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International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2008

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ABSTRACT

In the present study, experiments were performed to examine characteristics of flow boiling heat transfer and pressure drop of a refrigerant R410A flowing in a vertical copper smooth tube with 1.0 mm inside diameter. Local heat transfer coefficients were measured in a range of mass fluxes from 30 to 200 kg/(m^2·s), heat fluxes from 1 to 16 kW/m^2 and qualities from 0 to 1 at evaporation temperature of 10°C. Pressure drops were also measured at mass fluxes of 100 and 200 kg/(m^2·s) for upward flow and of 30, 50, 100 and 200 kg/(m^2·s) for downward flow and qualities from 0.1 to 0.9. Three types of flow pattern were observed in the tube: A slug, a slug-annular and an annular flow. Based on the measurements, the characteristics of frictional pressure drop, heat transfer coefficient and dryout qualities were clarified.

1. INTRODUCTION

Recently, as one of new high performance heat exchangers for air conditioning systems, a finless heat exchanger using small diameter tubes arranged vertically is developed. It is needed that the characteristics of frictional pressure drop, heat transfer coefficient and dryout quality can be predicted accurately to design the heat exchanger using small diameter tubes. The characteristics in a small tube might be different from those in a traditional diameter tube because surface tension becomes an important factor in a small tube. However, published data relating boiling heat transfer and two-phase pressure drop in small tubes are very few, compared to data for traditional large tubes.

For two-phase frictional pressure drop, including the studies by Lockhart-Martinelli (1949) and Chisholm (1967), there were some studies for a vertical small tube. Lazarek-Black (1982) performed an experiment of CFC113 flowing vertically upward and downward in a stainless steel tube of 3.15mm inside diameter with a 180° bend. The measured frictional pressure drop was correlated well by Lockhart-Martinelli correlation using the Chisholm parameter of 30. Mishima-Hibiki (1995) carried out an experiment of water-air flowing vertically upward in small diameter tubes with 1.09-3.90 mm inside diameter. They suggested a new correlation to Chisholm parameter used in the Lockhart-Martinelli correlation, based on their and other researchers' experimental results.

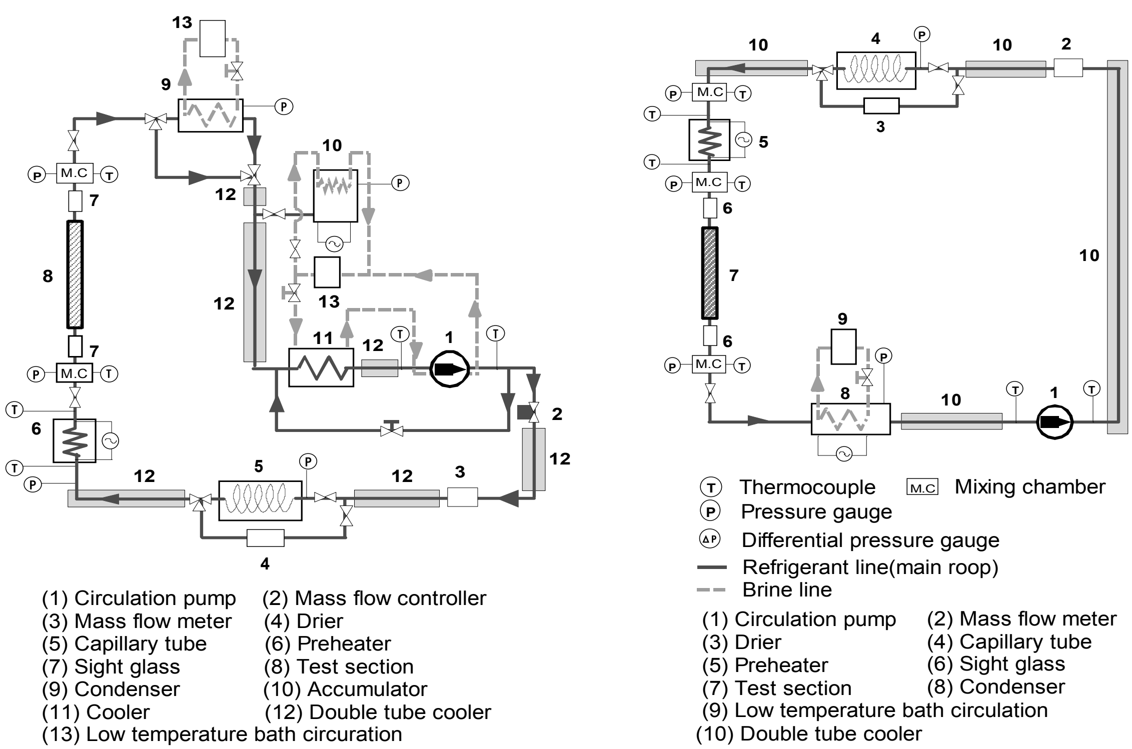
There are some studies for two-phase flow boiling heat transfer in a small vertical tube. Lazarek-Black investigated the flow boiling of CFC113 in the tube previously described for upward and downward flow. They found no significant difference between the heat transfer coefficient for up and down flow and suggested one correlation for both flows. Lin et al. (2001) made an experiment on the flow boiling of HCFC141b in 1.0mm diameter tube for upward flow. They found that nucleate boiling is the dominant heat transfer mechanism at low quality or high heat flux while convection boiling is the dominant at low heat flux and high quality. Feroz-Kaminaga (2002) investigated the flow boiling of CFC113 in a stainless steel tube with 1.45mm inner diameter for vertical upflow. They pointed out that the measured heat transfer coefficients agreed well with the prediction by a pool boiling correlation of Stephan-Abdelsalam (1980). The same result was obtained by Wambsganss et al. (1993) for flow boiling of CFC113 in a horizontal small diameter tube of 2.92mm inner diameter. Sumith et al. (2003) investigated the flow boiling of water in a stainless steel tube with 1.45mm for upward flow. They found that existing flow boiling correlations greatly underestimted the measured heat transfer coefficient especially in a low heat flux condition and the underprediction gradually decrease with increasing heat flux.

There are very few studies about flow boiling heat transfer and pressure drop in a vertical small tube, especially for downward flow. The characteristics are not clarified still enough. In this study, therefore, characteristics of flow boiling heat transfer and pressure drop of a refrigerant R410A, used in air conditioning systems widely, flowing in a vertical copper smooth tube with 1.0 mm inside diameter were investigated for upward and downward flow, respectably.

2. EXPERIMENTAL APPARATUS AND TEST CONDITIONS

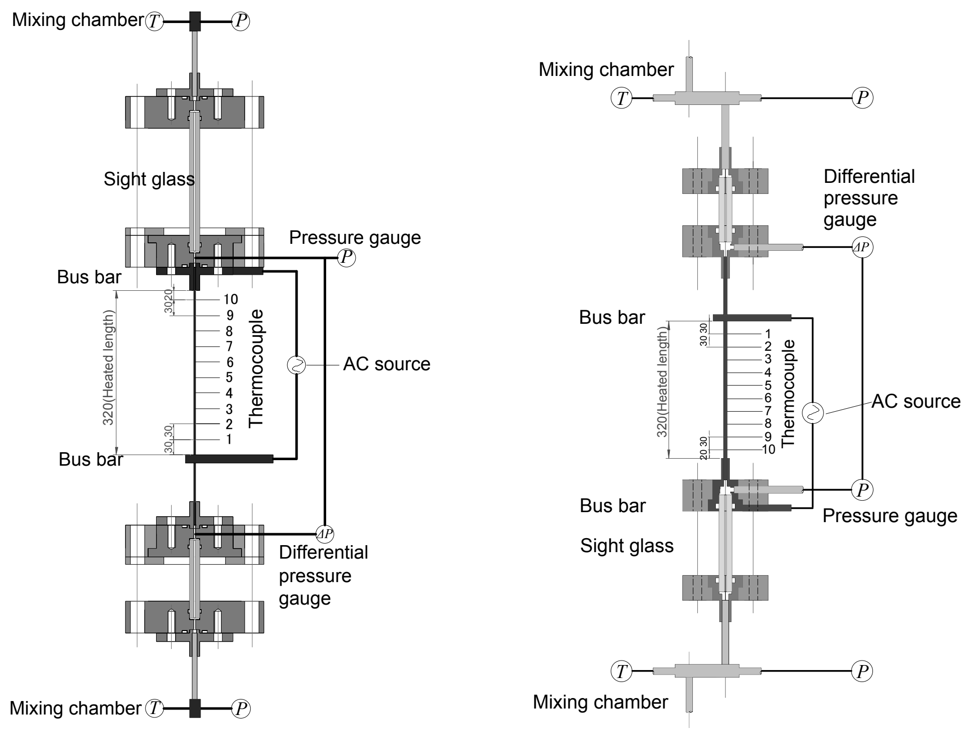
2.1 Experimental Apparatus

The experimental apparatus used in the present study for upward and downward flow are illustrated schematically in Figure1, and dimensions of the test section and locations of thermocouples are shown Figure 2.



(a) Upward flow (b) Downward flow

Figure 1: Schematic diagrams of the experimental apparatuses



(a) Upward flow (b) Downward flow

Figure 2: Test sections

The experimental apparatus for downward flow was simpler than for upward flow. But the major components were same in each apparatus. The refrigerant liquid discharged by a gear pump. The liquid flowed into a preheater through a mass flow meter. The preheater heated the refrigerant to a desired enthalpy at the inlet of the test section. The refrigerant returned to the pump through the test section and a condenser. Refrigerant in the condenser was cooled by a temperature controlled brine to maintain a desire saturation temperature at the outlet of the test section. The test section was made of copper smooth tube. The inside and outside diameter of the tube were 1.0 mm and 1.4 mm, respectively. The test tube was arranged vertically. A heated length of the test tube was 320 mm and a heated region was connected to an AC power supply. Pressure drop between inlet and outlet pressure tap was measured by differential pressure gauge. The wall temperature of the test section was measured at ten locations with 0.1 mm diameter copper-constantan thermocouples placed at 30mm interval on the outer surface of the tube. Visualization section of a sight glass tube was located after the test section. The sight glass was made of quartz glass. The inside and outside diameter of the sight glass were 1.0mm and 6.0mm, respectively. A temperature of the surrounding air was adjusted so as to be approximately equal to the temperature on the tube wall, thus minimizing the heat transfer between the test tube and the surroundings.

2.2 Test Conditions

In this study, experiments were performed to examine characteristics of flow boiling heat transfer, frictional pressure drop and flow patterns. The experiments include upward and downward flow configurations. Pressure drops were measured and flow patterns were observed at mass fluxes of 100 and 200 kg/(m^2·s) for upward flow and of 30, 50, 100 and 200 kg/(m^2·s) for downward flow, quality from 0.1 to 0.9 and evaporation temperature of 10°C and under adiabatic conditions. Local heat transfer coefficients were measured in a range of mass fluxes from 30 to 200 kg/(m^2·s), heat fluxes from 1 to 16 kW/m^2 and quality from 0.1 to over 1 at evaporation temperature of 10°C.

3. DATA REDUCTION

3.1 Frictional Pressure Drop

Two components, that is, the frictional pressure drop ΔP_f and static pressure drop ΔP_s are included in the measured total pressure drop ΔP_a at adiabatic condition. Therefore, the frictional pressure drop was calculated as

$$\Delta P_f = \Delta P_a - \Delta P_s \tag{1}$$

The measurement error of frictional pressure drop was within ±0.3 kPa in general. The maximum measurement errors, however, estimated to be ±1 kPa at quality of about 0.1, because the estimation error of static pressure drop became larger in lower qualities.

3.2 Heat Transfer Coefficient

The local heat transfer coefficient α is defined as

$$\alpha = \frac{q}{T_i - T_b} \tag{2}$$

where q is the heat flux, T_i is the inner wall temperature and T_b is the bulk temperature of the fluid. The heat flux q was calculated based on the power of AC supply. The heat transfer between the test tube and surrounding was considered to be negligible. The inner wall temperature T_i was defined equal to the measured outer wall temperature at each thermocouple location because the difference between the outer and inner wall temperatures was as small as can be neglected. The bulk fluid temperature T_b was determined from the bulk fluid enthalpy which was calculated based on the fluid enthalpy at the inlet of the preheater and heat input at the preheater. Measurement errors of thermocouples were estimated to be within ±0.08 °C. Therefore, errors of heat transfer coefficients were estimated to be within ±15% in general. The maximum errors, however, evaluated to be from -30 to +70 % under the condition at small heat flux of 1 kW/m^2 and of highly good heat transfer leading to the small temperature difference between the wall and the fluid saturation temperature.

4. RESULTS AND DISCUSSION

4.1 Flow Pattern

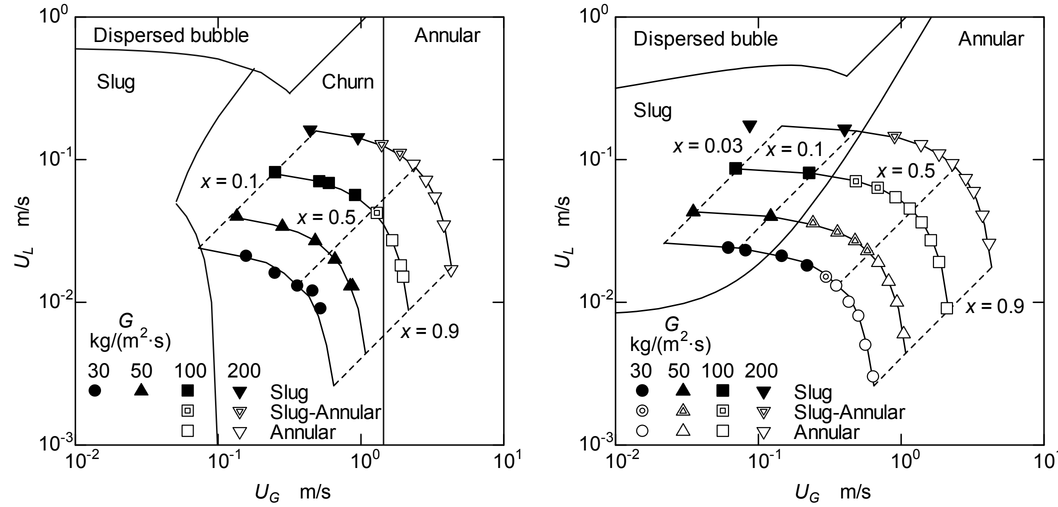
Figure 3 shows the classification of the observed flow pattern in the relation between superficial liquid velocity U_L and superficial vapor velocity U_G on the flow pattern maps by Taitel et al. (1980) for upward flow and by Barnea (1987) for downward flow. Three types of flow pattern were observed both in upward flow and downward flow: A slug, a slug-annular and an annular flow. In the case of the upward flow, while slug flow was observed in the whole range of quality at low mass velocity, transition from slug flow to annular flow with the increase of quality was noted at high mass flow rate. On the other hand, in the case of the downward flow, transition from slug to annular flow was observed even at low mass velocity condition. In the upward flow, a churn flow regime was not observed. The flow pattern map by Taitel et al. reproduced well observed flow patterns in the upward flow with replacing a churn flow in the map by a slug flow. For the downward flow, observed flow patterns were well reproduced by the Barnea flow pattern map except at mass velocity of 30 $kg/(m^2 \cdot s)$.

4.2 Frictional Pressure Drop

Figure 4 shows the relation between measured frictional pressure drop gradient $\Delta P_f / \Delta L$ and quality x for upward flow and downward flow. The symbols in Figure 4 are the same as those in Figure 3. In high quality region where the annular flow pattern was observed, measured frictional pressure drops showed almost equal values in both flow directions, and were reproduced well by Lockhart-Martinelli correlation using the Chisholm parameter of 8. In the slug flow region at low quality, however, significant difference in the pressure drop was found between upward and downward flows. The pressure drops for the downward flow were significantly large compared with those for the upward flow. This is supposed due to the vapor plug blocking in liquid flow in the downward flow; accordingly the two-phase fluid becomes difficult to flow.

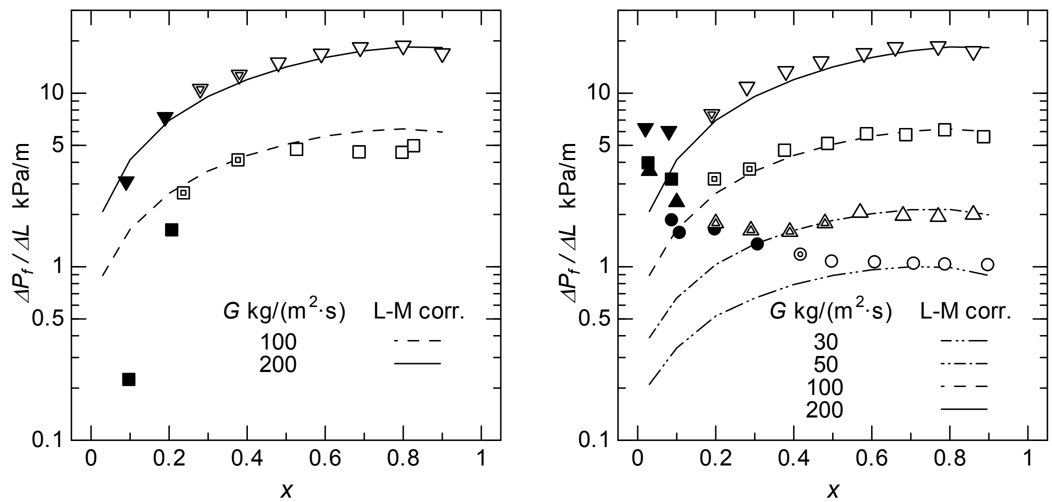
4.3 Boiling Heat Transfer

Measured heat transfer coefficients α are plotted against quality x in Figure 5. There was no noticeable difference in heat transfer between upward flow and downward flow. The characteristics of heat transfer were divided into two categories mainly dependently of the quality. First, in the region of quality below about 0.5, the effect of heat flux was apparent, but the effect of mass velocity seemed relatively small. Therefore, in the lower quality region,



(a) Upward flow with the map by Taitel et al. (b) Downward flow with the map by Barnea

Figure 3: Flow pattern diagrams and observation results



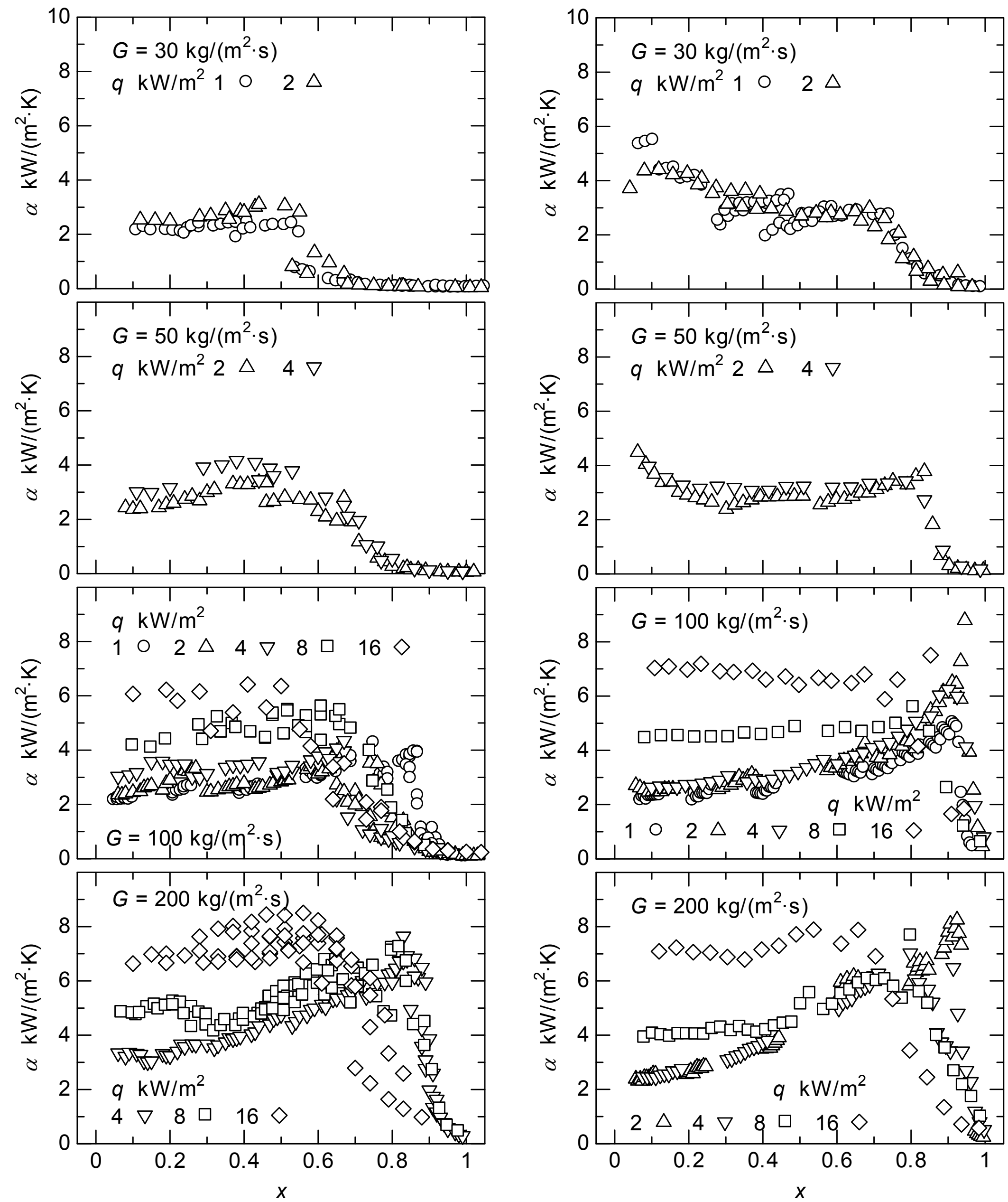
(a) Upward flow (b) Downward flow

Figure 4: Measured frictional drop and comparison with Lockhart-Martinelli correlation

nucleate boiling was considered to be dominant in heat transfer. However, the measured heat transfer coefficients were significantly higher than the calculation by the Stephan-Abdelsalam correlation for pool boiling, so the effect of convection heat transfer through the thin liquid film is necessary to be taken into consideration. Second, in the higher quality region, the effect of mass flow rate was marked and, in addition, there was seen the increase in heat transfer with quality. Therefore, forced convection was predominant in such high quality region.

At low mass velocity and low quality, there was seen large difference between upward flow and downward flow. In such cases, heat transfer coefficients became larger in the downward flow than in the upward flow. This difference in heat transfer between upward and downward flows coincided in the quality condition with the difference in frictional pressure drop between both flows described previously.

Comparing heat transfer correlations by Lazarek-Black (1982) and Sumith et al. (2003) applicable to flow boiling in a vertical tube with the present data, there was found no satisfying agreement.



(a) Upward flow (b) Downward flow

Figure 5: Heat transfer coefficient vs. quality

4.4 Dryout quality

Both in upward flow and downward flow, dryout was found to proceed over a certain quality range. Therefore defined were two dryout qualities; dryout inception quality and dryout completion quality. Such an example was shown in Figure 6. Figure 7 shows the relation between these dryout qualities and mass velocity. Dryout inception and completion qualities became higher with increasing mass velocity and furthermore the inception quality decreases with increasing heat flux. In general, both the dryout qualities were larger in downward flow than in upward flow, and the difference was seen larger with the decrease of mass velocity. As the reason of the difference, it is supposed that, in the downward flow, the more uniform thickness of annular liquid film on the perimeter is achieved and the difference in the velocity between vapor core and liquid film becomes smaller compared with upward flow.

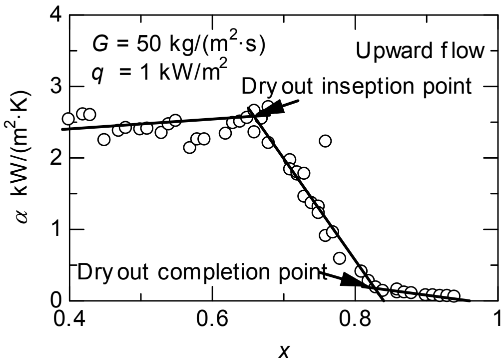
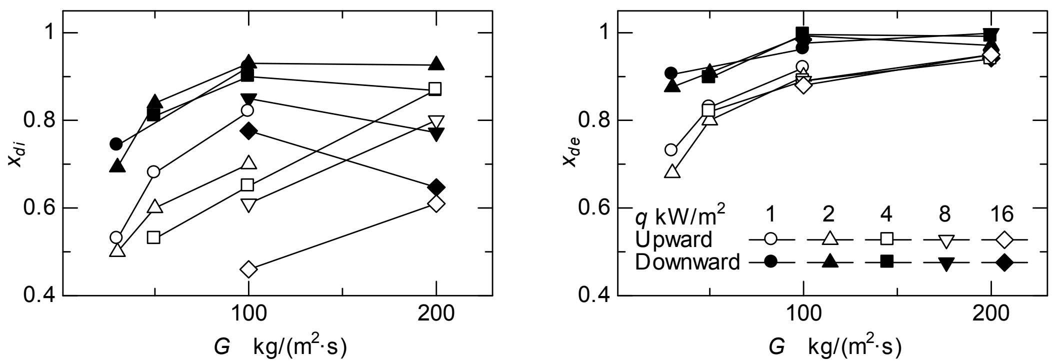


Figure 6: Example of define of dryout qualities



(a) Dryout inception quality

(b) Dryout completion quality

Figure 7: Dryout qualities vs. mass velocity

5. CONCLUSIONS

Experiments were performed to examine characteristics of flow boiling heat transfer and pressure drop of a refrigerant flowing in a vertical copper smooth tube with 1.0 mm inside diameter. Three types of flow pattern were observed both in upward flow and downward flow: A slug, a slug-annular and an annular flow. Annular flow pattern, which was confined to high mass flow rate condition in upward flow, was observed even under low flow rate condition in downward flow. The flow pattern maps by Taitel et al. and by Barnea reproduced well flow patterns observed in upward flow and downward flow, respectively, with replacing a churn flow in the Taitel map by a slug flow.

Measured pressure drops in the annular flow regime showed almost the same values both for upward and downward flows, which were reproduced well by Lockhart-Martinelli correlation using the Chisholm parameter of 8. In the slug flow, however, pressure drops of the downward flow were found to be significantly large compared with those for the upward flow.

There was no noticeable difference in heat transfer between upward flow and downward flow. The characteristics of heat transfer were placed into two categories mainly dependently of the quality. At qualities below about 0.5, the effect of heat flux was apparent, but the mass flow rate effect seemed relatively small. These were, however, opposite in the higher quality region, in which the effect of mass flow rate was marked and, in addition, there was seen the increase in heat transfer with quality. Nucleate boiling and forced convection were considered to be dominant in heat transfer in lower and higher quality regions, respectively. Comparing with the present data, there was found no conventional correlation, which was applicable to flow boiling heat transfer in a vertical tube, satisfying measured heat transfer characteristics.

Both in upward flow and downward flow, dryout was found to proceed over a certain quality range. Therefore defined were two dryout qualities; dryout inception quality and dryout completion quality. In general, both the dryout qualities were larger in downward flow than in upward flow.

NOMENCLATURE

C	Chisholm's parameter	(–)
G	mass velocity	(kg/(m^2·s))
q	heat flux	(kW/m^2)
T_i	inner wall temperature	(°C)
T_b	bulk temperature	(°C)
U_G	superficial vapor velocity	(m/s)
U_L	superficial liquid velocity	(m/s)
x	vapor quality	(–)
x_i	dryout inception quality	(–)
x_e	dryout completion quality	(–)
α	heat transfer coefficient	(kW/(m^2·K))
ΔP_a	total pressure drop	(kPa)
ΔP_f	frictional pressure drop	(kPa)
$\Delta P_f/\Delta L$	frictional pressure drop gradient	(kPa/m)
ΔP_s	static pressure drop	(kPa)

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ACKNOWLEDGEMENT

The present study was partially supported by Daikin Industries, Ltd., Japan, and the authors are grateful to this company.

MANUSCRIPT REFERENCE NO. 2421

19th International Compressor Engineering Conference at Purdue
12th International Refrigeration and Air Conditioning Conference at Purdue
July 14-17, 2008

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