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Boiling heat transfer of HFO-1234yf flowing in a smooth small-diameter horizontal tube

Shizuo Saitoh[a*], *Chaobin Dang*[b], *Yoshitaka Nakamura*[b], *Eiji Hihara*[b]

[a]*Department of Mechanical Engineering, The University of Tokyo,* 7-3-1, Bunkyo-ku, Tokyo 113-8656, Japan
[b]*Institute of Environmental Studies, Graduate School of Frontier Sciences, The University of Tokyo,* 5-1-5 Kashiwanoha, Kashiwa-shi, Chiba, 277-8563, Japan
*Corresponding author. E-mail: [saitoh@hee.k.u-tokyo.ac.jp] Tel./Fax: [+81 3 5841 6325]

Abstract
The flow boiling heat transfer coefficient of the low-GWP (global warming potential) refrigerant HFO-1234yf inside a smooth small-diameter horizontal tube (inner diameter: 2 mm) was experimentally investigated. The local heat transfer coefficient was measured at heat fluxes of 6–24 kW m^{-2}, mass fluxes of 100–400 kg m^{-2} s^{-1}, evaporating temperature of 288.15 K, and inlet vapor quality of 0–0.25. The results show that the effect of heat flux on the heat transfer was large at low vapor quality, while the effect of mass flux was large at high vapor quality. The heat transfer coefficient of HFO-1234yf was almost the same as that of R-134a. The heat transfer coefficients calculated based on correlations with Saitoh et al. agreed well with the measured values compared to other correlations. The measured pressure drop agreed well with that predicted by the Lockhart-Martinelli correlation.

Keyword: Boiling heat transfer, Pressure drop, HFO-1234yf, Small-Diameter tube

1. Introduction

The release of the MAC (mobile air conditioning) directive by the EU, which bans the use of refrigerants with global warming potential (GWP) above 150 in new types of mobile air conditioning from 2011 in the EU market, has triggered the research and development in a search for new refrigerants. Carbon dioxide is considered to be a promising candidate. Recently, HFO-1234yf, which is another promising candidate, was jointly developed by Honeywell and DuPont. Because the GWP of HFO-1234yf is as low as 4 and its thermophysical properties are similar to those of R-134a, it is expected to be a drop-in solution for current mobile air conditioners. Several experimental studies have been conducted on its thermophysical properties and cycle performance to estimate the feasibility of using this new refrigerant in mobile air conditioners. Increasing concern due to environmental predictions has led to the reconsideration of refrigerants in other applications.

One approach is to use a refrigerant mixture of HFO-1234yf + HFC-32 to obtain a high system coefficient of performance (COP). The thermophysical properties of the refrigerant mixture are also being evaluated (i.e., Arakawa et al. 2010).

The system performance of an actual heat pump is lower than that of the theoretical cycle because of heat transfer loss inside the heat exchangers and pressure drop along the duct. Therefore, the heat transfer performance must be analyzed to evaluate the system performance of an actual heat pump system when a new refrigerant is considered as well as when designing heat exchangers. In this study, the boiling heat transfer of the refrigerant HFO-1234yf flowing in a smooth small-diameter horizontal tube (inner diameter (ID): 2 mm) was experimentally investigated. The measured local heat transfer coefficient of HFO-1234yf was compared with that of R-134a, and a prediction method for the evaporation heat transfer coefficient of HFO-1234yf is discussed.

2. Experimental apparatus and procedure

Figure 1 shows a schematic of the experimental system used to measure the heat transfer coefficient and pressure drop and observe the flow patterns of HFO-1234yf. The test loop includes a Coriolis-type flow meter, refrigerant temperature controller, flow control valve, test tube (evaporator), and sight glass. The purity of the HFO-1234yf used was over 99.7%. To reduce the heat loss from the test tube to the environment, the entire test tube was placed inside an air duct with the air temperature controlled to be equal to the evaporating temperature. The flow rate and inlet pressure of HFO-1234yf were controlled by adjusting the frequency of the magnetic gear pump and opening of the flow control valve. The vapor quality at the inlet of the test evaporator was adjusted by the amount of heat supplied to the refrigerant in the pre-heater. A sub tank was used to adjust the amount of refrigerant in the test loop. When the performance of the condenser was poor, the evaporation pressure was controlled by the amount of refrigerant. Figure 2 shows the measurement points for outer surface temperature and pressure with the 2 mm ID test tube, and thermocouples attached to the tube. Table 1 lists the specification of the test tube and measurement interval of temperature and pressure. The test tube was heated by direct electrification using a DC power supply connected to two electrodes soldered at the flanges of the two ends of the test tube. The pressures of the refrigerant in the test tube were measured using a precision aneroid manometer. The temperatures of the outer surface of the test tube were measured midway between the top and bottom of the tube along the surface using T-type thermocouples (outer diameter (OD): 0.1 mm), and the temperatures of the inner wall of the tube were calculated from the measured temperatures of the outer wall of the tube using Fourier’s law. An 8 μm thick Teflon sheet was inserted between each thermocouple and the test tube to prevent the current from affecting the thermocouples.

All the thermocouples were calibrated by using a high-precision platinum resistance thermometer sensor (Chino, Model CNA) with an accuracy of ±0.03 K. The accuracy of the calibrated thermocouples was within ±0.1 K. The mass flow rate (and thus the mass flux *G*) was measured by using the Coriolis-type flow meter (Oval, Model E010S-IN-200) with an accuracy of ±0.1%. The pressure was measured using the precision aneroid manometer (Nagano Keiki, Model NKS) with an accuracy of ±1.5 kPa. The electrical input power was measured using a voltmeter and ammeter to confirm that the heat generated by direct electrification was transferred well to the fluid and that the heat gain from surroundings was within 3%. The experimental conditions are summarized in Table 2. To compare accumulated data of boiling heat transfer for R-134a, the evaporation temperature is chosen to be 288.15K as used in previous research for R-134a. The local heat transfer coefficient h_{exp} in the test tube was determined using the following equation:

$$h_{\mathrm{exp}} = \frac{q}{T_{\mathrm{wall}} - T_{\mathrm{sat}}} \tag{1}$$

where T_{wall} is the temperature of the inner wall and T_{sat} is the saturation temperature at the local refrigerant pressure calculated by interpolation of the adjoined pressure gauges. All experimental data were collected after the steady state was reached for temperature, pressure, and refrigerant flow.

In this study, the saturated vapor pressure of HFO-1234yf was correlated using experimental data from Tanaka and Higashi (2010), Nicola et al. (2010) and Hulse et al. (2009); thermodynamic properties at the saturation state were calculated using Akasaka et al.'s method (Akasaka et al., 2010). Thermal conductivities of liquid and vapor were calculated using Latini et al. and Chung et al.'s methods, respectively (Poling et al., 2001). Liquid and vapor viscosities were determined using Hulse's et al. correlation (2009) and the Chapman-Enskog equation (Poling et al., 2001), respectively. The properties of the refrigerant R-134a were calculated using REFPROP version 8.0 (Lemmon et al., 2007).

3. Results and discussion

3.1 Flow boiling heat transfer coefficient

3.1.1 Effect of heat flux

Figure 3 shows the variation in the heat transfer coefficient against the vapor quality. The mass flux was kept at 200 kg m^{-2} s^{-1}; we compared the measured results for three different heat fluxes: 6, 12, and 24 kW m^{-2}. At the lowest heat flux of 6 kW m^{-2}, the measured heat transfer coefficient increased with the vapor quality, showing that the convective heat transfer intensifies with increasing quality. The dryout quality was about 0.8 and did not change with heat flux. Increasing the heat flux from 6 kW m^{-2} to 12 and 24 kW m^{-2} showed that the heat transfer coefficient increases with heat flux at low vapor quality; thus, nucleate boiling is the dominant heat transfer coefficient mechanism at low vapor quality.

3.1.2 Effect of mass flux

Figure 4 shows the effect of mass flux on the boiling heat transfer at heat flux of 12 kW m^{-2}. The dryout occurs at vapor quality of 0.8 for all the conditions. In the high-quality region

(>0.4), the heat transfer coefficients at both mass fluxes (200 and 400 kg m^{-2} s^{-1}) increased with increasing vapor quality, and the heat transfer coefficient was higher at 400 kg m^{-2} s^{-1} than at 200 kg m^{-2} s^{-1}. At a mass flux of 100 kg m^{-2} s^{-1}, the effect of vapor quality on the heat transfer coefficient was weak. The results suggest that in the high vapor quality region, forced convective evaporation is dominant.

3.1.3 Comparison between HFO-1234yf and R-134a

Figure 5 shows a comparison between the boiling heat transfer performances of HFO-1234yf and R-134a at a mass flux of 300 kg m^{-2} s^{-1} and heat flux of 12 kW m^{-2}. The figure shows that in the wide vapor quality region, the difference between the heat transfer coefficients of HFO-1234yf and R-134a is small, which may be because the differences in their thermodynamic properties are small. Saturation properties of HFO-1234yf and R-134a at temperature of 288.15K are shown in table 3. The properties of HFO-1234yf and R-134a are calculated following Brown et al.(2009) and using REFPROP ver.8, respectively. The gas density, latent heat, liquid thermal conductivity, and surface tension of HFO-1234yf and R-134a at 288.15 K were 26.3 and 23.76 kg m^{-3}, 156.5 and 186.59 kJ kg^{-1}, 0.0724 and 0.0854 W m^{-1} K^{-1}, and 0.0077 and 0.0094 N m^{-1}, respectively. In the following, the heat transfer coefficient of HFO-1234yf and that of R-134a is compared using the prediction model by considering the contribution of nucleate boiling and convective heat transfer. The heat transfer coefficient due to nucleate boiling is evaluated by Stephan-Abdelsalam (1980) correlation, and that of forced convection for liquid alone in the tube 2 mm ID is evaluated by Dittus-Boelter equation. At a temperature of 288.15K, mass flux of 300 kg m^{-2} s^{-1} and heat flux of 12 kW m^{-2}, the heat transfer coefficients (HTCs) of nucleate boiling and forced convection for HFO-1234yf are 2.44 kW m^{-2} K^{-1} and 0.89 kW m^{-2} K^{-1}, respectively, and that for R-134a are 2.30 kW m^{-2} K^{-1} and 0.92 kW m^{-2} K^{-1}. The difference between the HTCs of HFO-1234yf and that of R-134a is small.

3.1.4 Boiling heat transfer coefficient vs. Lochart-Martinelli parameter

In general, the flow boiling heat transfer in a tube is considered to be a combination of nucleate boiling heat transfer and forced convective evaporation. In forced convective evaporation, flow boiling data can be correlated with the form $h_{\exp} \propto h_{\mathrm{L}} \left(1/X\right)^{n}$, where h_{L} is the heat transfer coefficient of the liquid alone and X is the Lockhart-Martinelli parameter. When the superficial liquid Reynolds number, Re_l, is smaller than 1000, flow in the liquid phase is laminar, and $h_{\mathrm{L}} = 4.36 \lambda_l / D$. When the Re_l is larger than 1000, liquid flow is turbulent and the h_{L} can be calculated by Dittus-Boelter equation. The magnitude of forced convective evaporation can be expressed by the gradient n of a linear regression of the experimental data. Figure 6 shows the measured heat transfer coefficients in the pre-dryout region plotted against $1/X$. Most of the present data can be fitted to the regression line

$h_{\exp} / h_L \propto (1/X)^{0.77}$. The scattering of the data is larger at low $1/X$ (<7) than at high $1/X$. At low $1/X$, i.e., low vapor quality, both nucleate boiling and forced convective evaporation occur, whereas at high $1/X$ (>7), the scattering of the data is small because forced convective evaporation is dominant. When flow in the liquid phase is laminar, the heat transfer coefficient is barely influenced by the Lockhart-Martinelli parameter.

3.1.5 Pre-dryout heat transfer and predictions of some correlations

Saitoh et al. (2005, 2007) experimentally studied the boiling flow heat transfer mechanism of R-134a in tubes with ID of 0.51, 1.12, and 3.1 mm; the Chen-type correlation was modified by considering the effect of the tube diameter characterized by the Weber number in the gas phase.

For the pre-dryout heat transfer, the experimental data and predictions of some correlations were compared. These included Saitoh et al.'s, Yoshida et al.'s (1994), Kandlikar's (1990), and Gungor-Winterton (Gungar and Winterton, 1987) correlations. Details on these correlations are shown in Table 4. The properties of the refrigerant HFO-1234yf were calculated using the results reported by Brown et al. (2009). Figures 7(a–d) show a comparison between the experimental heat transfer coefficient h_{exp} and calculated heat transfer coefficient h_{cal} based on our modified Chen-type correlation (Saitoh et al., 2007) with others for HFO-1234yf in the 2 mm ID smooth horizontal tubes. h_{cal} based on the Saitoh et al. correlation agreed well with h_{exp}. However, h_{cal} calculated based on the Yoshida, Kandlikar, and Gungor-Winterton correlations did not agree well with h_{exp}. Figure 7(e) and (f) show the comparison of four correlations and measured heat transfer coefficient against vapor quality. In the pre-dryout region, the predicted values by all the four correlations increase with vapor quality. In the low quality region (x<0.5), the predicted values of Saitoh et al., Kandlikar and Gugor-Winterton are close to the measured values except for that of Yoshida. In the high quality region (0.5<x<0.75), the difference of the predicted values of Kandlikar correlation and Gungor-Winterton correlation with measured results are large. The predicted value by the correlation of Saitoh et al., which was proposed for flow boiling heat transfer of R-134a, approximately coincides with the measured value of HTC for HFO-1234yf. Table 5 lists the mean deviation and accuracy (defined as the fraction of data within ±20% error) for each of the three correlations. The Saitoh et al. correlation showed an improved mean deviation and accuracy (9.2% and 92.8%, respectively) compared to the Yoshida (20.4% and 51.9%), Kandlikar (18.7% and 51.9%), and Gungor-Winterton (15.5% and 66.9%) correlations.

3.2 Pressure drop

The measured two-phase pressure drop (Δp) was compared with that predicted results using the Lockhart - Martinelli correlation, which defines the pressure drop as.

$$-\left(\frac{dp}{dz}\right) = -\left(\frac{dp}{dz}\right)_{\mathrm{l}} \phi_{\mathrm{l}}^2 = -\left(\frac{dp}{dz}\right)_{\mathrm{g}} \phi_{\mathrm{g}}^2 \tag{2}$$

where ϕ_{l} and ϕ_{g} are the two-phase multipliers in the liquid and gas phases, respectively. They are defined as $\phi_{\mathrm{l}}^2 = 1 + c/X + 1/X^2$ and $\phi_{\mathrm{g}}^2 = 1 + cX + X^2$ (Chisholm, 1967), where X is the Lockhart-Martinelli parameter, which is the square root of the ratio between the pressure drop assuming liquid flow alone and assuming gas flow alone. When flows of liquid and gas phases are turbulent, c = 20; when liquid phase is laminar and gas phase is turbulent, c = 12. In this study, the flow condition of the refrigerant was determined at the inlet of the evaporator. Figures 8 shows the measured pressure drops and those predicted using the Lockhart-Martinelli correlation. The measured pressure drops agreed well with that predicted using the Lockhart-Martinelli correlation.

4. Conclusions

The flow boiling heat transfer of the refrigerant HFO-1234yf in a small-diameter horizontal tube was experimentally investigated. The local heat transfer coefficients were measured at mass fluxes of 100–400 kg m^{-2} s^{-1}, heat fluxes of 6–24 kW m^{-2}, and an evaporating temperature 288.15 K. The results are summarized as follows.

1. At low vapor quality, nucleate boiling heat transfer is the dominant heat transfer mechanism, and at high vapor quality, forced convective evaporation is dominant.
2. The boiling heat transfer coefficient of HFO-1234yf is almost the same as that of R-134a.
3. The heat transfer coefficient predicted by the Saitoh et al. correlation in the pre-dryout region agrees with the measured data within a range of ±20%, while the measured pressure drops agreed well with that predicted using the Lockhart-Martinelli correlation.

Nomenclature

Bo	boiling number
C_{g}	friction factor for gas, C_{g} = 0.046
C_{l}	friction factor for liquid, C_{l} = 16
$c_{p\mathrm{l}}$	specific heat at constant pressure in liquid phase, J kg^{-1} K^{-1}
D	inner diameter of a tube, m
G	mass flux, kg m^{-2}s^{-1}
g	acceleration of gravity, m s^{-2}
h_{exp}	experimental boiling heat transfer coefficient, W m^{-2} K^{-1}
h_{L}	liquid phase heat transfer coefficient, W m^{-2} K^{-1}

$$\mathrm{Re}_{\mathrm{l}} > 1000,\ h_{\mathrm{L}} = 0.023\frac{\lambda_{\mathrm{l}}}{D}\left(\frac{G_{\mathrm{l}}D}{\mu_{\mathrm{l}}}\right)^{0.8}\left(\frac{c_{p\mathrm{l}}\mu_{\mathrm{l}}}{\lambda_{\mathrm{l}}}\right)^{1/3}$$

$$\mathrm{Re}_{\mathrm{l}} < 1000,\ h_{\mathrm{L}} = \frac{4.36\lambda_{\mathrm{l}}}{D}$$

p pressure, Pa

q heat flux, W m^{-2}

Re Reynolds number

$$\mathrm{Re}_{\mathrm{l}} = \frac{G_{\mathrm{l}} D}{\mu_{\mathrm{l}}}, \quad \mathrm{Re}_{\mathrm{g}} = \frac{G_{\mathrm{g}} D}{\mu_{\mathrm{g}}}$$

T_{wall} inside-wall temperature, K

T_{sat} saturation temperature, K

We Weber number

x vapor quality

X Lockhart-Martinelli parameter

$\mathrm{Re}_{\mathrm{l}} > 1000, \quad \mathrm{Re}_{\mathrm{g}} > 1000$

$$X = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_{\mathrm{g}}}{\rho_{\mathrm{l}}}\right)^{0.5} \left(\frac{\mu_{\mathrm{l}}}{\mu_{\mathrm{g}}}\right)^{0.1}$$

$\mathrm{Re}_{\mathrm{l}} < 1000, \ \mathrm{Re}_{\mathrm{g}} > 1000$

$$X = \left(\frac{C_{\mathrm{l}}}{C_{g}}\right)^{0.5} \mathrm{Re}_{\mathrm{g}}^{-0.4} \left(\frac{G_{\mathrm{l}}}{G_{\mathrm{g}}}\right)^{0.5} \left(\frac{\rho_{\mathrm{g}}}{\rho_{\mathrm{l}}}\right)^{0.5} \left(\frac{\mu_{\mathrm{l}}}{\mu_{\mathrm{g}}}\right)^{0.5}$$

z coordinate along the tube direction, m

Greek symbols

λ thermal conductivity, W m^{-1} K^{-1}

μ viscosity, Pa·s

ρ density, kg m^{-3}

σ surface tension, N m^{-1}

ϕ two-phase flow multiplier

Subscripts

g gas-phase, vapor-phase

L,l liquid-phase

TP two-phase

Acknowledgment

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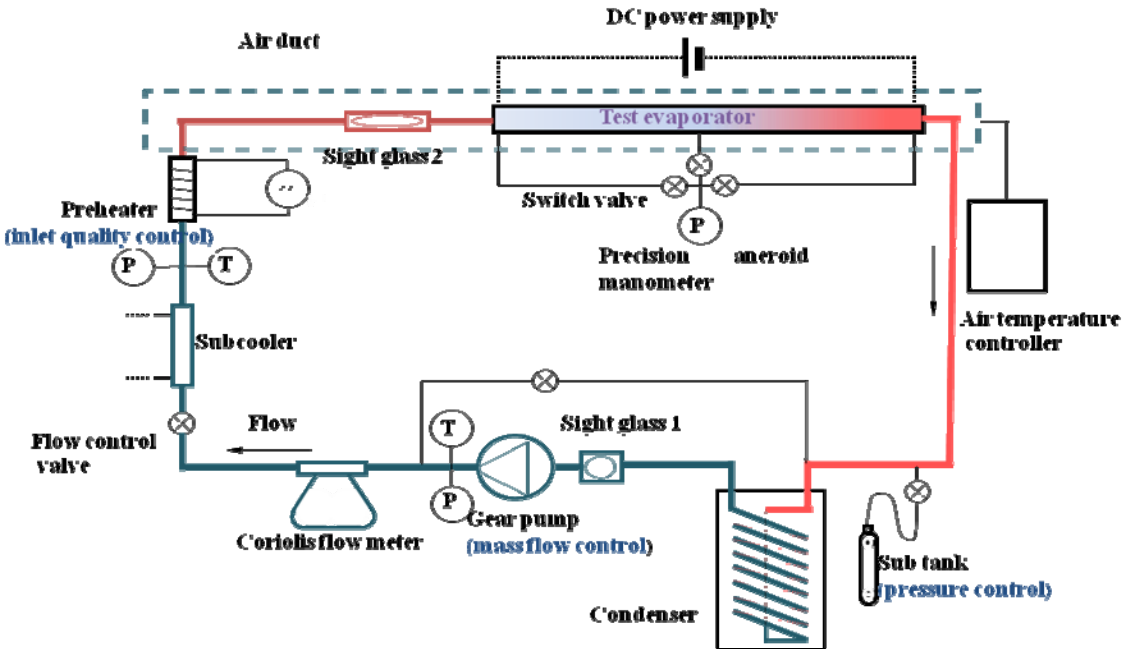
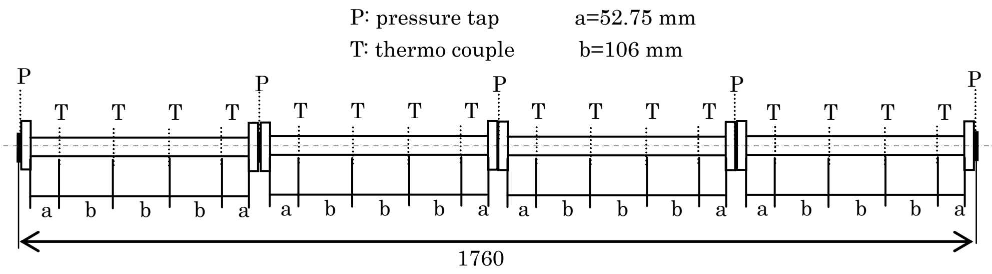
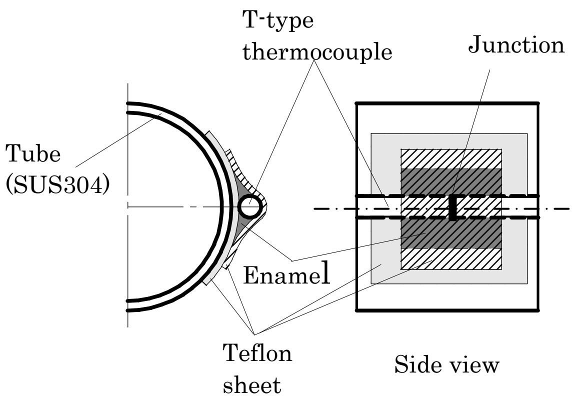


Figure 1. Schematic of experimental system used to measure flow boiling heat transfer.



Test section



Thermocouple attachment

Figure 2. Schematic of 2 mm ID test tube.

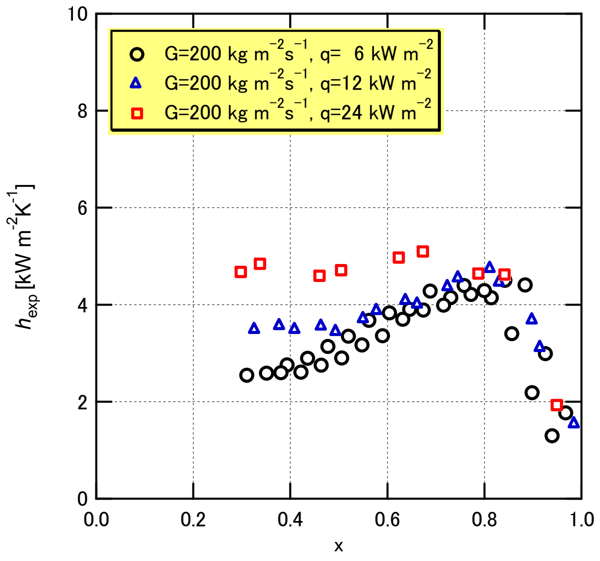


Figure 3. Effect of heat flux on local heat transfer coefficient.

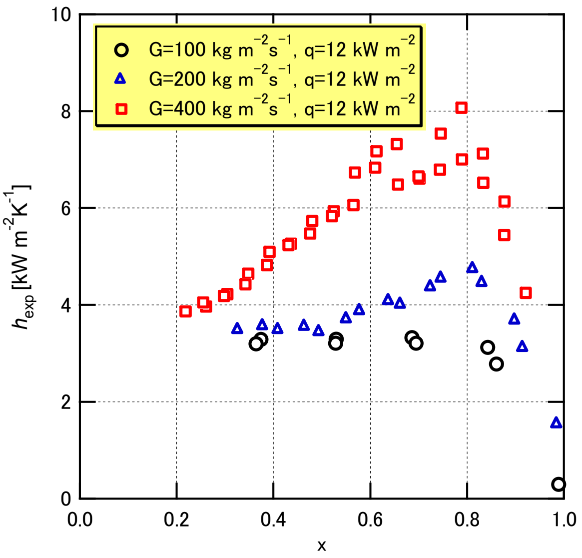


Figure 4. Effect of mass flux on local heat transfer coefficient.

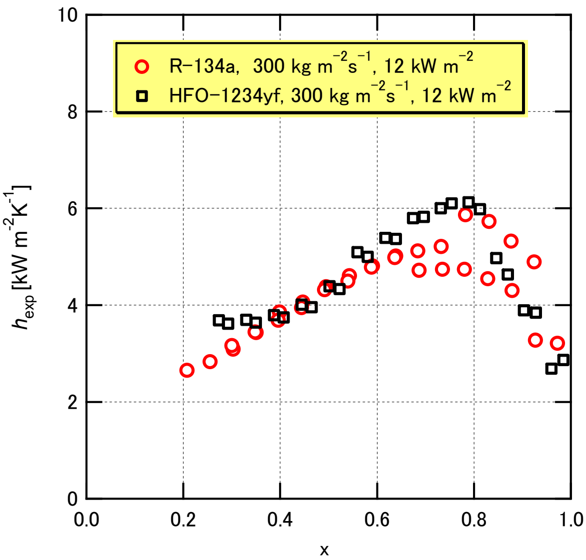


Figure 5. Comparison of heat transfer coefficients between HFO-1234yf and R-134a at mass flux of 300 kg m^{-2} s^{-1} and heat flux of 12 kW m^{-2}.

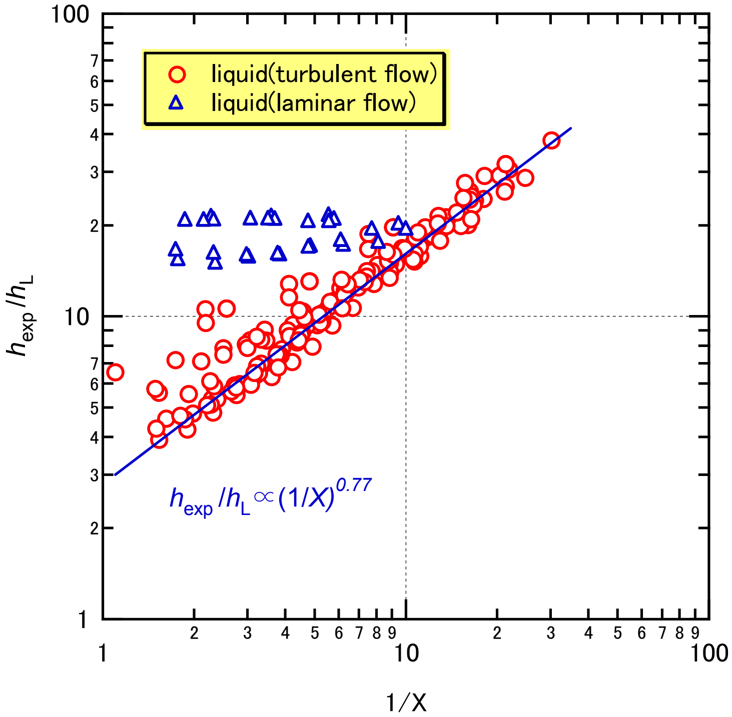
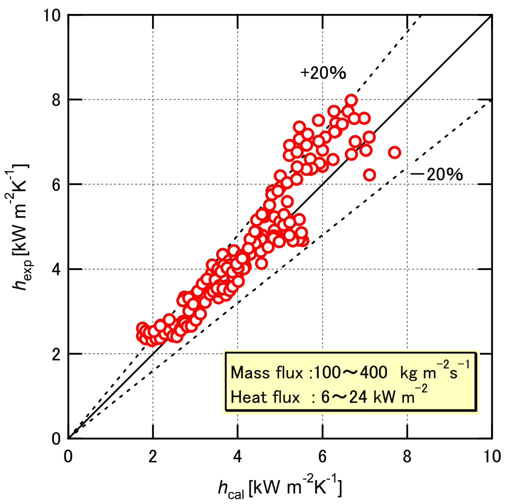
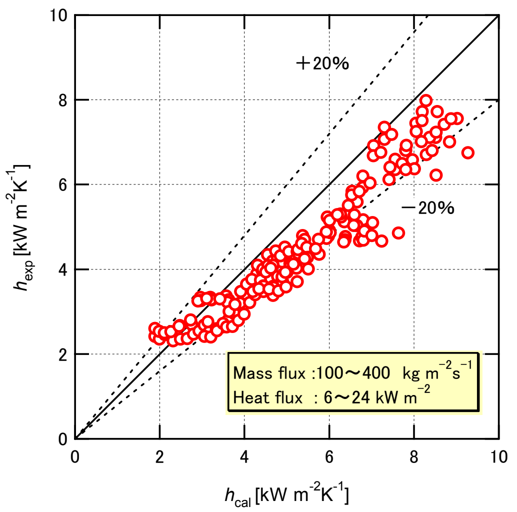


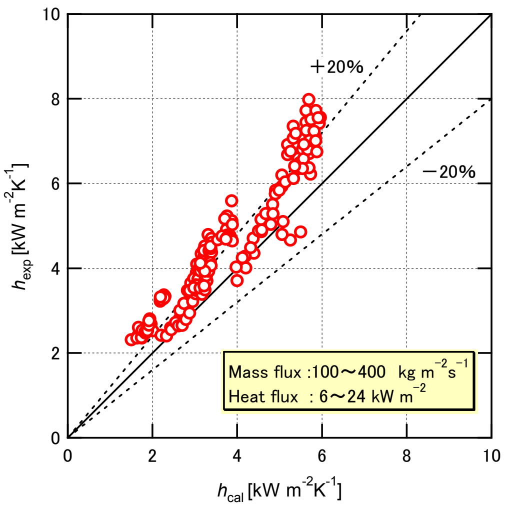
Figure 6. Boiling heat transfer coefficient as function of Lockhart-Martinelli parameter



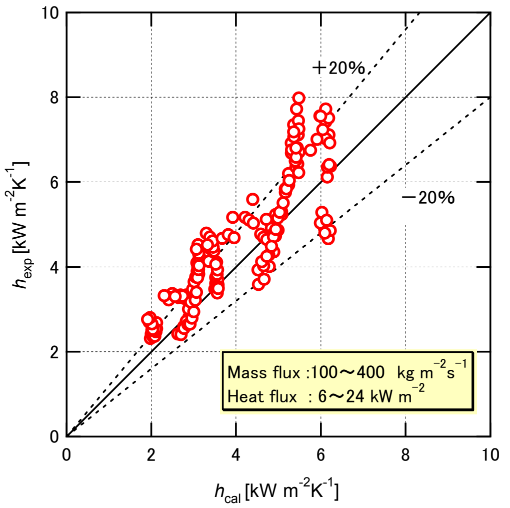
(a) Correlation of Saitoh et al.



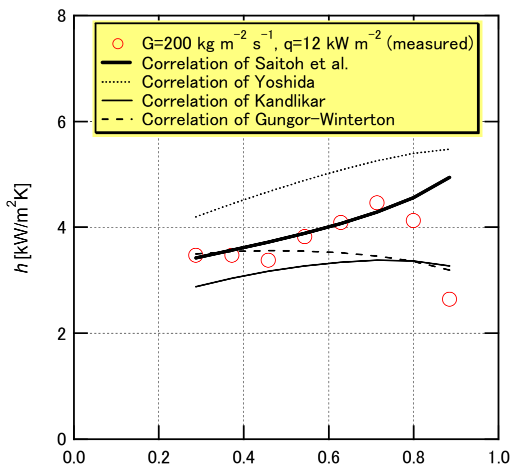
(b) Correlation of Yoshida



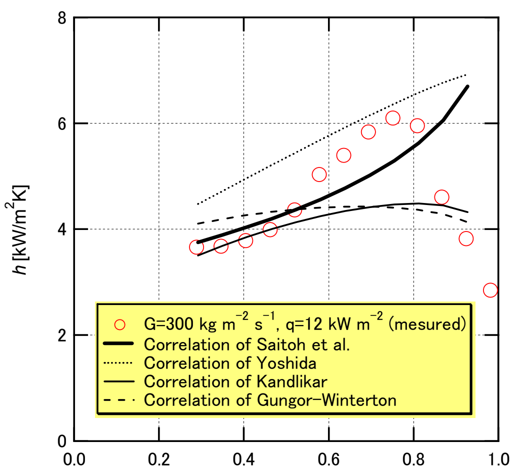
(c) Correlation of Kandlikar



(d) Correlation of Gungor-Winterton



(e) Predicted and measured values for flux 200kg m^{-2} s^{-1} and heat flux 12kW m^{-2}



(f) Predicted and measured values for mass flux 300kg m^{-2} s^{-1} and heat flux 12kW m^{-2}

Figure 7. Experimental flow boiling heat transfer coefficient h_{exp} vs. calculated h_{cal} for HFO-1234yf, and comparison of four predicted and measured data.

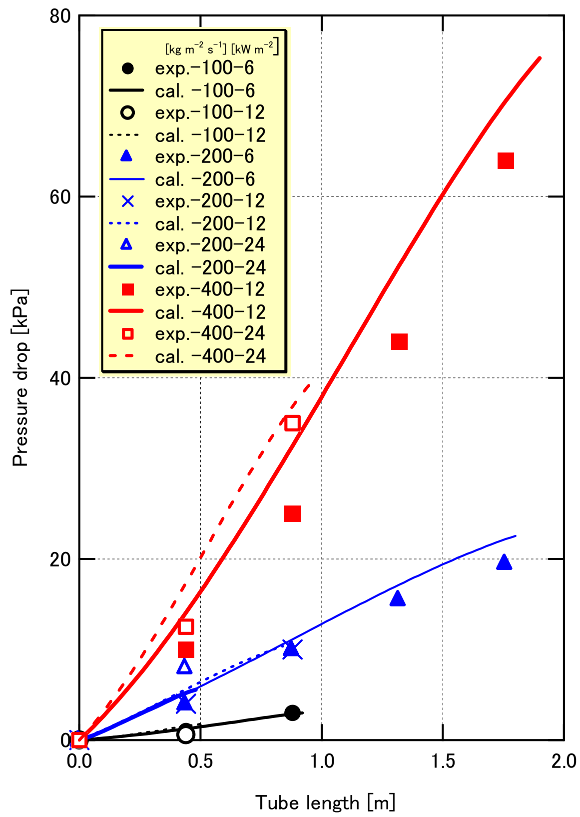


Figure 8. Comparison of pressure drops between measured values and calculated by Lockhart-Martinelli correlation.

Table 1 Specification of test tube and measurement intervals

Material of test tube	ID of tube(mm)	OD of tube(mm)	Length of tube(mm)	T.C.	OD of T.C.(mm)	Interval of T.C.(mm)	Interval of pressure(mm)
Stainless steel (SUS304)	2	3	1760	T-type	0.1	106	450

Table 2 Experimental conditions

Refrigerant	Inlet temperature [°C]	Quality [1]	Heat flux [kW m^{-2}]	Mass flux [kg m^{-2} s^{-1}]
HFO-1234yf	15	0.1–1.0	6–24	100–400

Table 3 Saturation properties of HFO-1234yf and R-134a at temperature of 288.15 K

	ρ_l [kg m^{-3}]	ρ_g [kg m^{-3}]	i_{lg} [kJ kg^{-1}]	λ_l [W m^{-1} K^{-1}]	μ_l [μPa s]	μ_g [μPa s]	c_{Pl} [kJ kg^{-1} K^{-1}]	σ [N m^{-1}]
HFO-1234yf	1077.3	26.3	156.5	0.0724	177.7	11.80	1.337	0.0077
R-134a	1243.4	23.76	186.59	0.0854	220.66	11.29	1.3869	0.0094

Table 4 Correlations for flow boiling heat transfer

Saitoh et al. ;

$$h_{TP} = Fh_l + Sh_{pool}, F = 1 + (1/X)^{1.05} \Big/ \left(1 + We_g^{-0.4}\right), S = 1 \Big/ \left[1 + 0.4\left(Re_{TP} \times 10^{-4}\right)^{1.4}\right],\ Re_{TP} = Re_l F^{1.25}$$

$$\mathrm{Re}_l > 1000, h_l = 0.023\frac{\lambda_l}{D}\left(\frac{G(1-x)D}{\mu_l}\right)^{0.8}\left(\frac{c_{pl}\mu_l}{\lambda_l}\right)^{0.4}, \quad \mathrm{Re}_l < 1000, h_l = \frac{4.36\lambda_l}{D}$$

$$h_{pool} = 207\frac{\lambda_l}{d_b}\left(\frac{qd_b}{\lambda_l T_l}\right)^{0.745}\left(\frac{\rho_g}{\rho_l}\right)^{0.581}\mathrm{Pr}_l^{0.533},\ d_b = 0.51\left[\frac{2\sigma}{g(\rho_l - \rho_g)}\right]^{0.5}$$

Yoshida;

$$h_{TP} = Fh_l + Sh_{pool}, F = 1 + 2(1/X)^{0.88}, S = 1 \Big/ \left\{1 + 0.9\left[\left(\mathrm{Re}_{tp} \times 10^{-4}\right)^{0.5} \Big/ \left(\left(Bo \times 10^4\right) \times X^{0.5}\right)\right]\right\},\ Bo = q \big/ Gh_{lv}$$

$$\mathrm{Re}_l > 1000, h_l = 0.023\frac{\lambda_l}{D}\left(\frac{G(1-x)D}{\mu_l}\right)^{0.8}\left(\frac{c_{pl}\mu_l}{\lambda_l}\right)^{0.4}, \quad \mathrm{Re}_l < 1000, h_l = \frac{4.36\lambda_l}{D}$$

$$h_{pool} = 207\frac{\lambda_l}{d_b}\left(\frac{qd_b}{\lambda_l T_l}\right)^{0.745}\left(\frac{\rho_g}{\rho_l}\right)^{0.581}\mathrm{Pr}_l^{0.533},\ d_b = 0.51\left[\frac{2\sigma}{g(\rho_l - \rho_g)}\right]^{0.5}$$

Kandlikar ;

$$h_{TP} = h_l\left[C_1 Co^{C_2}\left(25Fr_{lo}\right)^{C_5} + C_3 Bo^{C_4} F_K\right],\ Co = \left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}$$

for $Co < 0.65: C_1 = 1.136, C_2 = -0.9, C_3 = 667.2, C_4 = 0.7, C_5 = 0.3$

$Co > 0.65: C_1 = 0.6683, C_2 = -0.2, C_3 = 1058, C_4 = 0.7, C_5 = 0.3$

$C_5 = 0$ for vertical tubes, and for horizontal with $Fr_l > 0.04$

Gungor and Winterton ;

$$h_{\mathrm{TP}} = Eh_{\mathrm{l}},\ E = 1 + 3000Bo^{0.86} + 1.12\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_{\mathrm{l}}}{\rho_{\mathrm{g}}}\right)^{0.41}$$

If the tube is horizontal and the Froude number Fr_{l} is less than 0.05 then E should be multiplied by the factor

$$E_2 = Fr_{\mathrm{l}}^{(0.1-2Fr_{\mathrm{l}})}$$

Table 5 Mean deviation and accuracy of three correlations for flow boiling heat transfer coefficient

Correlation	Mean deviation (%)	Accuracy defined as fraction of data within ±20% error(%)
Saitoh et al.	9.2	92.8
Yoshida	20.4	51.9
Kandlikar	18.7	51.9
Gungor−Winterton	15.5	66.9

$Mean\ \ deviation = \frac{1}{n}\sum_{1}^{n}\frac{\left|h_{\mathrm{exp}} - h_{\mathrm{cal}}\right|}{h_{\mathrm{exp}}}\times 100\%$, the number of data is 181.