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Experimental study on void fraction of two-phase flow inside a micro-fin tube

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In this paper the void fraction and flow pattern of the two-phase flow in a micro-fin tube were investigated experimentally for a pure refrigerant HFC134a. The experiment was carried out at a pressure range of 0.6 and 1.2 MPa with mass velocities of 90 and 180 kg/m^2 s, in which the vapor quality varied from 0 to 1. The void fraction was measured by means of simultaneously closing valves of both sides of the test tube at adiabatic condition. Experimental results for the micro-fin tube were compared with previous correlations for smooth tube. It was found that experimental data of the micro-fin tube were lower than predicted results using correlations for smooth tube. As a trial, an empirical correlation was proposed, based on the present experimental results. The observation of the flow pattern through a sight glass was also carried out with a camera. The observation results were compared with modified Baker's map.

Introduction

Recently, many researchers and engineers have devoted themselves to improving the performance of air conditioning and refrigeration systems. Some of them have been investigating the heat transfer enhancement method, especially in two-phase flow with evaporation or condensation. In this research field, one of the most fundamental problems is to clarify the characteristics of the void fraction in evaporators and condensers. For instance, the knowledge of the void fraction is required in the evaluation of mean fluid density, pressure drop and heat transfer in two-phase flow.

In the middle of the 20th century, i.e. in the 1960s, many researchers proposed dozens of correlations for the void fraction in smooth tube. Matinelli-Nelson [1] made a first attempt to predict the void fraction in smooth tube. Then, Levy [2], Fauske [3], Thom [4], Bankoff [5], Zivi [6], Baroczy [7], Andeen [8], Smith [9] and others proposed different types of correlations reduced from different theories and assumptions. However, these correlations are only available for smooth tubes.

For the last two decades many kinds of micro-fin tubes have been widely used in air conditioning and refrigeration systems because their performance of heat transfer is higher than that of smooth tubes. In this relation, much research on two-phase flow in micro-fin tubes has been carried out. However, most of the studies only treated the pressure drop and heat transfer characteristics. Until 1999 Yashar *et. al.* [10] reported their experimental results of void fraction with micro-fin tubes and proposed a simple correlation which showed a good agreement with their experimental data. To the best of our knowledge, there is no much appropriate information about void fractions in micro-fin tubes to be found in the 'open' literature, although the research on void fractions is essential to evaluate the pressure drop and heat transfer characteristics.

In this work, the void fraction and flow pattern of the two-phase flow in a micro-fin tube are investigated experimentally for pure refrigerant HFC134a. This work is placed as a first step to clarify the fundamental characteristics of two-phase flow.

Nomenclature

d_i = mean inside diameter of micro-fin tube[m]
G = refrigerant mass velocity [kg/m^2.s]
h_{b0} = bulk refrigerant specific enthalpy at inlet of evaporator [J/kg]
h_b = bulk refrigerant specific enthalpy at the inlet of the test tube [J/kg]
$h_{l\,sat}$ = specific enthalpy of saturated liquid [J/kg]
$h_{v\,sat}$ = specific enthalpy of saturated vapor [J/kg]
m = mass of refrigerant [kg]

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m' = mass of refrigerant collected in sampling vessel [kg]
Q = heat transfer rate in evaporator [W]
Q_{loss} = heat loss of evaporator [W]
V = volume inside micro-fin tube [m^3]
V_l = volume of liquid in test tube [m^3]
V_v = volume of vapor in test tube[m^3]
V' = volume of the test tube and piping [m^3]
x = vapor quality [-]

Greek symbol

ρ_v = density of vapor [kg/m^3]
ρ_l = density of liquid [kg/m^3]
ρ_v' = refrigerant vapor density calculated[kg/m^3]
ξ = void fraction [-]
ξ_{homo} = void fraction for homogeneous flow [-]
ξ_{smith} = void fraction calculated from Smith [-]
ψ = parameter of Baker's flow pattern map [-]
ζ = parameter of Baker's flow pattern map [-]

Experimental Apparatus and Method

Figure 1 shows a schematic view of the experimental apparatus, which consists of three loops: a refrigerant loop, a water loop and a brine loop. In the refrigerant loop, subcooled refrigerant liquid is delivered with a magnetic gear pump (1) through a desiccant filter (2), a mass flow meter (3), a preheater (4) and a mixing chamber (5) to a heat exchanger (6). The preheater (4) and the heat exchanger (6) are used to heat the refrigerant liquid close to the saturation state. Then, the refrigerant flows through an evaporator (10) into a test section for the measurement of the void fraction (11). The evaporator (10), around which an electrical heater is wrapped, is used to regulate the vapor quality at the entrance of the test section (11). The refrigerant flowing from the test section returns through a sight section for observation of the flow pattern (12), an after-heater (14) and two condensers (15,16) to the pump (1). The after-heater (14) and two condensers (15,16) are used to adjust the refrigerant pressure level in the loop. The water loop, which consists of a heat source tank (7), a centrifugal pump (8) and a gear-type flow meter (9), is used to supply heating water to the heat exchanger (6). The brine loop, which consists of a brine tank (18), three centrifugal pumps (8), two float-type flow meters (19) and a chilling unit (20), is used to condense the refrigerant.

The refrigerant mass flow rate is measured using the mass flow meter with ±1.5 kg/h resolution. The DC electric power applied to the heater around the evaporator is measured to evaluate the heat transfer rate. Pressures at inlet and outlet of the evaporator and the test section are measured using an absolute pressure transducer with ±2 kPa resolution. Pressure drops through the evaporator and the test section are measured using a differential pressure transducer with ±0.2 kPa resolution. Refrigerant temperatures in the refrigerant loop are measured using several ϕ 0.5 mm sheathed K-type thermocouples, calibrated in advance within ± 0.05 °C uncertainty.

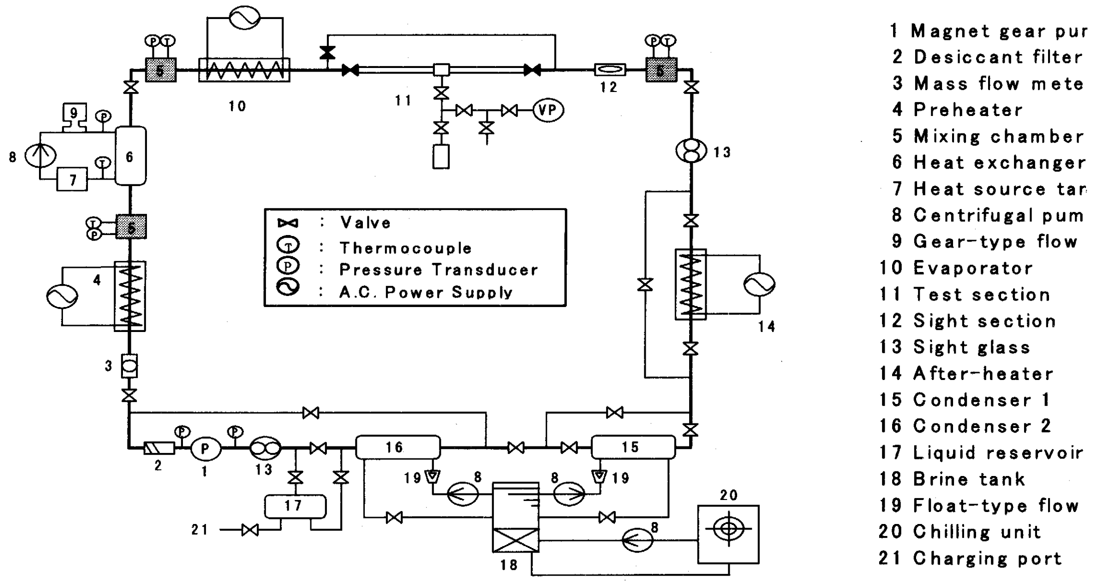
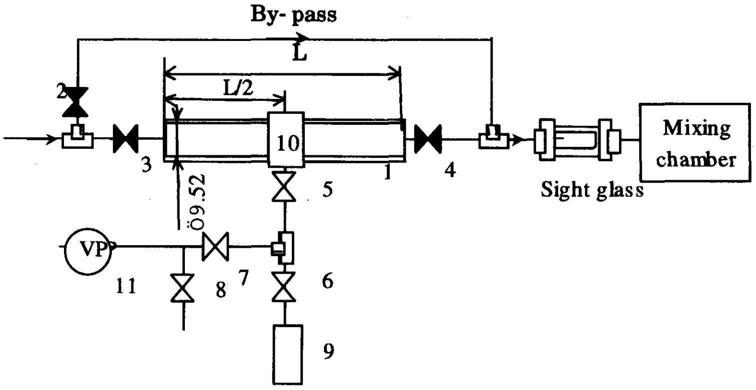


Figure 1 Schematic view of experimental apparatus



1	TestTube	9	SamplingVessel
2,3,4	Valve with Air Actuator	10	Sampling Port
5,6,7,8	Valve	11	Vacuum Pump

Figure 2 Schematic view of test section

Figure 2 shows the schematic view of the test section for the measurement of the void fraction. The test section consists mainly of a test tube (1), a bypass tube, three valves with air actuators (2, 3, 4), a sampling vessel (9), a sampling port (10) and a vacuum pump (11). The void fraction of two-phase flow is measured by means of simultaneously closing valves. The operating procedure of this measurement is as follows:

(1) Valves (2), (5), (7) and (8) are closed, while the others are opened. The vessel (9) is kept around 0 ℃ using an ice box.

(2) After opening the valve (7), the vessel (9) including the piping is evacuated with the vacuum pump (11). Then, the valve (7) is closed again.

(3) The refrigerant temperature and pressure in the test tube (1) are measured at the sampling port (10). The refrigerant flow rate, the heat transfer rate in the evaporator, and the refrigerant pressure and temperature in the refrigerant loop are also measured.

(4) Valves (3) and (4) are closed with air actuators instantaneously. At the same time, the valve (2) is opened.

(5) The valve (5) is opened. Then, the refrigerant is condensed and collected in the sampling vessel (9). After that, the valve (6) is closed.

(6) The refrigerant temperature and pressure in the test tube are measured again at the sampling port (10) to evaluate the amount of the remaining refrigerant vapor.

(7) The weight of the sampling vessel containing the refrigerant is measured with an accurate electrical balance of ±1 mg resolution. Then, the net weight of refrigerant is obtained.

The evaluation method of the void fraction and vapor quality in the test tube is as follows:

(1) The mass of the refrigerant existing in the test tube before closing valves (3) and (4), m, is calculated as,

$$m = m' + \rho_V' V' \qquad (1)$$

where m' is the mass of the refrigerant collected in the sampling vessel (9), ρ_V' is the refrigerant vapor density calculated using the pressure and temperature measured in the operating procedure (6), and V' is the volume of the test tube and the piping between the sampling port (10), and valves (6) and (7).

(2) The void fraction, ξ, is obtained from the following equation.

$$\xi = \frac{V_v}{V_v + V_l} = \frac{\rho_l V - m}{(\rho_l - \rho_v)V} \qquad (2)$$

where V is the volume of the test tube, V_v and V_l are the volumes of vapor and liquid in the test tube, respectively, and ρ_l and ρ_v are the refrigerant liquid and vapor densities at saturated state, respectively, both of which are calculated using the pressure measured in the operating procedure (3).

(3) The bulk refrigerant specific enthalpy at the inlet of the test tube, h_b, is calculated by

$$h_b = \frac{4(Q - Q_{loss})}{G \pi d_i^2} + h_{b0} \qquad (3)$$

where Q is the heat transfer rate in the evaporator, Q_{loss} is the heat loss from the evaporator to the ambience, G is the refrigerant mass velocity, d_i is the mean inside diameter of the test tube, and h_{b0} is bulk refrigerant specific enthalpy at the inlet of the evaporator.

(4) The quality of refrigerant at the inlet of the test tube, x, is evaluated from

$$x = \frac{h_b - h_{l\,sat}}{h_{v\,sat} - h_{l\,sat}} \qquad (4)$$

where $h_{l\,sat}$ and $h_{v\,sat}$ are the liquid and vapor specific enthalpies at the saturated state, respectively, both of which are also calculated using the pressure measured in the operating procedure (3).

In the present study, a commercially available micro-fin tube made of copper is used as test tube. The inside surface of this tube is grooved spirally; its dimensions are summarized in Table 1. Experiments for void fraction and flow pattern were conducted using refrigerant HFC134a as a working fluid in ranges summarized in Table 2. All of the data signals were collected and recorded by a data acquisition system. Uncertainties in measuring the void fraction and the quality in the present study are estimated to be ± 0.75% and ± 3.78%, respectively. The reliability in the measurement of void fraction is affected by the following parameters: (1) the degree of superheat of remaining vapor in the test tube after collecting the most amount of refrigerant in the sampling vessel, (2) the length of the test tube and (3) the volume of the sampling vessel. These effects were confirmed prior to the present experiments.

Table 1 dimensions of micro-fin tube

Item		unit	Scale
Outside diameter	D	mm	9.52
Mean inside diameter	d_i	mm	8.86
Mean wall height	t	mm	0.33
Number of fins	N	-	70
Fin height	h	mm	0.18
Bottom wall height	t_1	mm	0.28
Minimum inside diameter	d_1	mm	8.60
Top angle	γ	°	25
Helix angle	α	°	25
A_{aug} / A_{smooth}	η_A	-	1.67

Table2 Summary of test condition

P_{in}	0.6 ~ 1.2 MPa	G	90, 180 kg/m^2 s
T_{sat}	21.57 ~ 46.31 °C	x	0.0 ~ 1.0

Results and Discussions

Flow Pattern Distribution

The flow pattern of the two-phase flow was observed through the sight section located just after the test section for void fraction measurements. Figure 3 shows the present result plotted on modified Baker's map for smooth tube [11]. The type of flow pattern varies with the quality of refrigerant. At a definite mass flux, the slug, wavy-annular and annular flow types appear sequentially when the quality increases from 0.0 to 1.0. It is also found that the quality in the transition zone from slug to annular type decreases with increase in mass velocity. Furthermore, it is important to notice that the annular flow area of the micro-fin tube is enlarged toward the low quality region compared with the map for smooth tube.

Void Fraction Distribution

Figure 4 shows the comparison of the measured void fraction in the micro-fin tube with previous correlations for smooth tubes [1, 3, 6, 7, 9]. It is found that data of void fractions in the micro-fin tube are generally much lower than predicted values by correlations for smooth tubes at the same quality. The main reason seems to be the difference of shear force between micro-fin

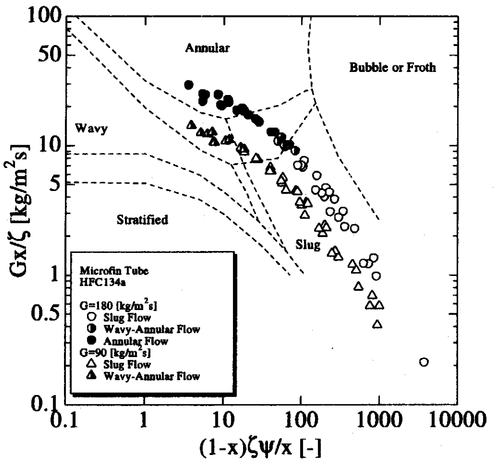


Figure 3 Flow pattern in modified Baker's Map

tube and smooth tube. In case of the micro-fin tube, the wall shear force acting on the liquid film increases due to the micro-fins, compared with smooth tubes. This leads to a lowering of the mean liquid velocity in micro-fin tube. This means that the mean liquid film thickness in micro-fin tube is larger than that in smooth tube at the same liquid flow rate. As a result, it is inferred that the void fraction in micro-fin tube is lower than that in smooth tube. It is noted that the entrainment of liquid into the vapor core also affects the relation between void fraction and quality. In this figure, a correlation for micro-fin tube, proposed by Yashar *et. al.* [10], is also plotted. There is no significant difference between Yashar's correlation and other correlations for smooth tube. The disagreement between present experimental results and Yashar's correlation should be studied deeply.

Figures 5 (a) and (b) show the influence of refrigerant pressure to void fraction in cases of $G = 180$ and 90 [kg/m^2 s], respectively. It is found from both figures that the void fraction at a definite quality increases with decrease of refrigerant pressure. This reason is explained as follows. The mean vapor velocity increases as the pressure decreases. This leads to an increase in the vapor shear stress acting on the liquid film. Therefore, the mean liquid film thickness decreases with a decrease of pressure. This means an increase of the void fraction.

Figures 6 (a), (b) and (c) show the relation between the void fraction and quality in case of P =1.2, 0.8 and 0.6 MPa, respectively. In all cases, it is found that there is only a slight influence of mass flow rate on the relation between void fraction and quality. There is no obvious change in void fraction with variation of mass fluxes, because the density of refrigerant has no significant change in different mass flux.

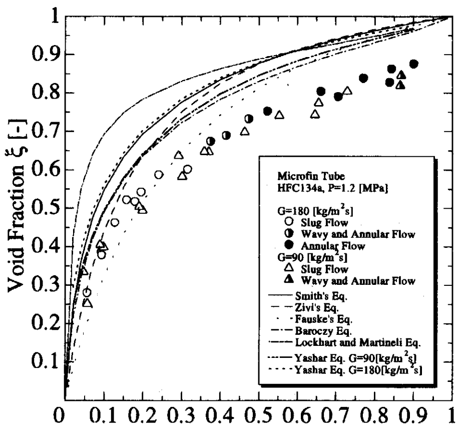
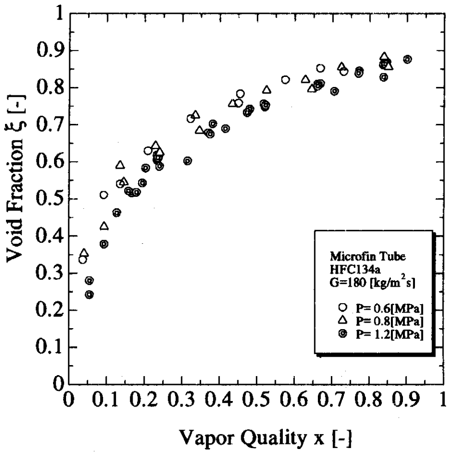
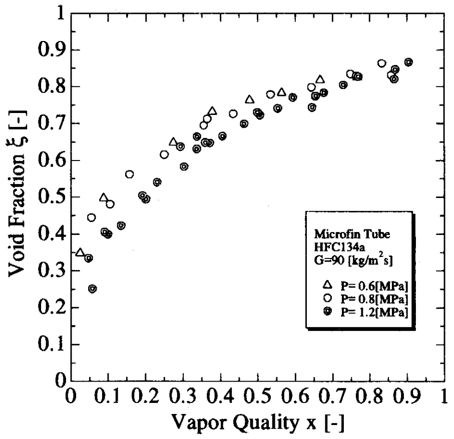


Figure 4 Comparison of void fraction with previous correlations for smooth tube and Yashar's correlation for micro-fin tube

There is no data at $x \geq 0.9$ due to the limitation of the present experiments, as shown in Figures 5 and 6. Therefore, further experimental study should be continued to clarify the characteristics of the void fraction in micro-fin tubes, especially, in the region of $x \geq 0.9$. However, in the present report, assuming the void fraction approaches $\xi = 1$ when $x = 1$, the following tentative correlation for the void fraction is proposed based on the present experimental results.



(a) G = 180 kg/m^2 s

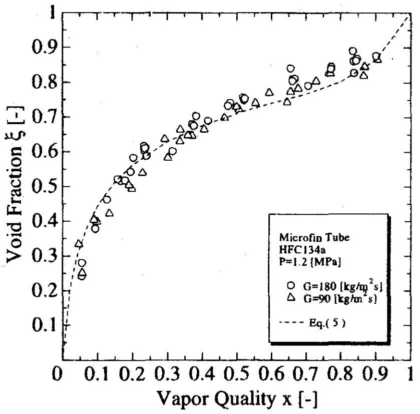


(b) G = 90kg/m^2 s

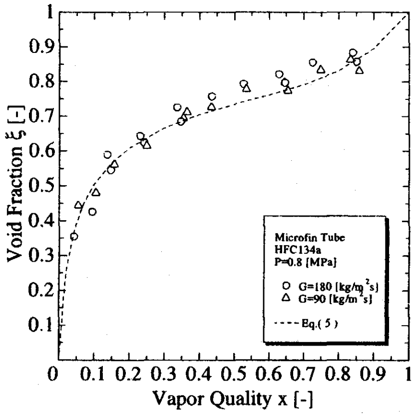
Figure 5 Effect of pressure on void fraction

$$\xi = 0.81\,\xi_{Smith} + 0.19\,x^{100(\rho_v/\rho_l)^{0.8}}\,\xi_{\mathrm{Homc}} \qquad (5)$$

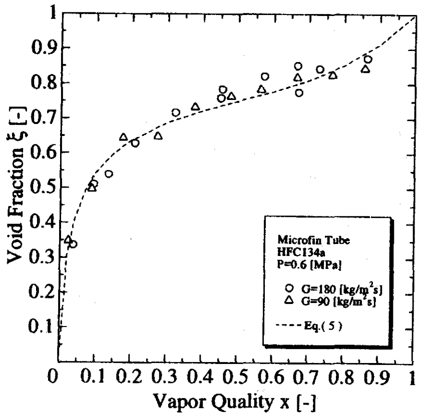
where ξ_{smith} and ξ_{homo} [9] are previous correlations



(a) P = 1.2MPa



(b) P = 0.8MPa



(c) P = 0.6MPa

Figure 6 Effect of mass velocity on void fraction

for the void fraction in smooth tubes. The correlation of equation (5) is drawn in Figure 6. The deviations between this correlation and the present data are within ±20%.

Conclusion

The flow pattern and the void fraction in a micro-fin tube were investigated experimentally for a pure refrigerant HFC134a. The main findings are:

(1) It is confirmed that the quality in the transition zone from slug to annular type decreases with increase in mass velocity in case of the micro-fin tube.
(2) The annular flow area of the micro-fin tube is enlarged toward the low quality region compared with the modified Baker's map for smooth tubes.
(3) The data of the void fraction in micro-fin tube are generally lower than the values predicted using previous correlations for smooth tubes at the same quality. Mainly, this can be explained by the difference of the shear force in micro-fin tube and in smooth tube.
(4) The effect of the refrigerant pressure on the void fraction in micro-fin tube is significant, whereas the mass flux has only a slight influence.
(5) An empirical correlation is proposed.

Further research about the void fraction in micro-fin tubes is important and required, including more detailed experiments and available analysis of theoretical models.

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