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An experimental study on condensation of refrigerant R134a in a multi-port extruded tube

Shigeru Koyama[a,*], Ken Kuwahara[a], Kouichi Nakashita[a], Ken Yamamoto[b]

[a]*Institute of Advanced Material Study, Kyushu University, Kasuga, Fukuoka 816-8580, Japan*
[b]*Air-Conditioning R&D Depart. 2, Denso Corporation, Kariya, Aichi 448-8661, Japan*

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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 pure refrigerant R134a in two kinds of 865 mm long multi-port extruded tubes having eight channels in 1.11 mm hydraulic diameter and 19 channels in 0.80 mm hydraulic diameter. The pressure drop is measured at an interval of 191 mm through small pressure measuring ports. The local heat transfer rate is measured in every subsection of 75 mm in effective cooling length using heat flux sensors. It is found that the experimental data of frictional pressure drop agree with the correlation of Mishima and Hibiki [Trans. JMSE (B) 61 (1995) 99], while the correlations of Chisholm and Laird [Trans. ASME 80 (1958) 227], Soliman et al. [Trans. ASME, Ser. C 90 (1998) 267], and Haraguchi et al. [Trans. JSME (B) 60 (1994) 239], overpredict. As a trial, the data of local heat transfer coefficient are also compared with correlations of Moser et al. [J. Heat Transfer 120 (1998) 410] and Haraguchi et al. [Trans. JSME (B) 60 (1994) 245]. The data of high mass velocity agree with the correlation of Moser et al., while those of low mass velocity show different trends. The correlation of Haraguchi et al. shows the trend similar to the data when the shear stress in their correlation is estimated using the correlation of Mishima and Hibiki.

Keywords: Heat transfer; Mass transfer; Condensation; Tube; Geometry; R134a; Heat transfer coefficient; Measurement

R134a : étude expérimentale sur la condensation dans un tube extrudé à passages multiples

Mots clés : Transfert de chaleur ; Transfert de masse ; Condensation ; Tube ; Géométrie ; R134a ; Coefficient de transfert de chaleur ; Mesure

1. Introduction

Environmental protection is one of the most crucial topics recently. From this point of view, it is urgently necessary for us to introduce environmentally acceptable new refrigerants and improve further the performance in the refrigeration and air-conditioning systems. One of the methods for improving the system performance could be to reduce the diameter of heat transfer tubes.

There are a few previous studies on the condensation heat transfer of refrigerants in small diameter tubes. Katsuta [1] carried out experiments of R134a in several 1000 mm long multi-port extruded tubes, and compared the local heat transfer characteristics with previous correlations proposed for large diameter tubes. Yang

* Corresponding author. Tel.: +81-92-583-7831; fax: +81-92-583-7833.
E-mail address: koyama@cm.kyushu-u.ac.jp (S. Koyama).

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Nomenclature

A	cooling area (m^2)
Cp	isobaric heat capacity (kJ kg^{-1} K^{-1})
d	inner hydraulic diameter (m)
G	mass velocity (kg m^{-2} s^{-1})
g	gravitational accelerator (m s^{-2})
Ga	Galileo number (–)
H	width of multi-port extruded tube (m)
$H(\xi)$	function of void fraction (–)
Nu	Nusselt number (–)
P	pressure (Pa)
Ph	phase change number (–)
Pr	Prandtl number (–)
q	heat flux (W m^{-2})
Re	Reynolds number (–)
S	wetted perimeter (m)
T	temperature (°C)
x	quality
α	heat transfer coefficient (W m^{-2} K^{-1})
Δh_{VL}	latent heat of condensation (kJ kg^{-1})
ΔP	pressure drop through Δz (Pa)
Δz	distance of neighboring pressure-measuring ports (m)
λ	thermal conductivity (W m^{-1} K^{-1})
μ	viscosity (Pa s)
ρ	density (kg m^{-3})
ξ	void fraction (–)
ϕ	two phase multiplier factor (−)
X_{tt}	Lockhart–Martinelli's parameter (−)

Subscripts

B	buoyancy convection
cal	calculated
exp	experiment
F	forced convection
in	inlet of section
L	liquid
Lo	liquid only
mix	mixing chamber
R	refrigerant
S	heat sink water
sat	saturation
V	vapor
wi	inner wall of tube
wo	outer wall of tube

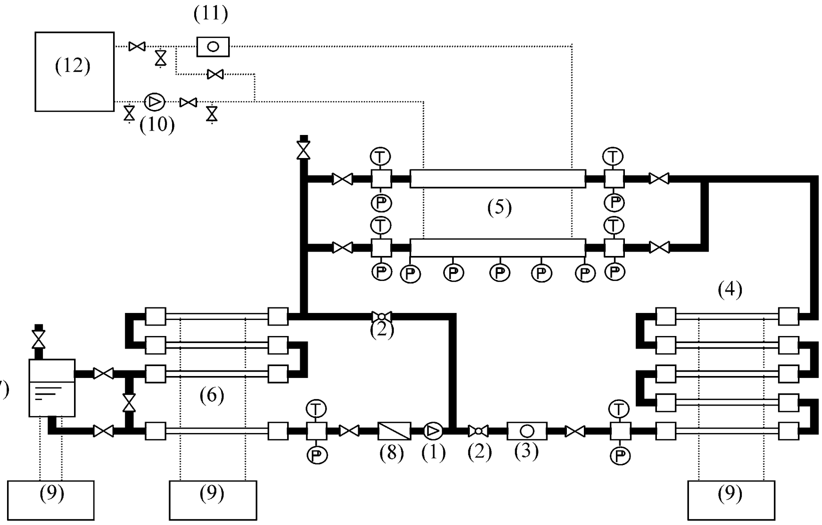
and Webb [2] carried out experiments on the heat transfer of R12 in a horizontal multi-port extruded tube of 2.64 mm in hydraulic diameter and a horizontal multi-port extruded fin tube of 1.56 mm in hydraulic diameter. They measured the average heat transfer coefficient through 508 mm long test tubes and indicated the effect of the average heat flux on the average heat transfer coefficient. Moser et al. [3] 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 in hydraulic diameter. However, it is very difficult to measure accurately the local heat transfer characteristics in a tube of around 1 mm or less than 1mm in hydraulic diameter using traditional methods such as the water calorimetric method, the Wilson-plot method. Accordingly, more research efforts are required to clarify the local characteristics of condensation process in a small diameter tube.

In the present study, the measurement of the local characteristics of pressure drop and heat transfer is carried out for the condensation of pure refrigerant R134a in two kinds of multi-port extruded tubes of around 1 mm in hydraulic diameter. Then, the experimental data are compared with previous correlations proposed for relatively large diameter tubes because of a lack of correlations for relatively small diameter tubes.

2. Experimental apparatus and method

Fig. 1 shows the schematic view of the experimental apparatus in the present study. The refrigerant liquid discharged from a gear pump (1) flows into an evaporator (4) through a mass flow meter (3). The refrigerant in the evaporator is heated by a constant temperature water bath (9). The refrigerant vapor generated at the evaporator flows into a test section (5). The refrigerant in the test section is cooled by a constant temperature brine bath (12). The refrigerant condensed in the test section returns to the pump through a subcooler (6) and a liquid receiver (7). In the liquid receiver, the refrigerant pressure level is controlled by a constant temperature water bath (9). The refrigerant flow rate is controlled by two manually operated flow control valves (2).

Fig. 2 shows a schematic view of the test section. The test section is composed of an inlet mixing chamber, a multi-port extruded tube, and an outlet mixing chamber. Two types of multi-port extruded tubes made of aluminum 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 a local value in the present study. The refrigerant temperature is measured with two 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



(1) Refrigerant pump (2) Flow control valve (3) Mass flow meter (4) Evaporator (5) Test section
(6) Subcooler (7) Liquid receiver (8) Filter (9) Constant temperature water bath
(10) Coolant pump (11) Volume flow meter (12) Constant temperature brine bath

Fig. 1. Schematic view of experimental apparatus.

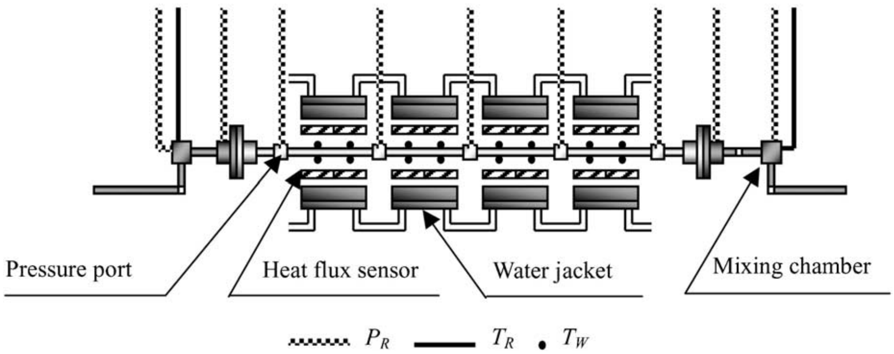


Fig. 2. Schematic view of test section.

thermocouples of 75 μm O.D. 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 nine pressure measuring ports using a differential pressure transducer.

Table 1 shows the dimensions of the test tubes. The dimensions of type A tube are: channel number 8, wetted perimeter length 43.59 mm, cross section area 12.14 mm^2, and hydraulic diameter 1.114 mm. The dimensions of type B tube are: channel number 19, wetted perimeter length 54.00 mm, cross section area 10.90 mm^2, and hydraulic diameter 0.807 mm. For reference, the photographs of the test tubes are shown in Fig. 3.

The experiments are carried out using pure refrigerant R134a as a test fluid. The experimental ranges are as follows: the mass velocity of $G = 100–700$ kg m^{-2} s^{-1} and vapor quality of $x = 1.0–0.0$ at a constant inlet pressure of 1.7 MPa.

In the data reduction process the following assumptions are employed:

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

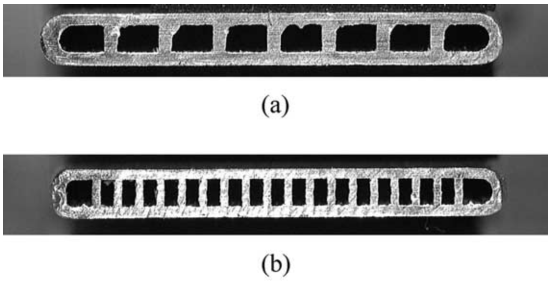


Fig. 3. Photographs of test tubes: (a) Tube type A; (b) Tube type B.

Table 1
Dimensions of multi-port extruded tube

Tube type	Type A	Type B
Channel number	8	19
Width H (mm)	15.0	15.0
Wetted perimeter length S (mm)	43.59	54.00
Cross sectional area (mm^2)	12.14	10.90
Hydraulic diameter d (mm)	1.114	0.807
Tube length (mm)	865.0	865.0
Effective heat transfer length (mm)	600.0	600.0

homogeneous 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 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 frictional pressure drop between the neighboring pressure measuring ports is obtained by subtracting the effect of momentum change from the measured pressure drop. Finally, the local heat transfer coefficient α, the local Nusselt number Nu and the two phase multiplier factor ϕ_V are obtained as,

$$\alpha = \frac{A_{\mathrm{wo}}}{A_{\mathrm{wi}}} \frac{q}{(T_{\mathrm{sat}} - T_{\mathrm{wi}})} = \frac{2H}{S} \frac{q}{(T_{\mathrm{sat}} - T_{\mathrm{wi}})} \tag{1}$$

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 based on the outer surface area of a subsection, H is the width of the test tube, S is the wetted perimeter length of the test tube, T_{sat} is the arithmetic mean of refrigerant temperature at the inlet and the outlet of a subsection, and T_{wi} is the inner wall temperature of the test tube,

$$Nu = \frac{\alpha d}{\lambda_{\mathrm{L}}} \tag{2}$$

where d is the hydraulic diameter of the test tube, and λ_{L} is the thermal conductivity of refrigerant liquid,

$$\phi_V = \sqrt{\frac{\Delta P_F/\Delta z}{\Delta P_V/\Delta z}} \tag{3}$$

where ΔP_{F} is the frictional pressure drop, ΔP_{V} is the pressure drop when only the vapor component flows in the test tube, and Δz is the distance between the neighboring pressure-measuring ports. Thermophysical properties in data reduction of each experiment are calculated using the REFPROP Version 6.0 [4].

3. Results and discussion

Fig. 4(a) and (b) show typical examples of distribution of temperature, pressure, heat flux, heat transfer coefficient and quality in cases of type B tube at $G=270$ and 650 kg m^{-2} s^{-1}, respectively. In the present experiments, the heat flux is maintained almost constant. The pressure drop in this tube is higher than that of relatively large diameter tube used normally in domestic air-conditioners. The value of heat transfer coefficient is the highest at a subsection containing the starting point of condensation, and decreases in the refrigerant flow direction gradually.

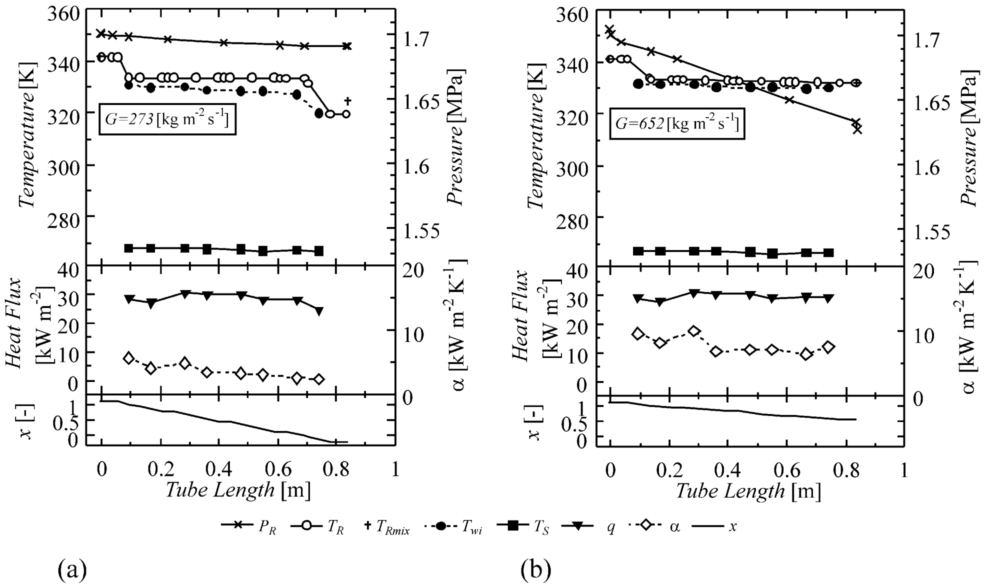


Fig. 4. Distribution of temperature, pressure, heat flux, heat transfer coefficient and quality: (a) Tube type B at $G = 270$ (kg m^{-2} s^{-1}); (b) Tube type B at $G = 650$ (kg m^{-2} s^{-1}).

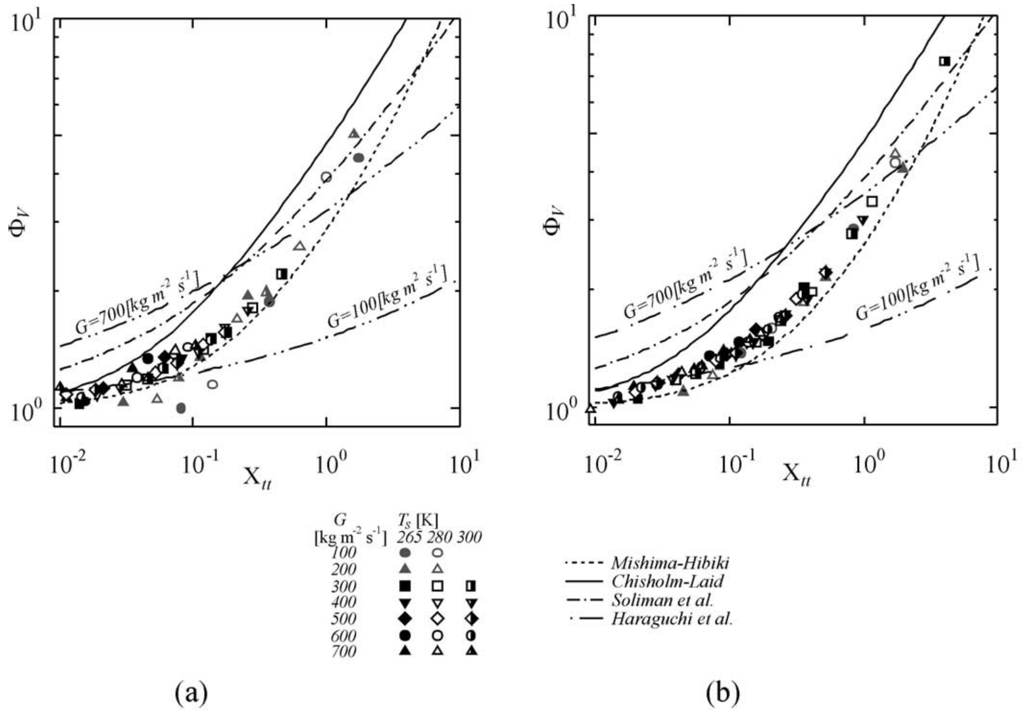


Fig. 5. Relation between the two-phase multiplier factor ϕ_V and Lockhart–Martinelli parameter X_{tt}: (a) Tube type A; (b) Tube type B.

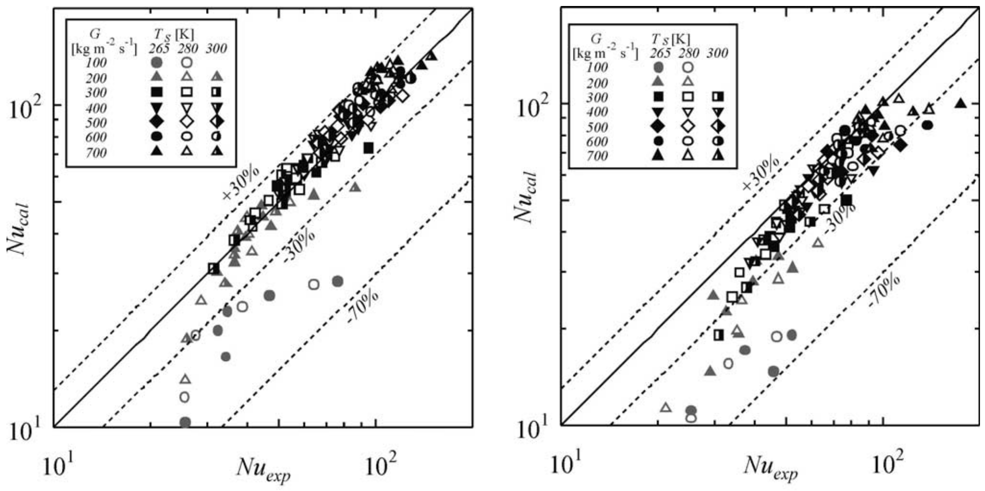
Fig. 5 shows the relation between the two-phase multiplier factor ϕ_V and the Lockhart–Martinelli parameter X_{tt}, where Fig. 5(a) and (b) are the results for test tubes of type A and type B, respectively. In both figures, correlations proposed by Mishima and Hibiki [5], Chisholm-Laird [6], Soliman et al. [7], and Haraguchi et al. [8] are represented by a dashed line, a solid line, a chain line and a double-dotted chain line, respectively. The experimental data in both tubes agree well with Mishima–Hibiki correlation in which the effect of tube diameter is considered as

$$\phi_{\mathrm{V}} = 1 + 21(1 - e^{-0.319d})X_{\mathrm{tt}} + X_{\mathrm{tt}}^2 \quad (4)$$

However, 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.

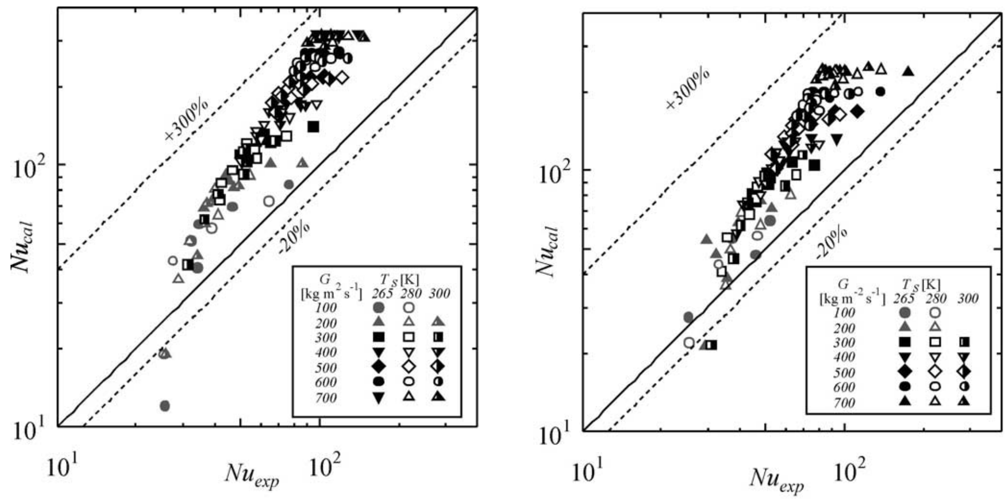
Fig. 6 shows the trial comparison between experimental data of local heat transfer and the correlation of Moser et al. [3], which is proposed for the in-tube condensation heat transfer coefficient based on the data of relatively large diameter tubes of 4.57–12.7 mm I.D. It is noted that the present experimental 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 kg m^{-2} s^{-1}, 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^{-1}, 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.

Fig. 7 show the trial comparison between the present experimental data of local heat transfer and the correlation of Haraguchi et al. [9]. This correlation is



(a) (b)

Fig. 6. Comparison between experimental data and correlation of Moser et al.: (a) Tube type A; (b) Tube type B.



(a) (b)

Fig. 7. Comparison between experimental data and correlation of Haraguchi et al.: (a) Tube type A; (b) Tube type B.

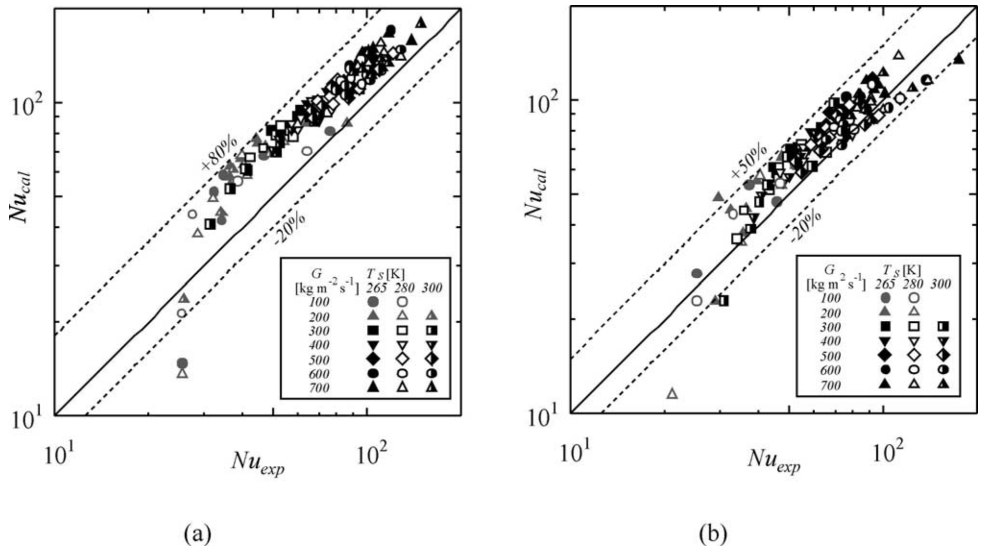


Fig. 8. Comparison between experimental data and the present correlation (a) Tube type A, (b) Tube type B.

Table 2
Tentative correlation based on the correlations of Haraguchi et al. and Mishima–Hibiki

Condensation heat transfer

$Nu = (Nu_F^2 + Nu_B^2)^{1/2}$

(1) Forced convection condensation term

$$Nu_F = 0.0152(1 + 0.6Pr_L^{0.8})\left(\frac{\phi_V}{X_{tt}}\right)Re_L^{0.77}$$

where

$$\phi_V^2 = 1 + 21(1 - e^{-0.319d})X_{tt} + X_{tt}^2$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_V}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_V}\right)^{0.1}$$

$$Re_L = \frac{G(1-x)d}{\mu_L}$$

(2) Gravity controlled convection condensation term

$$Nu_B = 0.725H(\xi)\left(\frac{GaPr_L}{Ph}\right)^{1/4}$$

where

$$H(\xi) = \xi + \left\{10\left[(1-\xi)^{0.1} - 1\right] + 1.7 \times 10^{-4}Re_{Lo}\right\}\sqrt{\xi}(1 - \sqrt{\xi})$$

$$Ga = \frac{g\rho_L^2 d^3}{\mu_L^2}$$

$$Ph = \frac{Cp_L(T_{sat} - T_{wi})}{\Delta h_{VL}}$$

$$Re_{Lo} = \frac{Gd}{\mu_L}$$

$$\xi = \left[1 + \frac{\rho_V}{\rho_L}\left(\frac{1-x}{x}\right)\left(0.4 + 0.6\sqrt{\frac{\frac{\rho_L}{\rho_V} + 0.4\frac{1-x}{x}}{1 + 0.4\frac{1-x}{x}}}\right)\right]^{-1}$$

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. It is also noted that the present experimental data cannot be compared with the correlation of Haraguchi et al. originally because it is out of range. 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 by the estimation of the forced convection term in their correlation. In the heat transfer correlation of Haraguchi et al., the forced convection term is calculated based on the frictional pressure drop correlation of Haraguchi et al. However, their correlation for frictional pressure drop overpredicts the present experimental data extremely, as shown in Fig. 5. In consequence, their correlation for heat transfer coefficient overpredicts the present experimental data.

As a trial, the frictional pressure drop correlation of Haraguchi et al. [8] used in the heat transfer correlation of Haraguchi et al. [9] is replaced by the Mishima and Hibiki correlation [5], because the Mishima–Hibiki correlation is in good agreement with the present experimental data. The results are shown in Fig. 8. This modification of the heat transfer correlation of Haraguchi et al. leads better agreement between the correlation and the experimental data. This correlation is summarized in Table 2.

4. 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 present experimental data of frictional pressure drop agree well with 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. In the present data reduction, the pressure drop due to the momentum change is estimated using the homogeneous model. However, to estimate such effect, flow patterns should be confirmed at first.
2. The present experimental data except for cases of low mass velocity are in relatively 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.
3. 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. As a trial, replacing the frictional pressure drop correlation used in the heat transfer correlation of Haraguchi et al. by the Mishima–Hibiki correlation leads to better agreement between the experiment and the prediction.

To establish a prediction method of the pressure drop and heat transfer characteristics of pure refrigerant condensing in a small diameter tube, experimental data should be accumulated further and the following terms should be investigated as future works; (1) flow patterns, (2) the effect of tube diameter, and (3) the interaction effect between the vapor shear stress and the gravitational acceleration and the surface tension.

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