Flow boiling heat transfer of HFO1234yf and R32 refrigerant mixtures in a smooth horizontal tube: Part I. Experimental investigation

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Abstract. HFO1234yf has been proposed for mobile air-conditioners due to its low global warming potential (GWP) and performance comparable to that of R134a. However, its performance is inferior to that of R410A. This makes it difficult to be applied to residential air-conditioners. In order to apply the low-GWP refrigerant to residential air-conditioners, refrigerant mixtures of HFO1234yf and R32 are proposed, and their flow boiling heat transfer performances were investigated at two mass fractions (80/20 and 50/50 by mass%) in a smooth horizontal tube with an inner diameter of 2 mm. The experiments were conducted under heat fluxes ranging from 6 to 24 kW/m^2 and mass fluxes ranging from 100 to 400 kg/m^2 s at the evaporation temperature of 15 °C. The measured heat transfer coefficients were compared with those of pure HFO1234yf and R32. The results showed that the heat transfer coefficients of the mixture with an R32 mass fraction of 20% were 10–30% less than those of pure HFO1234yf for various mass and heat fluxes. When the mass fraction of R32 increased to 50%, the heat transfer coefficients of the mixture were 10–20% greater than those of pure HFO1234yf under conditions of large mass and heat fluxes. Moreover, the heat transfer coefficients of the mixtures were about 20–50% less than that of pure R32. The performances of the mixtures were examined at different boiling numbers. For refrigerant mixture HFO1234yf and R32 (80/20 by mass%), the nucleate boiling heat transfer was noticeably suppressed at low vapor quality for small boiling numbers, whereas the forced convective heat transfer was significantly suppressed at high vapor quality for large boiling numbers. This indicates that the heat transfer is greatly influenced by the mass diffusion resistance and temperature glide of the mixture.

Keywords: HFO1234yf; R32; Refrigerant mixture; Flow boiling; Heat transfer coefficient; Low-GWP refrigerant

Nomenclature

Bo	boiling number
C	factor in Lockhart-Martinelli correlation
c_{pl}	specific heat at constant pressure in liquid phase, J/kg K
D	diameter of tube, m
f	friction factor
G	mass flux, kg/m^2 s
h	heat transfer coefficient, kW/ m^2 K
h_{fg}	latent heat, kJ/kg
I	current, A
l	length of test section, m
p	pressure, Pa
q	heat flux, W/m^2
Q	heat per volume, W/m^3
Re	Reynold number
T_{wall}	inside-wall temperature, K
T_{sat}	saturation temperature, K
V	voltage, v
x	vapor quality
$\tilde{x}$	mole fraction

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X	Lockhart-Martinelli parameter
z	coordinate along tube direction, m

Greek symbols

ρ	density, kg/m^3
σ	surface tension, N/m
ϕ	two-phase flow multiplier
λ	conductivity, W/mK
δh	uncertainty of heat transfer coefficient, kW/m^{2}K
δT	uncertainty of temperature, K
δq	uncertainty of heat flux, kW/m^2

Subscripts

v	gas-phase, vapor-phase
L, l	liquid-phase
tp	two-phase
exp	experimental
cal	calculated
m, mix	mixture
i,in	inner
o,out	outer

1   Introduction

The issuance of the mobile air-conditioner (MAC) directive of the Europe Union (EU), which bans refrigerants with a global warming potential (GWP) above 150 from use in new type mobile air-conditioners in the EU market from 2011, has triggered the research and development of new refrigerants for mobile air-conditioners. HFO1234yf, a newly developed refrigerant with a GWP as low as 4, is a proposed drop-in solution for mobile air-conditioners replacing R134a. At the same time, increasing concerns about environmental protection have led to the reconsideration of the refrigerant in other applications. With regard to stationary air-conditioners, HFCs with high GWPs or even HCFCs are used, so a transition from these refrigerants to low-GWP replacements is a crucial issue globally. However, at present, there is no ideal candidate that can meet the requirements of both system performance and environmental performance.

The thermophysical properties of HFO1234yf are similar to those of R134a [1]. Experimental measurements [2] have shown that the flow boiling heat transfer coefficient of HFO1234yf is almost the same as that of R134a. However, because HFO1234yf has smaller latent heat than R410A, the coefficient of performance (COP) would decrease if HFO1234yf were to be substituted directly into a stationary heat pump system. One approach to maintaining a high system COP is using refrigerant mixtures of HFO1234yf and R32. However, due to the relatively high GWP of R32, the trade-off between the performance of the system and the GWP of the refrigerant mixture must be considered. Therefore, detailed information about the thermodynamic and heat transfer characteristics of the refrigerant mixture is required, with the mixing ratio of R32 as the parameter.

Several drop-in experiments on system performance have been conducted using either pure HFO1234yf or HFO1234yf+R32 mixtures. Fujitaka *et al.* [3] compared the system performances of pure HFO1234yf and HFO1234yf+R32 mixtures to that of R410A in a room air-conditioner. The system performance of HFO1234yf was significantly lower than that of R410A. However, the system performance of the HFO1234yf+R32 mixture improved as the R32 concentration was increased. The COP of HFO1234yf+R32 (50/50 by mass %) under cooling or heating conditions, respectively, was 95% or 94% of that of R410A. Hara *et al.* [4] tested the performances of two different models of a room air-conditioner with HFO1234yf as the refrigerant. The improved model had a capacity and COP similar to those of R410A systems. Okazaki *et al.* [5] tested HFO1234yf and HFO1234yf+R32 in modified room air-conditioners. The modified unit with HFO1234yf reached 95% of the annual performance factor (APF) of R410A. When the mass fraction of R32 was 60%, the APF of the mixture was 93.3% of that of R410A. In addition, thermodynamic properties of HFO1234yf+R32 mixtures, including vapor-liquid equilibrium and dynamic viscosity, were experimentally measured at various R32 mass fractions by Arakawa *et al.* [6]. A temperature glide of

7.7 °C was found at saturation temperature of 15 °C for the mixture with a 22% mass fraction of R32. Van Wijk et al. [7] investigated the heat transfer characteristics of refrigerant mixtures and explained the effect of different boiling points on the heat transfer performance. An inhomogeneity in the fluid consistency leads to mass diffusion resistance and a temperature glide that influences the heat transfer coefficient of the mixture. Scriven [8] presented the ratio between the superheat in a binary mixture and the liquid superheat in an ideal binary mixture without mass diffusion. Specifically, the influence of mass transfer on the growth of a bubble in a binary mixture was investigated on the basis of the balances of mass, momentum, and energy. Aprea et al. [9], Greco and Vanoli [10,11], Shin et al. [12], Ross et al. [13], and Sami et al. [14] studied the flow boiling heat transfer of refrigerant mixtures. Jung et al. [15,16], Zhang et al. [17], and Bennet and Chen [18] studied the flow boiling heat transfer characteristics of different zeotropic mixtures and proposed their prediction correlations. Cheng and Mewes [19] addressed the two-phase flow and flow boiling of mixtures in small and miniature channels. However, no experimental study on the heat transfer characteristics of HFO1234yf+R32 mixtures is available in the literature.

In this study, variations in the heat transfer coefficients of HFO1234yf+R32 mixtures (80/20 and 50/50 by mass%) were investigated experimentally. The heat transfer characteristics of refrigerant mixtures were compared with those of pure HFO1234yf and R32 under the same conditions of mass flux and heat flux. The effects of the mass diffusion resistance on the heat transfer coefficient are discussed.

2.   Properties of Mixtures

The properties of the refrigerants at the saturation temperature of 15 °C are summarized in Table 1. In the case of a mixture, the saturation temperature is the bulk temperature at a vapor quality of 0.5. Considering that the F-gas regulation sets a GWP limit of 150, refrigerant mixtures of HFO1234yf with R32 at mass fractions of 20% and 50% were tested, corresponding to GWPs of 138 and 340, respectively. As can be seen in Table 1, the latent heat and conductivity of the mixtures are elevated in comparison with those of pure HFO1234yf, and this can improve the heat transfer of the fluids. The thermophysical properties of mixtures were acquired using a Peng-Robinson type state equation [20] and those of pure refrigerants were calculated using REFPROP [21]. The surface tension of the binary mixture was computed from the surface tensions of each component as follows [22]:

$$\sigma_m = \tilde{x}_1 \sigma_1 + \tilde{x}_2 \sigma_2 \tag{1}$$

where $\tilde{x}_i$ is the mole fraction of the ith component.

3   Experimental Apparatus and Procedure

3.1 Test facility

The experimental apparatus used in the heat transfer performance measurements and flow pattern observations of the refrigerant mixtures is shown in Fig. 1. The test loop contained a Coriolis-type flow meter, a condenser, a flow control valve, a test tube, sight glasses, and a gear pump. The test section was a smooth horizontal stainless steel tube with an inner diameter (ID) of 2 mm. The length of the test section tube for the mixture (80/20 and 50/50 by mass %) was from 0.7 m to 2.3 m depending on the mass fluxes and heat fluxes.

The flow rate and inlet pressure of the refrigerant were controlled by adjusting the rotational speed of the magnetic gear pump and the opening of the flow control valve. The mass flow rate was measured using a Coriolis-type flow meter with an accuracy of ±0.2% reading value. The inlet vapor quality of the test section was maintained at about 0.2 by adjusting the amount of heat supplied to a preheater located upstream of the test section. The input electrical power was measured using a voltmeter and an ammeter. The energy required for evaporation was supplied by directly heating the test tube using direct current. The measurement uncertainty of both the voltage and the current were 0.02% reading value.

The outer wall temperature of the test tube was measured at the side of the tube along the axis by using T-type thermocouples with 0.1-mm OD, and the inner wall temperature of the tube was calculated from the outer wall temperatures by using Fourier's law. An 8-μm-thick Teflon sheet was inserted between each thermocouple and the test tube to eliminate the influence of electric current on

the thermocouples. All the thermocouples were calibrated using a high-precision platinum resistance thermometer sensor with an accuracy of ±0.03 K. The accuracy of the calibrated thermocouples was ±0.1 K within the range of 0～50°C. Pressures were recorded by a pressure sensor with an accuracy of ±0.01% full scale (10 MPa). To reduce the heat loss to the environment, the pipe of the test section was covered by the insulation layer. The surface temperatures of the insulation layer were measured by thermocouples. The entire test tube was placed within an air channel whose temperature was matched to the outer surface temperature of the test tube. The escaping heat from the surface of the test tube is less than 1% of the total heat.

The purity of the refrigerant was 99.7% for HFO1234yf and 99.9% for R32. HFO1234yf and R32 were mixed at the desired mass fraction in advance and charged into the system as liquid. By using the Peng-Robinson type equation of state proposed in our laboratory [20], the concentration of the mixture in the test tube was calculated on the basis of the fluid parameters measured at the inlet and outlet of the preheater. In order to confirm the concentration of the mixture, gas chromatography was used to test the superheated vapor component at the outlet of the test section. The maximum deviation of the two values was less than ±2%, and the concentrations calculated from the measured parameters were adopted to derive the final results.

3.2 Heat transfer data reduction

Table 2 summarizes the experimental conditions. The saturation temperature of the mixture was set to 15°C at a vapor quality of 0.5 during the experiments. The local heat transfer coefficient of the test tube was defined as

$$h_{exp} = \frac{q_{in}}{T_{wall,in} - T_{sat}} \tag{2}$$

where q_{in} is the heat flux of the inner surface, $T_{wall,in}$ is the temperature of the inner wall which can be calculated from Equations (3) and (4). T_{sat} is the saturation temperature of the mixture and pure refrigerants as deduced from the local refrigerant pressure interpolated between the inlet and outlet pressures and the vapor quality is required for the calculation of the saturation temperature of the mixture.

$$T_{wall,in} = T_{wall,out} + q_{in}\frac{D_i}{2\lambda}\left(\log\frac{D_o}{D_i}\right) - \frac{Q}{8\lambda}\left(D_i^2\log\frac{D_o}{D_i} - \frac{1}{2}(D_o^2 - D_i^2)\right) \tag{3}$$

$$Q = \frac{I \cdot V}{\frac{\pi}{4}(D_o^2 - D_i^2)l} \tag{4}$$

All experimental data were collected after a steady state was achieved. The data for pure HFO1234yf are from the measurements of Satoh *et al.* [2].

3.3 Experimental uncertainty

The uncertainty of the experimental results was calculated by means of the following equation:

$$\delta R = \left[\sum_{n=1}^{k}\left(\frac{\partial R}{\partial y_n}\delta y_n\right)^2\right]^{1/2} \tag{5}$$

where δR is the total uncertainty associated with the dependent variable R, y is the independent variable which affects the dependent variable R, δy is the uncertainty of y. The relative uncertainty of h_{exp} can be calculated as follows.

$$\frac{\delta h_{exp}}{h_{exp}} = \sqrt{\left(\frac{\delta q}{q}\right)^2 + \left(\frac{\delta T_{wall,i}}{T_{wall,i} - T_{m,sat}}\right)^2 + \left(\frac{\delta T_{m,sat}}{T_{wall,i} - T_{m,sat}}\right)^2} \tag{6}$$

The calculation of uncertainty for q_{in} and $T_{wall,in}$ follows the same method by using the equation (5). The total measurement uncertainty varies with the operating conditions but mainly depends on the accuracy of the wall superheat, inner wall temperature and the saturation temperature. The uncertainty of inner wall temperature is determined from the outer wall temperature $T_{wall,out}$, and is

approximately equal to ±0.1°C in spite of effects of q_{in} and Q. The influence of the uncertainties of q_{in} and Q on $\delta T_{wall,in}$ is very small in comparison with that of the outer wall temperature $T_{wall,out}$. The uncertainty in the saturation temperature is related to the uncertainty in the equation of state as well as the accuracy of the local pressure (1 kPa corresponds to a maximum deviation of ±0.027 °C for R32, ±0.065 °C for HFO1234yf, ±0.044 °C for the HFO1234yf+R32 mixture at 20% mass fraction of R32, and ±0.03 °C for HFO1234yf+R32 at 50% mass fraction). Uncertainties of main measurement devices are shown in Table 3.

Under the actual operating conditions, the uncertainty in the heat transfer coefficients of R32 is ranged between 5.6% and 15.6%, that of HFO1234yf +R32 at 80% of R32 by mass% is from 5.3% to15.2% and that of HFO1234yf +R32 at 50% mass fraction is from 5.2% to 17.2%.

4 Results and Discussions

4.1 Comparison of heat transfer coefficients for mixtures and pure HFO1234yf

Normally two main heat transfer mechanisms are considered for the flow boiling heat transfer inside a tube; i.e., nucleate boiling and forced convection. The significance of the interaction of the two mechanisms is usually evaluated by means of the boiling number [22], which is defined as

$$Bo = q/(h_{fg}G) \tag{7}$$

where h_{fg} is the latent heat. The study of Dang *et al*. [23] discusses the influence of forced convective boiling of carbon dioxide on the nucleated boiling in a grooved tube. To explain the suppression of bubble nucleation and bubble growth at small heat flux or large convective heat transfer coefficient (small Bo), a formula, q_w/h_{tp}, described by Collier and Thome [24] was cited, where h_{tp} is the two-phase heat transfer coefficient in the absence of bubble nucleation. Based on summarizing a large amount of experimental data, Yoshida *et al*. [25] and Kandlikar [26] considered *Bo* a very meaningful factor to characterize nucleate boiling or forced convection and thus introduced *Bo* into their correlations. Due to the effect of mass diffusion in the evaporative heat transfer of a mixture, the flow boiling of mixtures for variable *Bo* follows the similar rule as that of pure refrigerants yet presents a unique appearance. Therefore, the measured results are regulated according to the *Bo* of the mixture. Table 4 shows the *Bo* values for HFO1234yf, R32, and the refrigerant mixtures of HFO1234yf and R32 (80/20 and 50/50 by mass %). The heat transfer coefficients of HFO1234yf and the HFO1234yf+R32 mixtures are compared on the basis of *Bo* shown in Table 4.

Figure 2(a) shows the heat transfer coefficients of the HFO1234yf+R32 mixture (80/20 by mass%) and pure HFO1234yf at a small *Bo*. Similar tendencies in the heat transfer coefficients of the pure HFO1234yf and the refrigerant mixture were found with the vapor quality under the same *Bo* condition. As shown in Fig. 2(a), for the pure HFO1234yf the heat transfer coefficient increases almost lineally with the vapor quality until the dryout. The forced convective heat transfer is prominent in this evaporative process. At low vapor quality, the heat transfer coefficient of the mixture is nearly the same as that of pure HFO1234yf. As the vapor quality increases, the difference between the heat transfer coefficient of the mixture and that of pure HFO1234yf gradually increases. It seems that the forced convective heat transfer of the mixture is suppressed at high vapor quality. Analogous results have been demonstrated by Shin et al. [12], who tested the flow boiling heat transfer coefficient of R290+R600a mixtures (25/75, 50/50, 75/25 by mass%). It might be explained that the influence of the heat flux on the heat transfer is weaker than that of the mass flux when *Bo* is small. The nucleate boiling is completely suppressed at high vapor quality and the forced convective heat transfer dominates the heat transfer process. The evaporative phase transition of the superheated liquid takes place mainly at the vapor-liquid interface, not at the wall surface. For pure refrigerants, no compositional imhomogeneity exists within the liquid layer. For a mixture, however, the more volatile component at the vapor-liquid interface, here R32, evaporates more easily than the less volatile component, so the concentration of the less volatile component at the vapor-liquid interface is higher than that in the bulk liquid. Consequently, the more volatile component in the bulk liquid must first move towards the interface and then evaporate. Therefore, the evaporation of a mixture is related with the ability of the more volatile component to reach the vapor-liquid interface. For this reason, the forced convective heat transfer is suppressed at high vapor quality and the heat transfer coefficient of the mixture is depressed significantly.

In Fig. 2(b), *Bo* is large under all conditions and the heat transfer coefficient of pure HFO1234yf does not tend to increase as much with the vapor quality. This means that the nucleated boiling almost dominates the evaporative heat transfer. For the HFO1234yf+R32 mixture (80/20 by mass%), a tendency similar to that of pure HFO1234yf is observed. At a mass flux of 200 kg/m^2 s, the heat transfer coefficient of the mixture rises at high vapor quality to approach that of pure HFO1234yf. Conversely, at large *Bo* the heat transfer coefficient is much lower than that of pure HFO1234yf, which implies that the nucleate boiling is significantly suppressed at low vapor quality. The reason is that a large heat flux generates vigorous bubbles, so the inhomogeneity of the concentration distribution in the liquid film becomes serious. With the increase in vapor quality, the velocity of the flow increases and the disturbances of the fluid become great enough to decrease the compositional inhomogeneity of the bulk fluid. The effect of mass diffusion on the nucleate boiling is thus moderated. This is why at a mass flux of 200 kg/m^2 s the heat transfer coefficient of the mixture rises at high vapor quality. At a lower mass flux of 100 kg/m^2 s the flow is laminar, so the small disturbances caused by the flow cannot help to decrease the compositional variation within the liquid film. Therefore, the increment in the heat transfer coefficient of the mixture does not occur at a mass flux of 100 kg/m^2 s.

In Fig. 2(c), *Bo* is moderate under all conditions. The data marked with asterisks were calculated from the correlations proposed by Satoh *et al.* [2], which produced a good fit to their experimental data within ±20% deviation. The data for pure HFO1234yf also increase with the generation of vapor, although the incremental tendency is much gentler than that in Fig. 2(a). Under such *Bo* conditions, the influence of both nucleate boiling and forced convective boiling on heat transfer are observed. The heat transfer coefficient does not increase noticeably until the vapor quality reaches 0.5. After that, the heat transfer coefficient increases linearly with vapor quality, showing the effect of convection. The trend in the heat transfer coefficient of the mixture almost parallels that of the pure HFO1234yf, while the absolute value of the heat transfer coefficient of the mixture is about 10–20% less than that of pure HFO1234yf. The suppression of the nucleate boiling at low vapor quality and that of the forced convective boiling at high vapor quality together moderately influence the heat transfer coefficient of the mixture.

Through the comparisons at different mass fluxes, heat fluxes, and *Bo* values, it is found that the heat transfer coefficient of the mixture with an R32 mass fraction of 20% is less than that of pure HFO1234yf by 10–30%. The main reason is that mass diffusion resistance existing in the heat transfer process influences the heat transfer coefficients significantly. Compared with the results for pure HFO1234yf, the heat transfer coefficient of the mixture obeys a different regulation as *Bo* varies. When *Bo* is small, the forced convective heat transfer of the mixture is noticeably suppressed at high vapor quality. When *Bo* becomes large, the nucleate boiling is drastically suppressed at low vapor quality.

Figure 3 shows results comparing the heat transfer coefficient of the HFO1234yf+R32 mixture at 50% mass fraction and that of pure HFO1234yf at two boiling numbers listed in Table 4. In Fig. 3(a), *Bo* is small under all conditions. The heat transfer coefficient of the HFO1234yf+R32 mixture (50/50 by mass%) increases with vapor quality the same as that of pure HFO1234yf. The forced convective heat transfer dominates the heat transfer process as for pure HFO1234yf. The heat transfer coefficient of the mixture is slightly greater than that of pure HFO1234yf at low vapor quality and is almost same as that of pure HFO1234yf at high vapor quality. It seems that the forced convective heat transfer is slightly suppressed at small *Bo*, obeying the same regulation as the HFO1234yf+R32 mixture with an R32 mass fraction of 20%. As shown in Fig. 3(b), at large *Bo* the heat transfer coefficient of the HFO1234yf+R32 mixture at 50% mass fraction is greater than that of pure HFO1234yf. Because of the increased mass ratio of R32, the thermodynamic properties of R32 make a positive contribution to the heat transfer coefficient of the mixture. Due to a moderate temperature glide of 4.6°C at the R32 mass fraction of 50%, the difference in composition between the bubbly interface and the bulk liquid is smaller than at the R32 mass fraction of 20%. The mass diffusion can be effectively mitigated by the perturbation due to a large mass or heat flux. At a lower mass flux of 100 kg/m^2 s, the effects of mass diffusion on heat transfer obviously appear due to laminar flow.

According to the comparisons between the heat transfer coefficients of the HFO1234yf+R32 mixtures (80/20 and 50/50 by mass %) and that of pure HFO1234yf, it can be concluded that the heat transfer of the mixture is significantly influenced by the mass diffusion.

4.2 Comparison of heat transfer coefficients for mixtures and pure R32

Figure 4 shows the comparison between the measured heat transfer coefficient of R32 and those of the HFO1234yf+R32 mixtures (80/20 and 50/50 by mass%) at a mass flux of 200 kg/m^2 s and heat fluxes from 6 to 24 kW/m^2. As shown in Fig. 4, at low vapor quality the heat transfer coefficient of R32 increases with heat flux, showing that the nucleate boiling is dominant in the low vapor quality region. At a low heat flux of 6 kW/m^2, the heat transfer coefficient increases almost linearly with vapor quality until the dryout. However, at a greater heat flux of 12 or 24 kW/m^2, the heat transfer coefficient does not begin to increase until the vapor quality reaches 0.5, but after that the heat transfer coefficient increases linearly with vapor quality, showing the effect of convection. It is also noticed that the measure heat transfer coefficient at dryout for the heat flux of 6 kW/m^2 is higher than that of 12 kW/m^2 and 24 kW/m^2 in Fig. 4, which is due to the instability of liquid film at dryout. In addition, it can be seen that the heat transfer coefficient of R32 is much higher than that of pure HFO1234yf in Fig,2. The main reason is that the thermophysical properties of R32, such as conductivity, latent heat, are much higher than that of HFO1234yf and the mixtures, which contribute a higher heat transfer coefficient.

Similar variations of the heat transfer coefficient with the vapor quality and the heat flux can be seen in Fig. 4 for the HFO1234yf+R32 mixtures. Generally speaking, the heat transfer coefficient of the mixture with a 20% mass fraction of R32 is less than that of pure R32 by 20–50% for various mass and heat fluxes. Despite the increment in the heat transfer coefficient of the HFO1234yf+R32 mixture with the increase in the mass fraction of R32, the heat transfer coefficient of the HFO1234yf+R32 mixture at 50% mass fraction is about 20–40% less than that of pure R32.The above results indicate that R32 has very good heat transfer characteristics. However, as shown in Fig. 4, the effect of beneficial contributions by R32 on the heat transfer coefficients of the mixtures is not obvious, due to the small concentrations of R32 and the influence of mass diffusion resistance.

4.3 Effects of mass flux and heat flux on heat transfer coefficients

The effects of mass flux and heat flux on the heat transfer coefficients of HFO1234yf+R32 mixtures (80/20 and 50/50 by mass%) are shown in Figs. 5 and 6. Figure 5 shows the variations in the heat transfer coefficients with the vapor quality at three heat fluxes when the mass flux of the refrigerant mixture is 200 kg/m^2. For pure HFO1234yf, the heat transfer coefficient approaches the same value at high vapor quality for different heat fluxes when the mass flux is the same. This implies that the forced convective heat transfer dominates the heat transfer process when the vapor quality is high. In the case of zeotropic HFO1234yf+R32 mixtures, a similar variation can be observed when the heat flux is large, as shown in Fig. 5. It also can be seen that the heat transfer coefficients of pure HFO1234yf and the mixtures all increase with the heat flux. A large heat flux can help to improve the heat transfer of the mixtures as well as that of the pure refrigerants. However, for the HFO1234yf+R32 mixture with a 20% mass fraction of R32, when the heat flux is large the difference between the heat transfer coefficient of pure HFO1234yf and that of the mixture is large at low vapor quality, as shown in Fig. 5(a). The reason, as explained in Subsection 4.1, is the suppression of nucleate boiling as the large heat flux causes an inhomogeneity of the concentration in the liquid film. In addition, the suppression of the forced convection at high vapor quality is far more obvious at small heat flux than at large heat flux. The reason is that nucleate boiling at large heat flux and high vapor quality is not completely suppressed and some bubble generation on the wall surface can agitate the vapor-liquid interface and thus moderate the inhomogeneity of the concentration in the interfacial layer. For the HFO1234yf+R32 mixture with a 50% mass fraction, the effect of suppression on the heat transfer coefficient is obvious at small heat flux, as shown in Fig. 5(b). At large heat flux, a variation similar to that of the HFO1234yf+R32 mixture with a 20% mass fraction of R32 is not observed. This is because the glide temperature of the mixture with an R32 concentration of 50% is only 4.6 °C, whereas that of the mixture with an R32 concentration of 20% is 7.7 °C.

Figure 6 shows the heat transfer characteristics of the HFO1234yf+R32 mixtures (80/20 and 50/50 by mass%) at three mass fluxes when the heat flux is 12 kW/m^2. If the heat flux is held constant while the mass flux is changed, the heat transfer coefficient of pure HFO1234yf is fairly unaffected at low vapor qualities, as shown in Fig. 6. This means that the nucleate boiling dominates the heat transfer at low vapor qualities. However, for the HFO1234yf+R32 mixtures, no similar phenomenon occurs.

The mass flux is larger and the heat transfer coefficients of the mixtures are greater at low vapor quality. The reason is that the nucleate boiling of the mixture at low vapor quality is suppressed due to the mass diffusion resistance. A large mass flux can generate a turbulent flow, which decreases the inhomogeneity of the consistency and increases the heat transfer coefficient of the mixture. As more and more vapor quality builds, forced convection heat transfer becomes the main contribution to the evaporative heat transfer. The greater the mass flux, the greater the heat transfer coefficients obtained. At a mass flux of 100 kg/m^2 s, the heat transfer of the mixture is significantly suppressed due to the weak mixing activity.

5 Pressure Drop of Mixtures

Two-phase friction pressure drops (Δp) were predicted using the Lockhart-Martinelli correlations [27] . The total predicted pressure drops were compared with the measured results. The correlations are defined as follows:

$$-\left(\frac{dp}{dz}\right)_{tp} = -\left(\frac{dp}{dz}\right)_l \phi_l^2 \tag{8}$$

$$\left(-\frac{dp}{dz}\right)_l = 2f_l \frac{G_l^2}{D_i \rho_l} \tag{9}$$

$$\left(-\frac{dp}{dz}\right)_v = 2f_v \frac{G_v^2}{D_i \rho_v} \tag{10}$$

$$\phi_l^2 = 1 + C/X + 1/X^2 \tag{11}$$

$$X^2 = \frac{(dp/dz)_l}{(dp/dz)_v} \tag{12}$$

$$\begin{cases} f_l = 0.079 \mathrm{Re}_l^{-0.25} &, \mathrm{Re}_l > 2000 \\ f_l = 16/\mathrm{Re}_l &, \mathrm{Re}_l < 2000 \end{cases} \tag{13}$$

$$\begin{cases} f_v = 0.079 \mathrm{Re}_v^{-0.25} &, \mathrm{Re}_v > 2000 \\ f_v = 16/\mathrm{Re}_v &, \mathrm{Re}_v < 2000 \end{cases} \tag{14}$$

where ϕ_l is two-phase multiplier in the liquid phase, while f_l, f_v are the friction factors of the liquid and the gas phase defined as Equations (13) and (14). Values of C [28] are listed in Table 5. Figure 7 shows the pressure drop gradient measured for the HFO1234yf+R32 mixtures (80/20 and 50/50 by mass%), pure R32 and pure HFO1234yf and those prediction data. 62%～65% of predicted pressure drop gradients are in good agreement with the measured values within a deviation limit of ±20% and this correlation can captures 72%～80% of measured values within a deviation limit of ±30% for both pure HFO1234yf and the mixtures at two concentrations In case of R32, 54% of data are predicted within ±20% and the prediction deviation of 61% of data are within ±30%. The results show that Lockhart-Martinelli correlation cannot predict the pressure drop of R32 good enough. Then, more correlations for prediction of pressure drop need to be investigated.

6 Conclusions

Flow boiling heat transfer of the refrigerant mixture HFO1234yf+R32 was experimentally investigated in this study. Local heat transfer coefficients and pressure drops were measured for the mixture at R32 mass fractions of 20% and 50%, with the mass flux ranging from 100 to 400 kg/m^2 s and heat flux ranging from 6 to 24 kW/m^2. Inner diameter of the test tube was 2 mm and the evaporation temperature was set to 15 °C at a vapor quality of 0.5. Main results are summarized as follows:
1) Heat transfer coefficient of the refrigerant mixture HFO1234yf+R32 at R32 mass fraction of 20% is less than that of pure HFO1234yf. The heat transfer coefficient of the mixture at 50% mass

fraction is greater than that of pure HFO1234yf at large heat and mass fluxes.

2) The heat transfer coefficients of the mixtures at two concentrations are 20–50% lower than that of pure R32. The effects of mass diffusion on the heat transfer of a mixture are significant.

3) For HFO1234yf+R32 (80/20 by mass%), the nucleate boiling heat transfer is noticeably suppressed at low vapor quality for small boiling numbers, whereas the forced convective heat transfer is significantly suppressed at high vapor quality for large boiling numbers.

4) Large mass and heat fluxes can enhance the heat transfer coefficient of the mixture.

5) The Lockhart-Martinelli correlation can be used to predict the pressure drop of the mixtures. The 72%～80% of predicted results were in good agreement with the measured pressure drops within the deviation of ±30% .However, the prediction results for R32 are not satisfactory.

Acknowledgements

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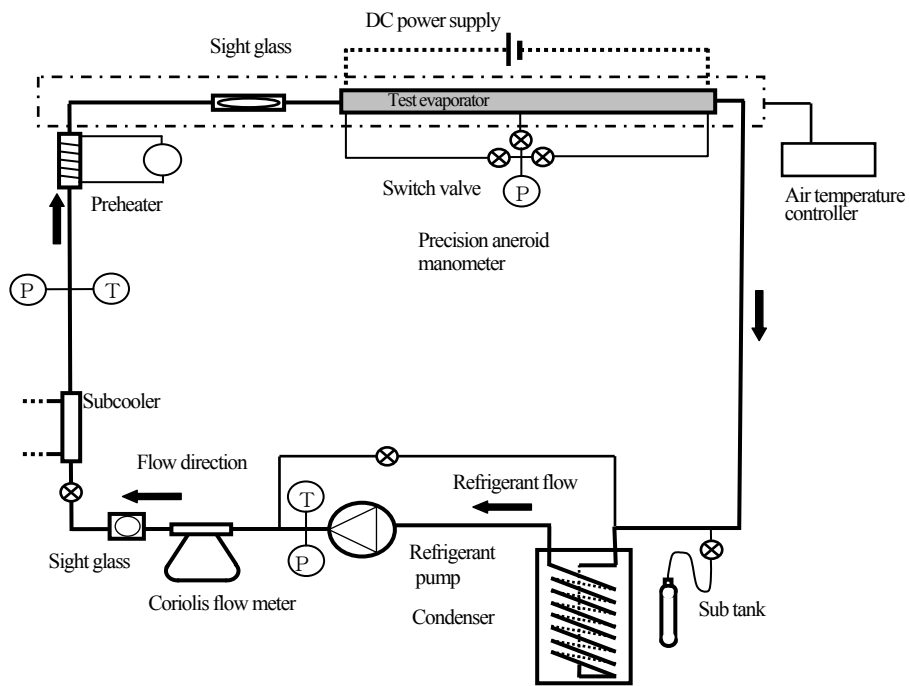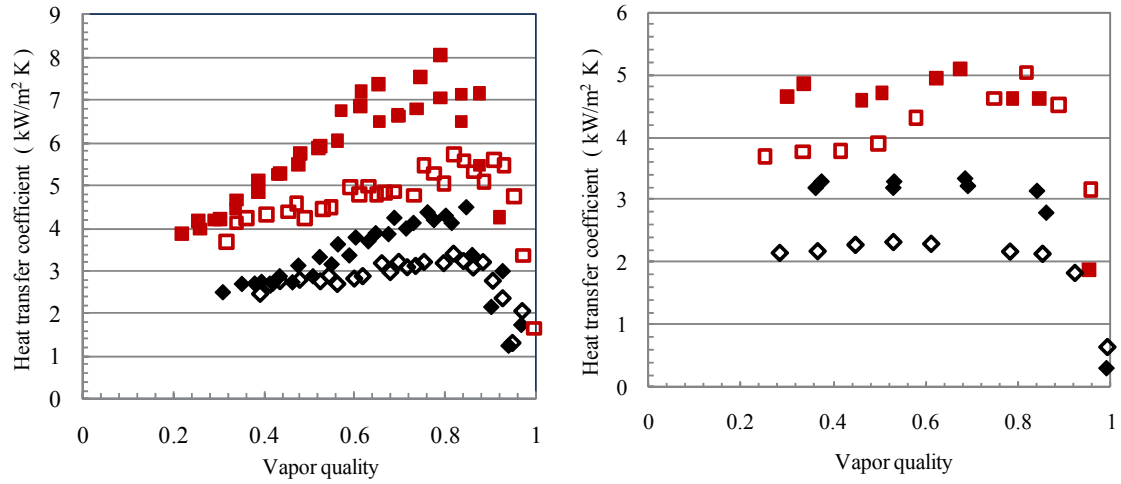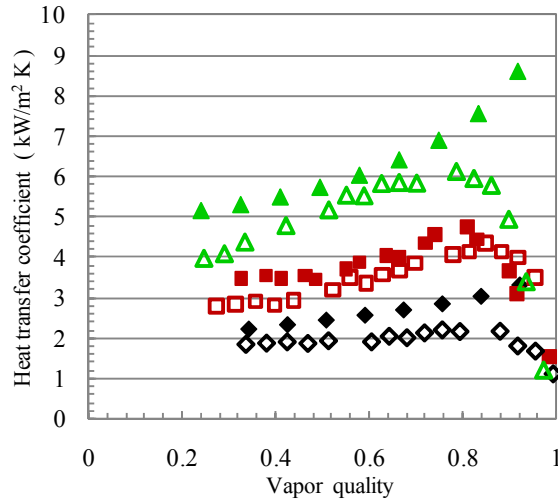
Figure 1 Schematic of experimental system used to measure flow boiling heat transfer.

(a)

(b)

Note: *The data marked with asterisks were calculated from the correlations proposed by Satoh *et al*. [2]

(c)

Figure 2. Comparison of heat transfer coefficient of the HFO1234yf+R32 mixture (80/20 by mass%) with that of pure HFO1234yf on basis of boiling number.

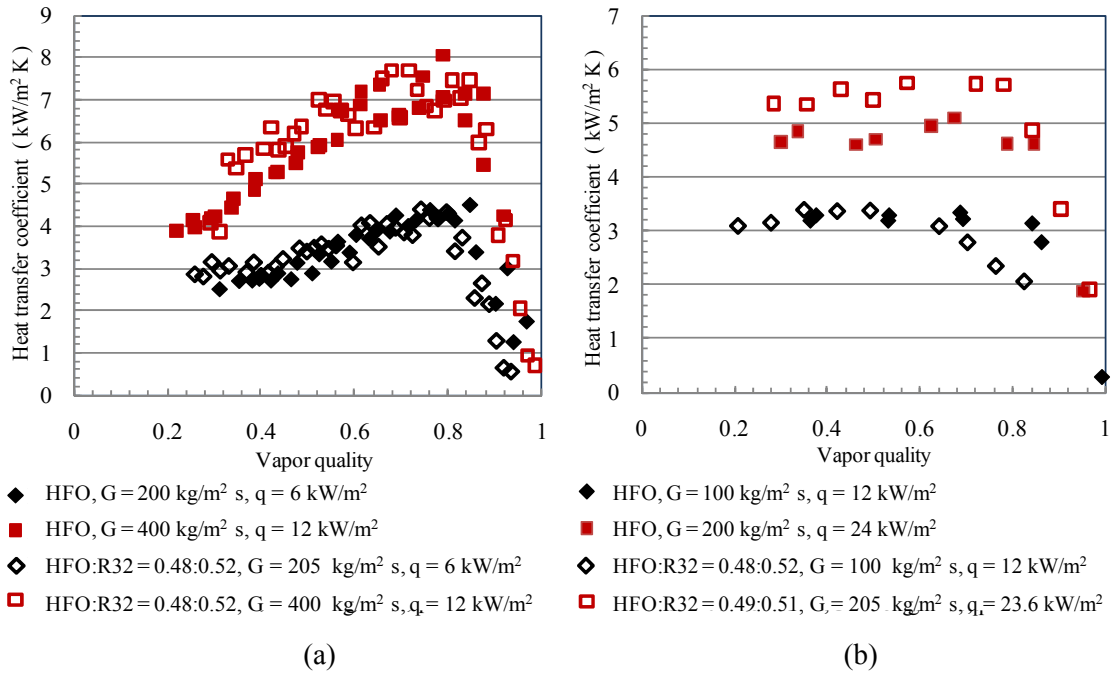|	
◆ HFO, G = 200 kg/m² s, q = 6 kW/m²	◆ HFO, G = 100 kg/m² s, q = 12 kW/m²
■ HFO, G = 400 kg/m² s, q = 12 kW/m²	■ HFO, G = 200 kg/m² s, q = 24 kW/m²
◇ HFO:R32 = 0.48:0.52, G = 205 kg/m² s, q = 6 kW/m²	◇ HFO:R32 = 0.48:0.52, G = 100 kg/m² s, q = 12 kW/m²
□ HFO:R32 = 0.48:0.52, G = 400 kg/m² s, q = 12 kW/m²	□ HFO:R32 = 0.49:0.51, G = 205 kg/m² s, q = 23.6 kW/m²

(a) (b)

Figure 3. Comparison of heat transfer coefficient of the HFO1234yf+R32 mixture (50/50 by mass%) with that of pure HFO1234yf on basis of boiling number.
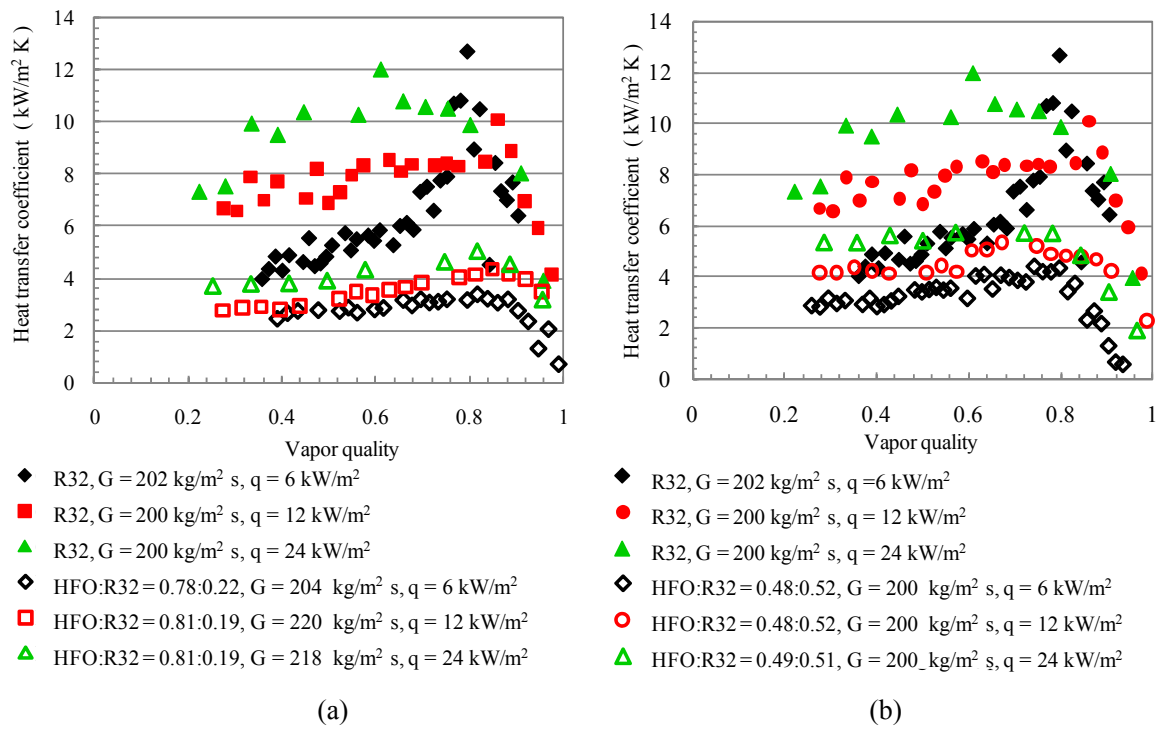
Figure 4. Measured heat transfer coefficients of pure R32 and the HFO1234yf+R32 mixtures.
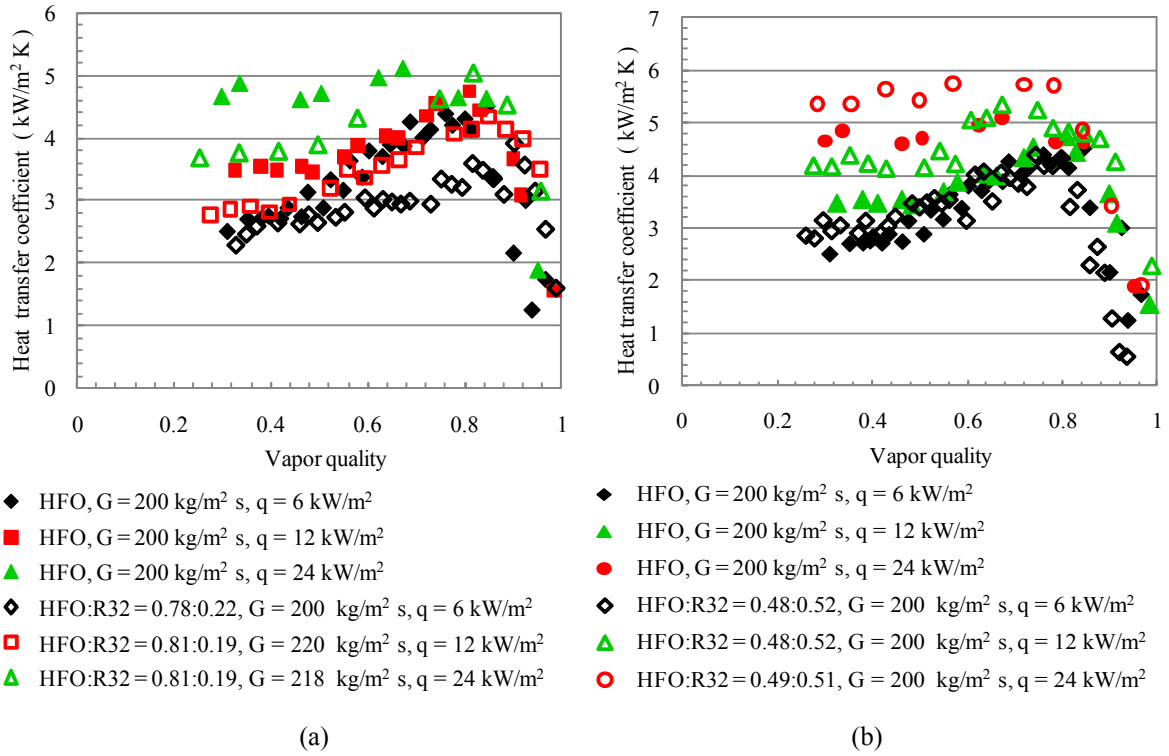
Figure 5. Measured heat transfer coefficients of pure HFO1234yf and the HFO1234yf+R32 mixtures at the same mass flux and various heat fluxes.

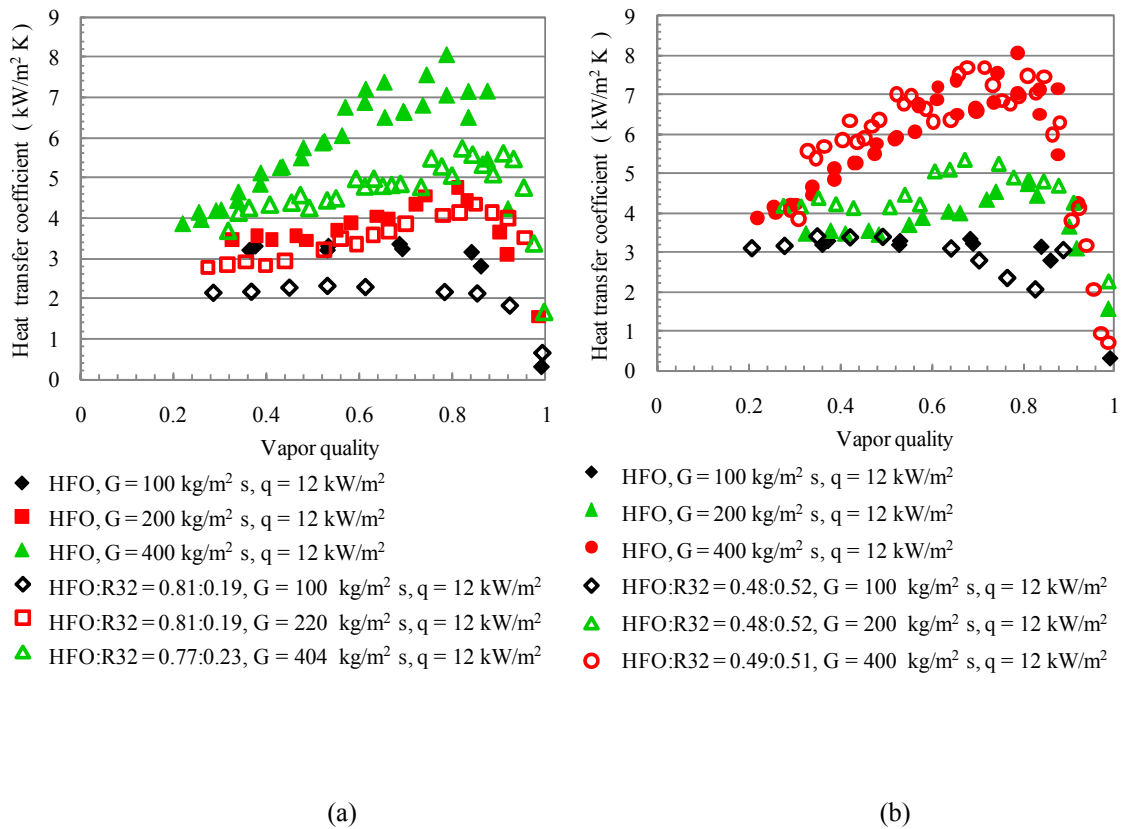|			
(a)		(b)	

Figure 6. Measured heat transfer coefficients of pure HFO1234yf and the HFO1234yf+R32 mixtures at the same heat flux and various mass fluxes.
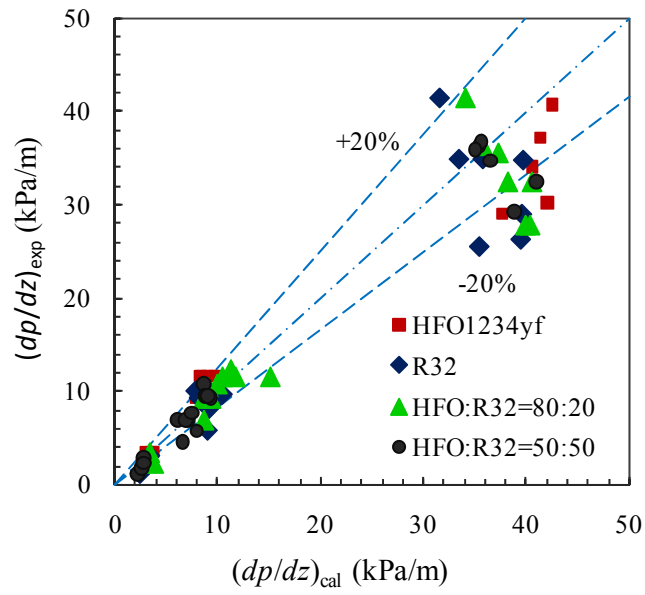
Figure 7. Comparison of pressure drops gradient measured and predicted for refrigerant mixtures and pure refrigerants.

Table 1　Properties of refrigerants

Parameters	Unit	R32 100%	HFO1234yf:R32 = 50%:50%	HFO1234yf:R32 = 80%:20%	HFO1234yf 100%
Molecular weight	kg/kmol	52	71	92	114
GWP (100 year)	—	675	340*	138*	4
Liquid density	kg/m^3	1000.9	1048	1063	1127
Vapor density	kg/m^3	35.19	35.89	44	28.25
Liquid thermal conductivity	mW/m K	133.54	82	74	71.13
Specific heat	kJ/kg K	1.843	1.296	1.259	1.366
Surface tension	mN/m	8.41	8.09	7.75	7.39
Latent heat	kJ/kg	290	223	175	153.05
Temperature glide	°C	----	4.55	7.65	----

Note: The saturation temperature is 15 °C.　　* Cited from Akira Fujitaka *et al.* [3]

Table 2　Experimental conditions

Refrigerant	HFO1234yf, R32, HFO1234yf + R32
HFO1234yf:R32 wt%	100:0; 0:100; 80:20; 50:50
Saturation temperature of the refrigerant	15°C
Quality	0.2–1.0
Heat flux (kW/m^2)	6, 12, 24
Mass flux (kg/m^2 s)	100, 200, 400

Table 3　Uncertainty of measurement device

Variable	Device	Accuracy	Range
Temperature of tube wall	Thermocouples	±0.1 K	0–50 °C
Temperature of refrigerant	Pt 100	±0.03 K	-50–200 °C
Mass flow rate	Coriolis type flow meter	±0.2% reading value	1.5–225 g/min
Pressure	Digiquartz pressure sensor	±0.01% full scale	1–10 MPa
Voltage	Voltmeter	±0.02% reading value	0–64 V
Length	Ruler	±0.001 m	0–2 m

Table 4　Boiling numbers under different experimental conditions

Heat flux (kW/m^2) / Mass flux (kg/m^2 s)	HFO1234yf	HFO1234yf +R32 (80/20 by mass%)	HFO1234yf +R32 (50/50 by mass%)	R32	Degree of comparison
6/200, 12/400	1.96×10^{-4}	1.98×10^{-4}	1.50×10^{-4}	1.03×10^{-4}	Small
12/200, 24/400	3.92×10^{-4}	4.0×10^{-4}	3.24×10^{-4}	2.07×10^{-4}	Moderate
12/100, 24/200	7.84×10^{-4}	7.45×10^{-4}	5.91×10^{-4}	4.14×10^{-4}	Large

Note: The saturation temperature is 15 °C

Table 5　Value of C in Equation (11)

Re$_l$	Re$_v$	C
> 2000	> 2000	20
> 2000	< 2000	10
< 2000	> 2000	12
< 2000	< 2000	5