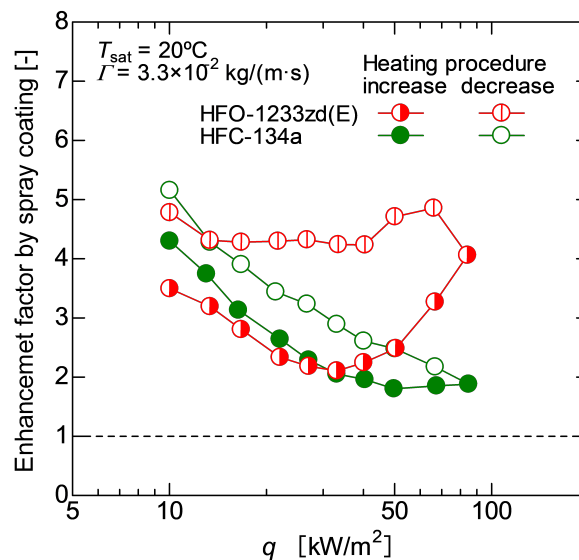

Figure 6: Enhancement factor of falling film evaporation heat transfer by thermal spray coating.

3.3 Pool boiling heat transfer

Figures 7 (a) and (b) show the heat transfer coefficients of falling film evaporation and pool boiling for smooth and spraying surfaces, respectively. Heat transfer coefficients of falling film evaporation are the same with those in Figure 5.

In Figure 7 (a), the gradient of the pool boiling heat transfer coefficient against heat flux was constant and agreed with that of Jung's correlation. There was almost no difference between the heat flux increasing process and decreasing process. Compared with falling film evaporation, the heat transfer coefficient in pool boiling was slightly higher, and the deterioration of heat transfer coefficient under the high heat flux condition did not appear. The both heat transfer modes were dominated by nucleate boiling, but the dry patches leading to the heat transfer deterioration were formed at the high heat flux in the falling film evaporation.

For the coated surface in Figure 7 (b), the heat transfer coefficient was almost the same in both heating procedures. A hysteresis effect of the heating procedure appeared only in the falling film evaporation experiments. In falling
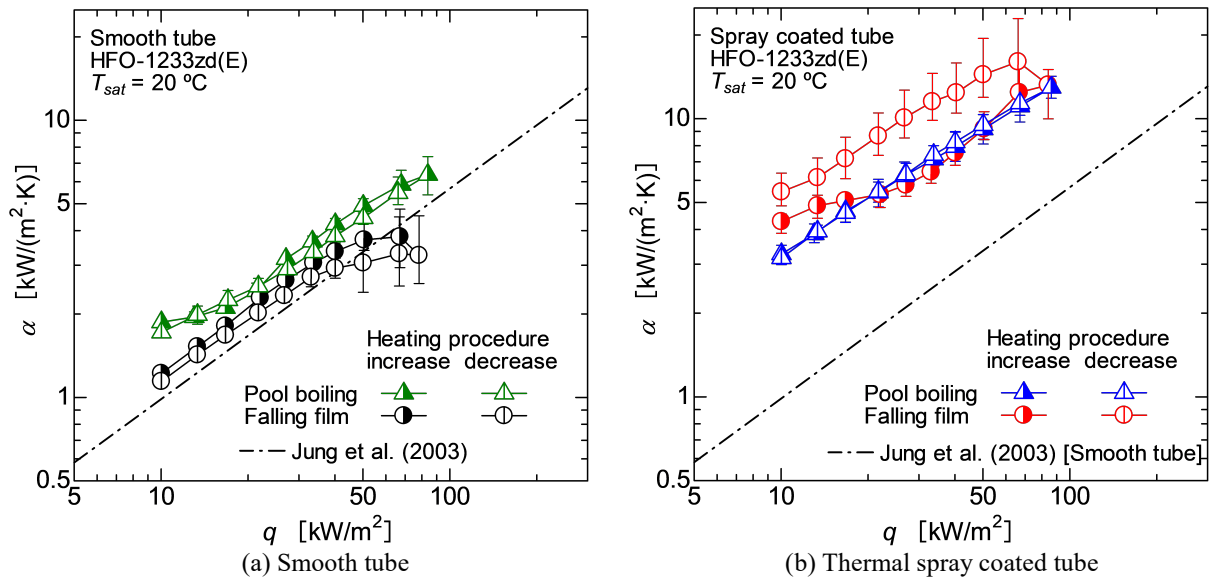
(a) Smooth tube   (b) Thermal spray coated tube

Figure 7: Comparison of heat transfer coefficient between falling film evaporation and pool boiling.

film evaporation, the increase in the wall superheat at the high heat flux might activate nucleation sites before the decreasing process. In the heat flux increasing process, falling film evaporation performed a better heat transfer than the pool boiling did below the heat flux of 16.7 kW/m². Evaporation on the liquid film might be dominant at the low heat flux because of weak nucleate boiling.

4. CONCLUSIONS

The falling film evaporation heat transfer on a horizontal cylinder was experimentally investigated using HFO-1233zd(E). The effect of micro structures by the thermal spray coating on the heat transfer coefficient were evaluated under various liquid film flow rate and heat flux conditions. Moreover, to evaluate the role of nucleate boiling in the liquid film, the heat transfer coefficients were compared with those of pool boiling heat transfer using the same apparatus. The results were also compared with those for HFC-134a to discuss on the effect of the thermo-physical properties of refrigerants. The main conclusions are summarized as follows:

- Microstructures on the thermal spray coating enhanced spreading of liquid film and rewetting, so they prevented dry patch formations leading to a heat transfer deterioration.
- At the constant liquid film flow rate of 3.3×10^{-2} kg/(m·s), which was enough to form liquid film over the heating surface, the coated surface performed higher heat transfer over the experimental heat flux range. Hysteresis effect depending on the heating process appeared only on the coated surface.
- By applying the thermal spray coating, the heat transfer could be successfully enhanced for HFO-1233zd(E) up to 4.8 times compared to the smooth surface. The enhancement at the low heat flux was due to the activation of nucleation site in the liquid film and that at the high heat flux was due to the suppression of the dry patch formation.
- Although the heat transfer coefficient of HFO-1233zd(E) was lower than HFC-134a at the same saturation temperature of 20 ºC because of the lower nucleation site density, the enhancement factor by the thermal spray coating was larger for HFO-1233zd(E).

NOMENCLATURE

A	heat transfer area	(m²)
D	outer diameter	(m)
L	heating length	(m)
m	mass flow rate	(kg/s)
Q	heating rate	(W)
q	heat flux	(W/m²)

T	temperature	(K)
α	heat transfer coefficient	(W/(m²·K))
Γ	mass flow rate of liquid film	(kg/(m·s))

Subscript

wall	heat transfer surface
sat	saturated condition

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