An Experimental Study On Condensation Of R134a In A Multi-Port Extruded Tube

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ABSTRACT

In the present study, the local characteristics of pressure drop and heat transfer are investigated experimentally for the condensation of a pure refrigerant R134a in multi-port extruded tubes of 8 channels in 1.36mm hydraulic diameter and 19 channels in 0.80mm hydraulic diameter. The experimental data of frictional pressure drop (F.P.D.) and condensation heat transfer coefficient (H.T.C.) are compared with previous correlations, most of which are proposed for the condensation of pure refrigerant in a relatively large diameter tube. Considering the effects of surface tension and kinematic viscosity, new correlation of F.P.D. is developed based on the Mishima-Hibiki correlation. New correlation of H.T.C. is also developed modifying the effect of diameter in the correlation of Haraguchi *et al*.

NOMENCLATURE

A : heat transfer surface of tube
Bo : Bond number
Cp : isobaric specific heat
d : hydraulic diameter
G : mass velocity
g : gravitational acceleration
Ga_L : Galileo number
H : width of tube
$H(\xi)$: function of void fraction
Nu : Nusselt number
P : pressure
Ph_L : phase change number
Pr : Prandtl number
q : heat flux
Re : Reynolds number
S : wetted perimeter length
T : temperature
α : heat transfer coefficient
Δh_{VL} : latent heat of condensation
ΔP : pressure drop
ΔP_f : frictional pressure drop
ΔP_m : pressure drop due to momentum change
λ : thermal conductivity
μ : viscosity
ν : kinematic viscosity
ξ : void fraction
ρ : density
σ : surface tension
Φ_V : two-phase multiplier factor
X_{tt} : Lockhart-Martinelli parameter

Subscripts

B : free convection condensation term
cal : prediction
exp : experiment
F : forced convection condensation term
L : liquid
R : refrigerant
S : heat sink (cooling water)
V : vapor
wi : inside surface
wo : outside surface

INTRODUCTION

From the viewpoint of global environment protection, it is urgently necessary to introduce environmentally acceptable new refrigerants and improve further the performance in the refrigeration and air-conditioning systems. As one of the methods for improving the system performance, the reduction of the diameter of heat transfer tubes is taken an interest in.

There are a few previous studies on the condensation heat transfer of refrigerants in small diameter tubes. Katsuta (1994) carried out experiments of R134a in several multi-port extruded tubes, and compared the local heat transfer characteristics with several correlations proposed for relatively large diameter tubes. Yang and Webb (1996) carried out experiments on the heat transfer of R12 in a horizontal multi-port extruded tube of 2.64 mm in hydraulic diameter and a horizontal multi-port extruded microfin tube of 1.56 mm in hydraulic diameter. Moser *et al.* (1998) proposed a correlation using the equivalent Reynolds number model, based on experimental data of heat transfer in many horizontal tubes of 4.57-12.7 mm I.D. However, it is very difficult to measure accurately the local heat transfer characteristics in a small diameter tube using the traditional methods such as the water calorimetric method, the Wilson-plot method. Accordingly, more research efforts are required to clarify the condensation process in a small diameter tube.

In the present study, local characteristics of pressure drop and heat transfer are investigated experimentally for the condensation of pure refrigerant R134a in two kinds of multi-port extruded tubes made of aluminum. The present experimental results are compared with previous correlations proposed for relatively large diameter tubes. Then, based on the present experimental data, new correlations are proposed for the frictional pressure drop and heat transfer coefficient of pure refrigerant condensing in a small diameter tube.

EXPERIMENTAL APPRUTUS

Figure 1 shows the schematic view of the experimental apparatus. The refrigerant liquid discharged from a gear pump (1) flows into an evaporator (4) through a mass flow meter (3). The refrigerant vapor generated at the evaporator flows into a test section (5). The refrigerant condensed in the test section returns to the pump through a subcooler (6) and a liquid receiver (7).

Figure 2 shows the test section, which is composed of an inlet mixing chamber, a multi-port extruded tube, and an outlet mixing chamber. Two types of multi-port extruded tubes, dimensions of which are listed in Table 1, are tested. Each test tube is 865 mm in total length and 600 mm in effective cooling length. Eight cooling water jackets are attached on both upside and downside surfaces of the test tube; the length of each jacket is 150 mm. Sixteen heat flux sensors are inserted in between the water jackets and the test tube; the length of each sensor is 75 mm. The heat flux measured with each sensor is considered as local value in the present study. The refrigerant temperature is measured with two ϕ 0.5 mm K-type sheathed thermocouples inserted in the inlet and outlet mixing chambers. The outer wall temperature of the test tube is measured with 16 T-type thermocouples of ϕ 75 μm O.D.

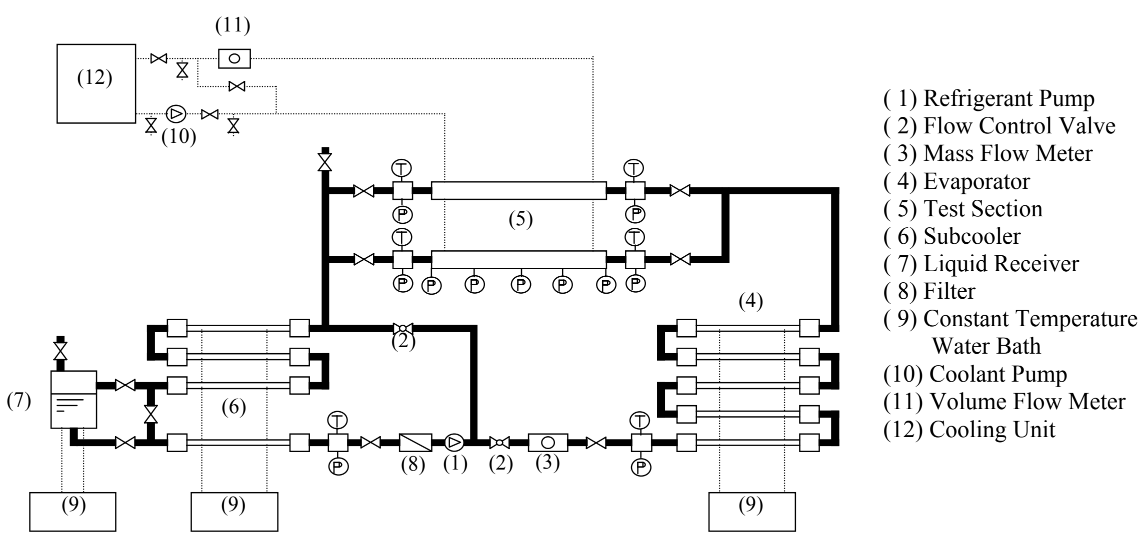


Figure 1: Schematic view of experimental apparatus.

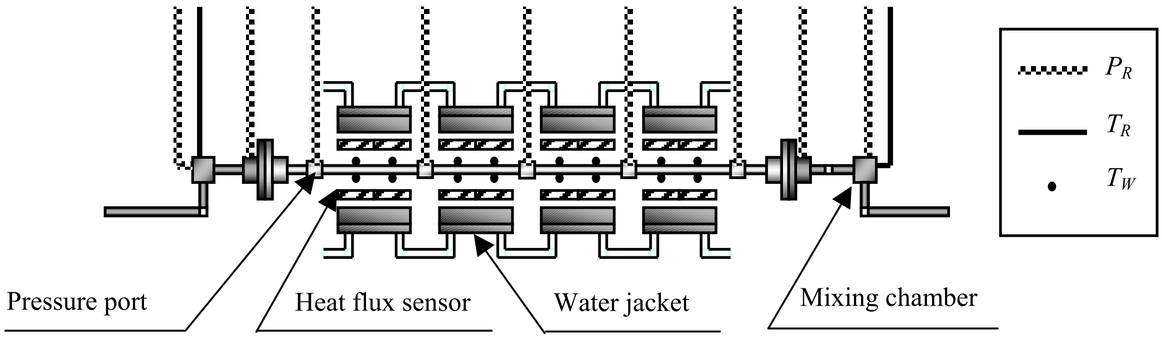


Figure 2: Schematic view of test section.

Table 1. Dimension of multi-port extruded tube

Tube Type	Type A	Type B
Channel Number	8	19
Wetted Perimeter Length [mm]	43.59	54.00
Cross Sectional Area [mm^2]	12.14	10.90
Hydraulic Diameter [mm]	1.114	0.807
Tube Length [mm]	865.0	865.0

Table 2. Experimental ranges

Tube Type	A and B
Refrigerant	R134a
G [kg/m^2 s]	100 – 700
P_{Rin} [MPa]	1.7
T_{Rin} [K]	340
T_{S} [K]	265,280,300

buried in the test tube at central points of every heat flux sensor. The inlet refrigerant pressure in the inlet mixing chamber is measured by an absolute pressure gauge. The local pressure distribution from the inlet mixing chamber to the outlet mixing chamber is measured through 8 pressure measuring ports using a differential pressure transducer

Pure refrigerant R134a is used as a test fluid. The experimental ranges are summarized in Table 2. The following assumptions are employed in the data reduction process:

(1) In the vapor single-phase region, the pressure change is estimated using the Colburn equation; this assumption is employed to estimate the total pressure drop in the vapor single-phase region and determine the starting point of condensation.
(2) In the two-phase region, the pressure drop due to the momentum change is estimated using the separated flow model; subtracting this pressure drop from the measured pressure drop leads to the frictional pressure drop.
(3) In a subsection where the condensation starts, the heat flux is assumed to be uniform in the refrigerant flow direction; this assumption is employed to explore the starting point of condensation.
(4) In the determination of the ending point of condensation, the heat flux is also assumed to be uniform in a subsection where the condensation terminates.
(5) The representative temperature at the inner wall of the test tube is estimated assuming the one dimensional heat conduction in the tube wall.

By solving the energy balance equation in each subsection successively in the refrigerant flow direction along with measured data of the refrigerant flow rate, the wall heat flux and the pressure, the quality change in each subsection is calculated. The local heat transfer coefficient α and local Nusselt number Nu are obtained as,

$$\alpha = \frac{A_{\mathrm{wo}}}{A_{\mathrm{wi}}} \frac{q}{(T_{\mathrm{R}} - T_{\mathrm{wi}})} = \frac{2H}{S} \frac{q}{(T_{\mathrm{R}} - T_{\mathrm{wi}})} \quad , \quad Nu = \frac{\alpha\, d}{\lambda_{\mathrm{L}}} \qquad (1), (2)$$

where A_{wo} is the outer cooling area of the test tube, A_{wi} is the inner cooling area of the test tube, q is the wall heat flux over a subsection, H is the width of the test tube, S is the wetted perimeter length of the test tube, T_{R} is the arithmetic mean of refrigerant temperature at the inlet and the outlet of a subsection, T_{wi} is the inner wall temperature of the test tube, d is the hydraulic diameter of the test tube, and λ_{L} is the thermal conductivity of refrigerant liquid. The frictional pressure drop between the neighboring pressure measuring ports, ΔP_{f} , is obtained by subtracting the effect of momentum change from the measured pressure drop as,

$$\frac{\Delta P_{\mathrm{f}}}{\Delta z} = \frac{\Delta P}{\Delta z} - \frac{\Delta P_{\mathrm{m}}}{\Delta z} \quad , \qquad \frac{\Delta P_{\mathrm{m}}}{\Delta z} = -\frac{\Delta}{\Delta z}\left[\frac{G^2\ x^2}{\xi\,\rho_{\mathrm{V}}} + \frac{G^2\,(1-x)^2}{(1-\xi)\rho_{\mathrm{L}}}\right] \tag{3), (4}$$

where ΔP is the measured pressure drop, ΔP_{m} is the pressure drop due to momentum change, ξ is the void fraction, Δz is the distance between the neighboring pressure measuring ports, and ρ_{L} and ρ_{V} are the densities of refrigerant liquid and vapor, respectively. The void fraction is estimated by the Smith equation (Smith 1971) as,

$$\psi = \left[1 + \frac{\rho_{\mathrm{V}}}{\rho_{\mathrm{L}}}\left(\frac{1-x}{x}\right)\left(0.4 + 0.6\sqrt{\frac{\dfrac{\rho_{\mathrm{L}}}{\rho_{\mathrm{V}}} + 0.4\dfrac{1-x}{x}}{1 + 0.4\dfrac{1-x}{x}}}\right)\right]^{-1} \tag{5}$$

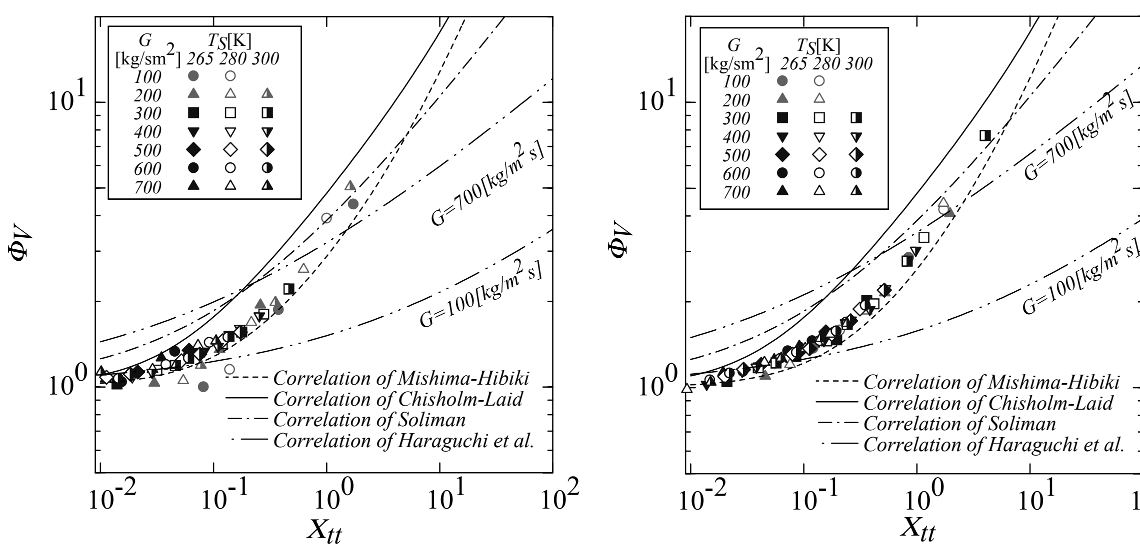
Then, the two-phase multiplier factor Φ_{V} is obtained as,

$$\Phi_{\mathrm{V}} = \sqrt{\frac{\Delta P_{\mathrm{f}}/\Delta z}{\Delta P_{\mathrm{V}}/\Delta z}} \tag{6}$$

where ΔP_{V} is the pressure drop when only the vapor component flows in the test tube, which is estimated by the Colburn equation. Thermophysical properties in data reduction of each experiment are calculated using the REFPROP Version 6.0 (McLinden *et al*. 1998).

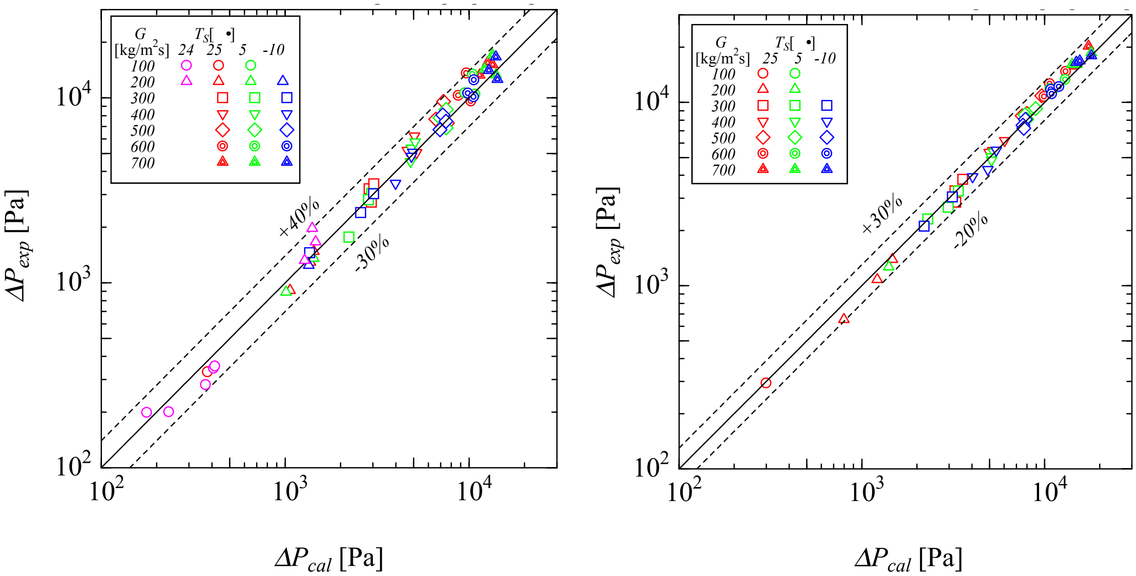
RESULTS AND DISCUSSION

Figure 3 shows the relation between the two-phase multiplier factor ϕ_V and the Lockhart-Martinelli parameter X_{tt}, where Figs. 3(a) and (b) are the results for test tubes of Type A and Type B, respectively. In both figures, correlations proposed by Mishima-Hibiki (1995), Chisholm-Laird (1958), Soliman *et al*. (1968), and Haraguchi *et al*. (1994a) are represented by a dashed line, a solid line, a chain line and a double-dotted chain line, respectively. Both correlations of Chisholm-Laird and Soliman *et al*. overpredict the experimental data, and the correlation of Haraguchi *et al*. shows completely different trend toward the experimental data. The main reason of these discrepancies is due to neglecting the effect of tube diameter in their correlations. On the other hand, Mishima-Hibiki correlation, in which the effect of tube diameter is considered, shows the same trend as the experimental data.



(a) Tube Type A (b) Tube Type B

Figure 3: Relation between Φ_{V} and X_{tt}.



(a) Tube Type A (b) Tube Type B

Figure 4: Comparison between measured and predicted pressure drop by equation (7).

Table 3: Correlations of frictional pressure drop and heat transfer

Frictional pressure drop:

$$\Phi_{\mathrm{V}}^2 = 1 + 13.17\left(\frac{\nu_{\mathrm{L}}}{\nu_{\mathrm{V}}}\right)^{0.17}\left(1 - e^{-0.6\sqrt{Bo}}\right)\mathrm{X}_{tt} + \mathrm{X}_{tt}^2 \qquad (7)$$

Heat Transfer Correlation:

$$Nu = \left(Nu_{\mathrm{F}}^2 + Nu_{\mathrm{B}}^2\right)^{1/2} \qquad (8)$$

where $Nu_{\mathrm{F}} = 0.0112 Pr_{\mathrm{L}}^{1.37}\left(\Phi_{\mathrm{V}} / \mathrm{X}_{tt}\right) Re_{\mathrm{L}}^{0.7}$, $Nu_{\mathrm{B}} = 0.725\left(1 - e^{-0.85\sqrt{Bo}}\right) H(\xi)\left(\frac{Ga_{\mathrm{L}}\, Pr_{\mathrm{L}}}{Ph_{\mathrm{L}}}\right)^{1/4}$

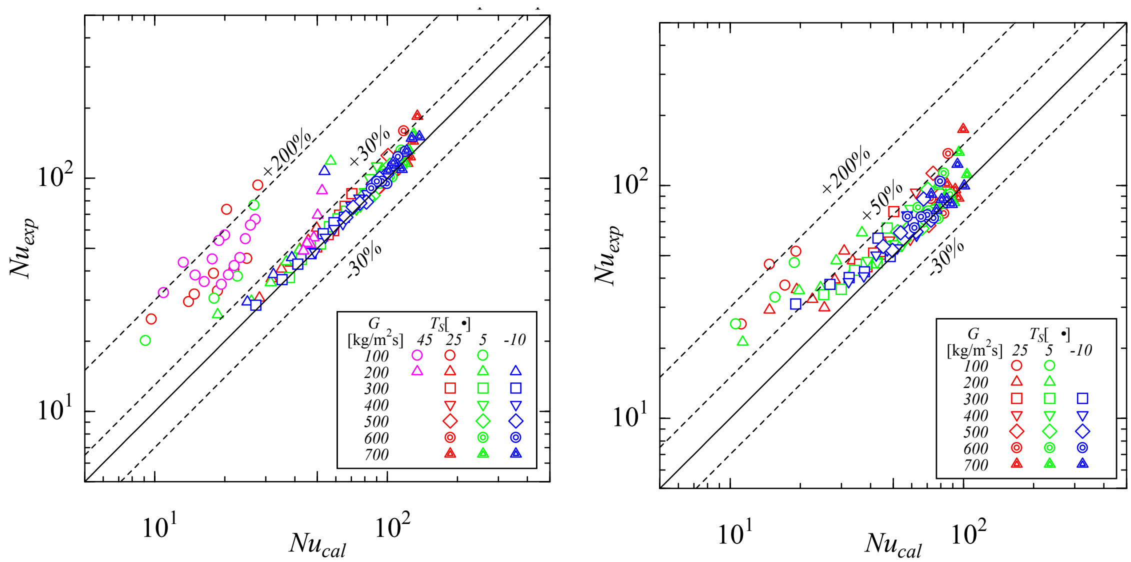
(note)

$Bo = \dfrac{d^2 g(\rho_{\mathrm{L}} - \rho_{\mathrm{V}})}{\sigma}$, $Ga_{\mathrm{L}} = \dfrac{g\,\rho_{\mathrm{L}}^2\, d^3}{\mu_{\mathrm{L}}^2}$, $H(\xi) = \xi + \left[10(1-\xi)^{0.1} - 8.9\right]\sqrt{\xi}\left(1 - \sqrt{\xi}\right)$, $Ph_{\mathrm{L}} = \dfrac{Cp_{\mathrm{L}}(T_{\mathrm{R}} - T_{\mathrm{wi}})}{\Delta h_{\mathrm{VL}}}$,

$Pr_{\mathrm{L}} = \dfrac{\mu_{\mathrm{L}}\, Cp_{\mathrm{L}}}{\lambda_{\mathrm{L}}}$, $Re_{\mathrm{L}} = \dfrac{G(1-x)d}{\mu_{\mathrm{L}}}$, $\mathrm{X}_{tt} = \left(\dfrac{1-x}{x}\right)^{0.9}\left(\dfrac{\rho_{\mathrm{V}}}{\rho_{\mathrm{L}}}\right)^{0.5}\left(\dfrac{\mu_{\mathrm{L}}}{\mu_{\mathrm{V}}}\right)^{0.1}$

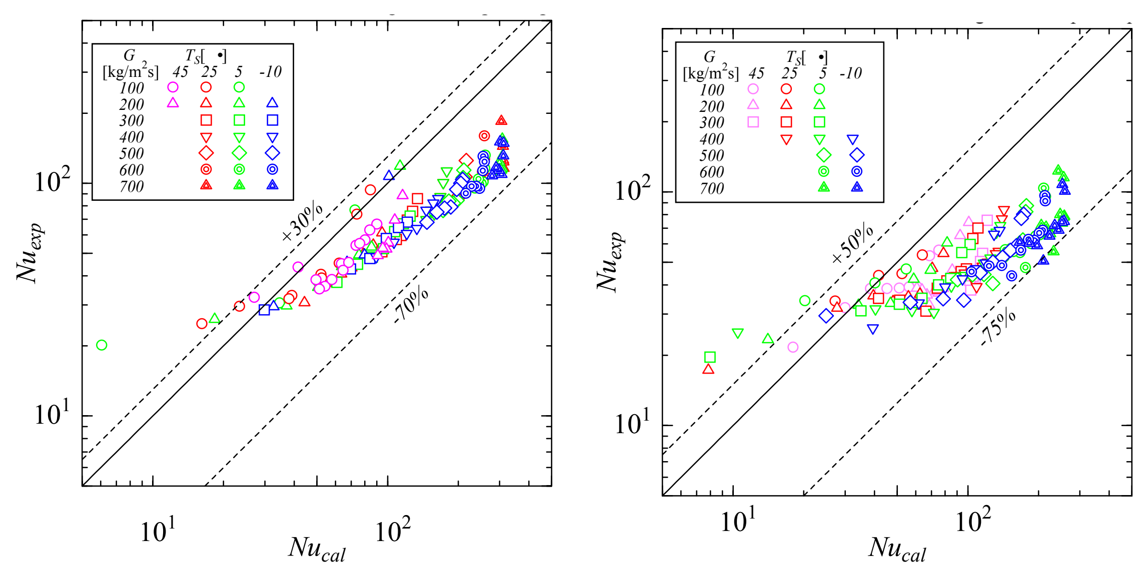
The Mishima-Hibiki correlation is based on the experimental data of air-water adiabatic two-phase flow. Therefore, in the present study, new collation, listed in Table 3, is proposed taking an account of the surface tension effect based on the Mishima-Hibiki equation. The comparison between the measured pressure drop and the pressure drop predicted using equation (7) is shown in Figure 4. The values of the predicted pressure drop agree well with measured ones.

Figure 5 shows the comparison between experimental data of local heat transfer and the correlation of Moser *et al*. (1998), which is proposed for the in-tube condensation heat transfer coefficient based on the data of relatively large diameter tubes of 4.57 - 12.7 mm I.D. It is noted that the experimental data cannot be compared with the correlation of Moser *et al*. originally because it is out of range. In the cases of high mass velocity, G =300 - 700



(a) Tube Type A (b) Tube Type B

Figure 5: Comparison between experimental data and correlation of Moser *et al.*

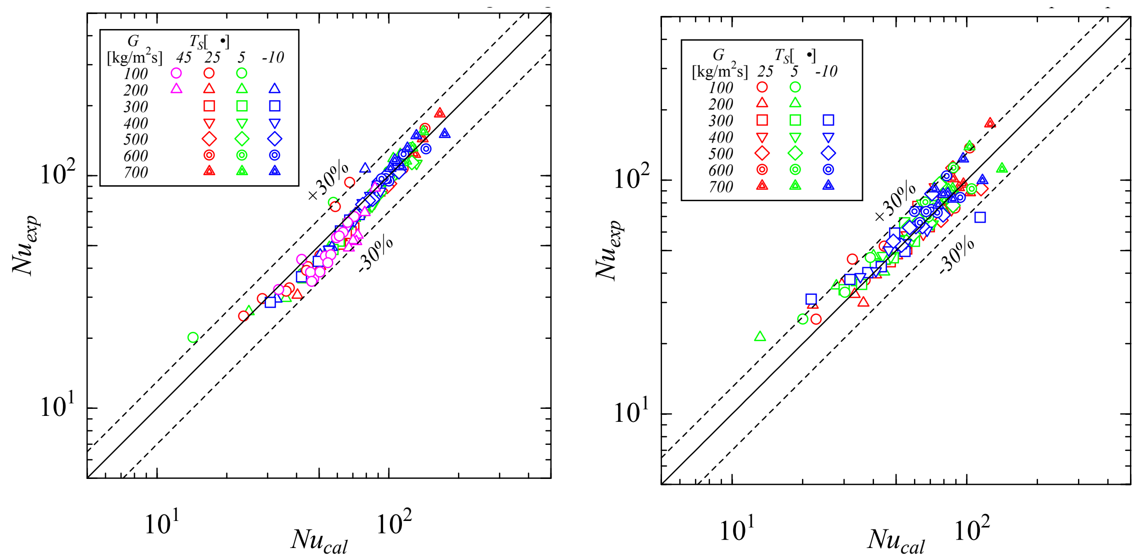


(a) Tube Type A (b) Tube Type B

Figure 6: Comparison between experimental data and correlation of Haraguchi *et al.*

[kg/m^2 s], most of the present data Nu_{exp} agree with the correlation within an error of $\pm$30%. However, in the cases of low mass velocity, G =100 and 200 [kg/m^2 s], the data of Nu_{exp} are higher than the predicted values Nu_{cal}. The reason is mainly due to neglecting the free convection effect in their correlation.

Figure 6 shows the comparison between the present experimental data and correlation of Haraguchi *et al.* (1994b). This correlation is proposed for in-tube condensation heat transfer coefficient based on the data of a relatively large diameter tube of 8.4 mm I.D; in this correlation, both effects of the forced convection and free convection are taken into account. The present data, Nu_{exp}, are lower than the predicted values, Nu_{cal}, and there is a different trend between the experimental data and the predicted values. The main reason of this difference is caused



(a) Tube Type A (b) Tube Type B

Figure 7: Comparison between experimental data and equation (8)

by the estimation of the forced convection term in their correlation. In the correlation of Haraguchi *et al*. (1994b), the forced convection term is calculated based on the frictional pressure drop correlation of Haraguchi *et al*. (1994a). However, their correlation for frictional pressure drop overpredicts the present experimental data extremely. In consequence, their correlation for heat transfer coefficient overpredicts the present experimental data. Therefore, in the present study, new heat transfer correlation, listed in Table 3, is proposed by modifying the effect of tube diameter in the correlation of Haraguchi *et al*.

Figure 7 shows the comparison between experimental data and prediction by equation (8). The agreement between the experiment and the prediction is fairly good.

CONCLUSIONS

The characteristics of pressure drop and heat transfer are experimentally investigated on the condensing two-phase flow of pure refrigerant R134a in two kinds of horizontal multi-port extruded tubes. The conclusions are as follows:

(1) The trend of the present experimental data of frictional pressure drop is the same as the Mishima-Hibiki correlation. However, correlations of Chisholm-Laird, Soliman *et al*. and Haraguchi *et al*. overpredict the experimental data. This result suggests that the effect of tube diameter should be included in the correlation of frictional pressure drop.

(2) The present experimental data except for cases of low mass velocity are relatively in good agreement with the heat transfer correlation of Moser *et al*., although it is proposed for the in-tube forced convective condensation based on the data of relatively large diameter tubes. It is inferred from this result that the effect of gravitational acceleration should be considered in the case of low mass velocity. The present experimental data were also compared with the heat transfer correlation of Haraguchi *et al*., in which both effects of the forced convection and the free convection are taken into account. However, the agreement between the experiment and the prediction is not so good. This reason may be caused mainly by the estimation of the forced convection term in their correlation.

(3) Considering the effects of surface tension and kinematic viscosity, new correlation of frictional pressure drop is developed based on the Mishima-Hibiki correlation. New correlation of heat transfer coefficient is also developed modifying the effect of diameter in the correlation of Haraguchi *et al*.

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