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Evaporation heat transfer of R-32, R-134a, R-32/134a, and R-32/125/134a inside a horizontal smooth tube

T.Y. Choi, Y.J. Kim, M.S. Kim*, S.T. Ro

Department of Mechanical Engineering, Seoul National University, Seoul 151-742, South Korea

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Abstract

Evaporative heat transfer coefficients of R-32, R-134a, R-32/134a, and R-407C (R-32/125/134a: 23/25/52 wt%) have been measured in a horizontal smooth tube. Experiments were conducted for test section average temperatures of -12.0–17.0°C, with a mass flux of 240–1060 kg m^{-2} s^{-1}, and a heat flux of 4.1–28.6 kW m^{-2}. Evaporative heat transfer characteristics of R-407C as a possible alternative to R-22 have been compared with those of R-22. A new correlation based on superposition model for pure refrigerants and refrigerant mixtures was presented. Experimental results were compared with several correlations which predict evaporative heat transfer characteristics. Comparison with the experimental data in the open literature showed that our correlation gave a satisfactory result. © 2000 Elsevier Science Ltd. All rights reserved.

1. Introduction

Several new environmental problems have been encountered in recent decades. Among these, global warming and ozone depletion caused mainly by CFC and HCFC refrigerants are major issues. These effects were not considered when the refrigeration industry first adopted these refrigerants as working fluids in refrigerators and heat pumps. The release of these refrigerants to the atmosphere has caused the accumulation of these refrigerants in the stratosphere, which is known to deplete the ozone layer. The development of new alternative refrigerants and a prompt shift to new refrigerants are mostly required to protect the environment. The proper choice of a new refrigerant should be the first step, however, in this process several things have to be considered. Thermodynamic properties are the most important consideration, and additionally, heat transfer characteristics are very important in real application. Extensive studies on the alternative refrigerants are underway including the evaporation and condensation heat transfer as well as studies on system performance using alternative refrigerants [1–13].

When the evaporation heat transfer is considered, experimental studies measuring two-phase heat transfer coefficients and the prediction of heat transfer coefficients for new refrigerants are two major streams of the research. By collecting the measurement data, several correlations predicting evaporative heat transfer coefficients have been proposed [14–19]. Unfortunately, it is not so easy to find the measured data of evaporative heat transfer coefficients and the correlations to predict the heat transfer coefficient for hydrofluorocarbon (HFC) refrigerants and their mixtures, even though HFC refrigerants are considered as possible alternative refrigerants.

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* Corresponding author. Tel.: +82-2-880-8362; fax: +82-2-883-0179.

E-mail address: minskim@plaza.snu.ac.kr (M.S. Kim).

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Nomenclature

Bo	boiling number, $q''/h_{fg}G$
c_1–c_5	constants defined in Eqs. (6) and (7)
c_M	constant defined in Eq. (9)
Co	convection number, $\{(1\text{-}x)/x\}^{0.8}(\rho_l/\rho_v)^{0.5}$
C_P	specific heat (kJ kg^{-1} K^{-1})
D_i	inner diameter of the test tube (m)
E	enhancement factor defined in Eq. (6)
F_M	correction factor for mixtures defined in Eq. (8)
G	mass flux (kg m^{-2} s^{-1})
g	gravitational acceleration (m s^{-2})
h	heat transfer coefficient (kW m^{-2} K^{-1})
h_{fg}	latent heat (kJ kg^{-1})
k	thermal conductivity (kW m^{-1} K^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
$\Delta\dot{m}$	mass flow rate change (kg s^{-1})
Pr	Prandtl number, $\mu C_P/k$
q	transferred heat (kW)
q''	heat flux (kW m^{-2})
Re_l	Reynolds number, $G(1-x)D_i/\mu_l$
S	suppression factor defined in Eq. (7)
T	temperature (°C)
ΔT	temperature change (°C)

x	mass quality
X_{tt}	Martinelli parameter, $\{(1-x)/x\}^{0.9}\times(\rho_l/\rho_v)^{0.5}(\mu_l/\mu_v)^{0.1}$

Greek symbols

μ	viscosity (Pa s)
ρ	density (kg m^{-3})
σ	surface tension (N m^{-1})

Subscripts

ave	average
bub	bubble point
DB	Dittus–Boelter correlation
dew	dew point
k	top, bottom, or side
l	liquid
meas	measurement
pred	prediction
s	saturation
SA	Stephan–Abdelsalam correlation
tp	two phase
v	vapor
wi	inner wall

For refrigerant mixtures, boiling site density is affected by the mixture composition [20], and the bubble growth rate is reduced by the effect of mass transfer effect [21]. Therefore, it is expected that the heat transfer for HFC mixtures will be suppressed when compared with that for pure refrigerants for similar experimental conditions. Earlier studies on in-tube flow boiling revealed that liquid mixtures do not evaporate better than pure substances. Several explanations for this phenomenon were the non-linear property variation with respect to mixture composition and mass transfer resistance near the interface [7].

Currently, correlations by Jung et al. [2], Kandlikar [17], and Gungor & Winterton [18,19], which adopted the superposition or enhancement model have been used to predict evaporative heat transfer coefficients. In the superposition model used by Jung et al. [2] and Gungor and Winterton [18], two terms of nucleate boiling and convective evaporation contribute to the evaporative heat transfer. Nucleate boiling contribution is expressed as a pool boiling correlation multiplied by a suppression factor and convective evaporation contribution is expressed as a single phase heat transfer correlation multiplied by an enhancement factor.

This paper presents measured local heat transfer coefficients for two single component HFC refrigerants, R-32 and R-134a; HFC zeotropic mixtures of R-32/134a and R-32/125/134a (23/25/52 wt%) and R-22 as a reference refrigerant. Test conditions are chosen by considering actual refrigerator and heat pump operations. A new correlation based on the measured data is presented which can be applied for both pure refrigerants and refrigerant mixtures.

2. Experiment

2.1. Experimental apparatus

Schematic diagram of the experimental apparatus is shown in Fig. 1. The figure shows a loop for heat transfer coefficient measurement. The heat transfer test rig is composed of a magnetic pump, a mass flow meter, pre-heaters, a heat transfer test section, a heat exchanger for subcooling the fluid, a liquid receiver and a chiller. The magnetic pump circulates subcooled liquid from the liquid receiver to the pre-heater. This pump is driven by a magnet, so no oil is necessary for lubrication. The mass flow meter using Coriolis effect (estimated error of 0.23%) is installed before the pre-heater to measure the flow rate of the refrigerant in the liquid phase. Two pre-heaters are installed to adjust the inlet quality of the refrigerant to a desired

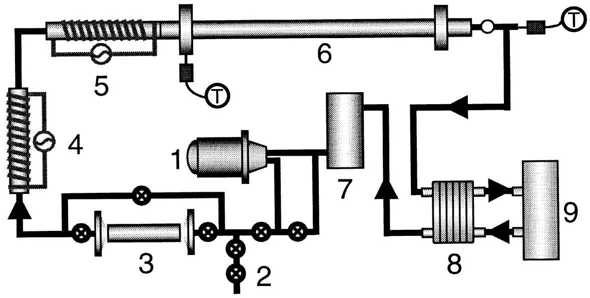

Fig. 1. Schematic diagram of experimental apparatus of evaporative heat transfer test (1: magnetic pump, 2: sampling port, 3: mass flow meter, 4: pre-heater, 5: controlled pre-heater, 6: main test section, 7: reservoir, 8: heat exchanger, 9: chiller).

value. It is expected that the preheating will change the average temperature of the test section because the test loop is a closed system. Therefore, it is necessary to use the pre-heater and chiller to adjust both the inlet quality and the test section average temperature. The refrigerant flowing out of the test section is subcooled by the secondary heat transfer fluid, a mixture of ethylene glycol and water.

The schematic diagram of the heat transfer test section is represented in Fig. 2. A seamless stainless steel tube is used for heat transfer test section in order to maintain electrical and thermal uniformity in the test section. The inner and outer diameters of the tube are 7.75 and 9.55 mm, respectively, and the heated length of the test section is 5.9 m. Heat is transferred to the refrigerant at the test section by applying electric current directly to the tube which is measured by a powermeter with an estimated error of 0.67%. Pressure taps are installed with an interval of 1 m to measure the pressure difference of each test sub-section. The pre-heater at the upstream of the test section and heat exchanger at the downstream of the test section are connected to the test section with teflon connectors to be electrically insulated from the test section. T type (copper–constantan) thermocouples to measure the temperatures of the outer wall of the tube are attached at 12 locations along the tube. In each location, the temperatures are measured at three or four circumferential positions of top, bottom, and one side or two sides.

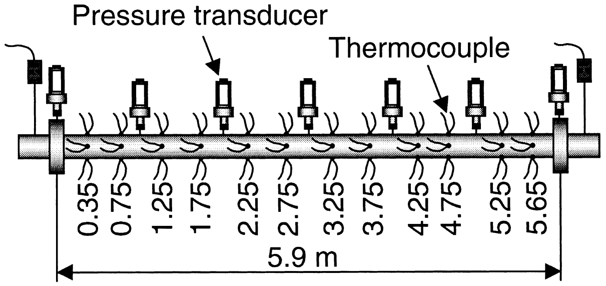

Fig. 2. Schematic diagram of the heat transfer test section (numbers indicate the length from the tube inlet).

2.2. Test conditions

Mass flux and heat flux are the major factors which affect evaporative heat transfer. The mass flux was varied from 240 to 1060 kg m^{-2} s^{-1} by using a magnetic pump, and heat flux ranged from 4.1 to 28.6 kW m^{-2}. The test section average temperature was defined as an average of calculated temperatures of refrigerant at each measuring point based on the measured pressure and estimated enthalpy considering the heat input to the refrigerant. The test section average temperature was varied from −12.0 to 17.0°C to investigate the influence of refrigerant temperature on heat transfer. Table 1 summarizes test conditions in detail for each refrigerant in this study.

2.3. Data reduction

While the main chiller was operated, the desired value of the heat flux was applied gradually to the test section until the steady state was reached. The measured steady state values were transmitted to the personal computer through a multi-channel recorder, using GPIB interface. The transmitted values were displayed on a monitor with their average value and standard deviation of 30 data sets. The data were saved in a file when the deviations of temperature, mass flux, and pressure were below 0.1°C, 3 kg m^{-2} s^{-1}, and 2 kPa, respectively. The properties of pure refrigerants and refrigerant mixtures were calculated by REFPROP [24]. For mixtures, mass quality was obtained considering the heat input to the test section, and this was used to calculate the local refrigerant temperature using the energy balance. The heat transfer coefficient is defined as in Eq. (1),

$$h_{\mathrm{k}} = \frac{q''}{T_{\mathrm{wi,\,k}} - T_{\mathrm{s}}} \tag{1}$$

where q'' is the heat flux and T_{wi} is the inner wall temperature calculated from the measured outer wall temperature by considering one-dimensional heat conduction through the wall. T_{s} represents saturation temperature for pure fluids, and T_{s} for mixtures are calculated considering vapor–liquid equilibrium of the mixture and energy balance using REFPROP [24]. Subscript k represents top, bottom, or side. Local heat transfer coefficient, h, at each location was obtained by

Table 1
Test conditions for measuring evaporation heat transfer coefficients

Parameter	R-22	R-134a	R-32	R-32/13a	R-407C
Average pressure (kPa)	360–660	200–430	570–1080	320–940	340–930
Average temperature (°C)	−9–7	−7–9	−9–9	−8–8	−12–17
Mass flux (kg m^{-2} s^{-1})	240–1060	240–650	240–850	240–850	240–850
Heat flux (kW m^{-2})	4.3–27.9	4.2–27.4	10.4–28.6	4.1–28.3	4.2–28.0

averaging heat transfer coefficients measured at the top, bottom, and two sides (or one side).

3. Experimental results

3.1. Verification of test accuracy

Single phase heat transfer tests for R-22 as a reference fluid have been done to check the accuracy in heat transfer measurement. One method for checking the accuracy is to compare the heat gained by the refrigerant with the heat supplied by the electricity through the wall. In 15 experimental data sets for R-22, the absolute average of deviation between the heat supplied and the heat gained by the refrigerant was about 3.1%. Also, the single phase heat transfer coefficient was calculated and compared with Gnielinski's correlation [23] for single phase flow. Absolute average of relative deviation was about 8.6%. The comparison with Dittus–Boelter correlation showed slightly higher relative deviation.

3.2. Evaporation test results

Measured heat transfer coefficients for R-22 at the top and bottom of the tube were compared with the data of Jung et al. [1]. As shown in Fig. 3(a) (test condition of Jung et al. [8]: test section average pressure of 400 kPa, mass flux of 353.6 kg m^{-2} s^{-1}, and heat flux of 17 kW m^{-2}), test results are consistent with those of Jung et al. [1] except for the high quality region. However, the data of Jung et al. are greater than ours at the same test condition because their heat flux and mass flux were slightly lower than ours. Heat transfer enhancement in the low quality region was observed due to a strong nucleation of bubbles which is dominant heat transfer mechanism in the low quality region. The heat transfer coefficients at the top of the tube are greater than those at the bottom for R-22. On the other hand, for the refrigerant mixture R-407C, there exists a region where the heat transfer coefficients at the bottom become greater than those at the top (Fig. 3(b)). As explained by Ross et al. [9], this phenomenon was ascribed to the local composition difference between the top and bottom liquid layers, where more volatile component is deficient in the thin liquid layer at the top. The experimental uncertainty in measuring the heat transfer coefficients during evaporation was estimated about 14%.

Fig. 4 shows variations of heat transfer coefficients with respect to test section average temperature for R-22, R-32/134a, and R-407C. At different evaporation temperature (or pressure), the wall superheat required for the formation of bubble changes; lower saturation

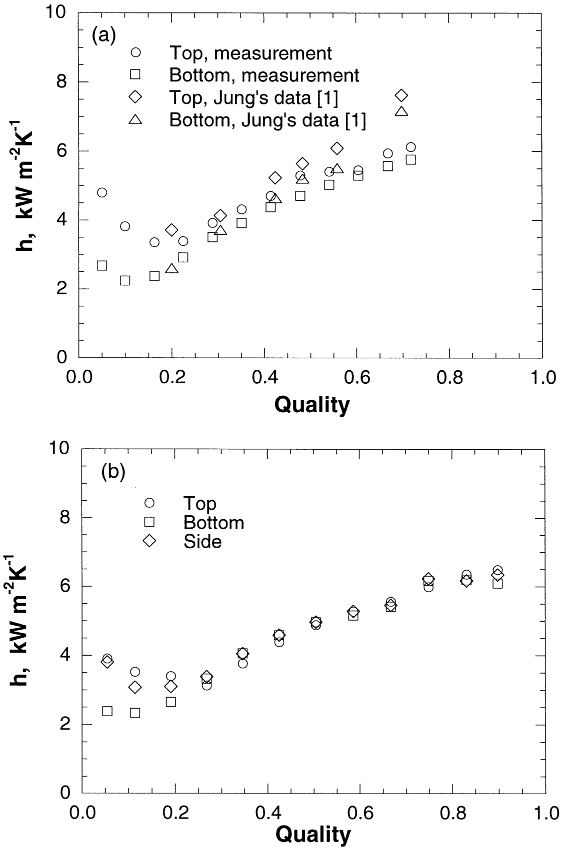

Fig. 3. Circumferential variations of heat transfer coefficients for R-22 and R-407C. (a) R-22; $G = 427$ kg m^{-2} s^{-1}; $q'' = 20.9$ kW m^{-2}; $T_{ave} = -3.0$°C (b) R-407C; $G = 427$ kg m^{-2} s^{-1}; $q'' = 27.9$ kW m^{-2}; $T_{ave} = -2.8$°C.

pressures require greater wall superheat than higher pressures. As Ross et al. [9] concluded that nucleate boiling became fully suppressed if pressure was low enough from their test at two different pressures of 1.8 and 5.7 bar for R-152a, lower heat transfer coefficients were obtained for low quality region at lower pressure (or temperature). Since the major heat transfer mechanism in annular flow regime is the heat conduction through the liquid layer, wall superheat will slightly decrease in the later stage of evaporation for pure refrigerant before the dryout point, which results in an increase of heat transfer coefficients. This is because the thickness of the liquid layer is decreased while the saturation (interface) temperature remains constant at uniform heat flux condition. As for refrigerant mixtures, the dependence of the heat transfer coefficients on the evaporation temperature was not so obvious. It should be noted that Wang et al. [12] and Kattan et al. [3] also indicated that the change in the heat transfer coefficient with respect to temperature is small for zeotropic refrigerant mixtures.

As the saturation temperature increases for the same heat flux and mass flux condition, latent heat of refrigerant decreases, which means greater amount of refrigerant should be evaporated to take the heat away from the wall. For pure refrigerant and mixtures, the heat supplied from the small segment of the wall is used to change the temperature of the liquid phase and to evaporate small amount of liquid. The relation is expressed as follows:

$$\Delta q = \dot{m}(1 - x)C_{p,1}\Delta T + \left(h_{fg} - C_{p,1}T\right)\Delta\dot{m} \tag{2}$$

For pure fluid, temperature change is negligible ($\Delta T \sim 0$), therefore $\Delta\dot{m}$ is greater than that for zeotropic refrigerant mixtures whose temperature glide for each segment of heat exchanger is generally positive ($\Delta T > 0$) for given Δq.

Variations of heat transfer coefficients with respect to mass flux for R-22, R-32/134a, and R-407C are shown in Fig. 5. Eckels and Pate [6] showed that the heat transfer is much enhanced in the annular flow than in the stratified wavy flow. Generally, the annular flow pattern can be obtained more easily as mass flux increases, and this change in flow pattern affects the heat transfer. The heat transfer coefficient increases as the mass flux increases, since greater number of molecules are taking the heat from the wall per unit time by reducing the wall temperature (or wall superheat). Another factor which influences the enhancement of heat transfer is the wetting of the wall by the liquid film with higher velocity (higher mass flux), which results in rather uniform liquid film thickness. Turbulence in the liquid phase and the disturbance by the interfacial waves for high mass flux also cause heat transfer enhancement by increasing the mixing effect. As shown in Fig. 5, heat transfer coefficients increase for higher mass flux at the same mass quality for all pure refrigerants and refrigerant mixtures. One thing to note is that the heat transfer coefficient increases as mass quality increases, which is due to the increased velocity of refrigerant resulting from the increased specific volume with evaporation in a circular tube.

Fig. 6 represents the variations of heat transfer coefficients with respect to heat flux for R-22 and R-407C. As heat flux increases, heat transfer coefficient increases greatly in the low quality region. In this region, nucleate boiling is dominant and the heat transfer coefficient is strongly dependent on heat flux

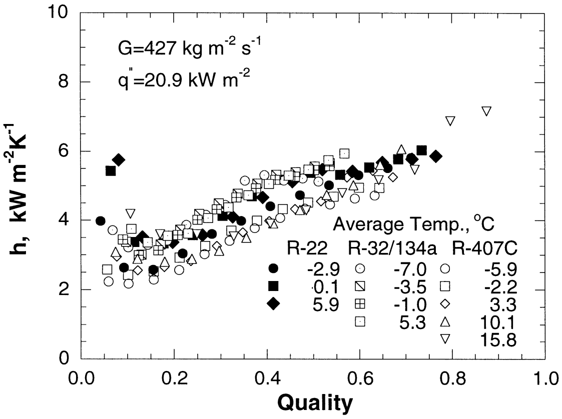

Fig. 4. Variations of heat transfer coefficients with respect to test section average temperature for R-22, R-32/134a, and R-407C.

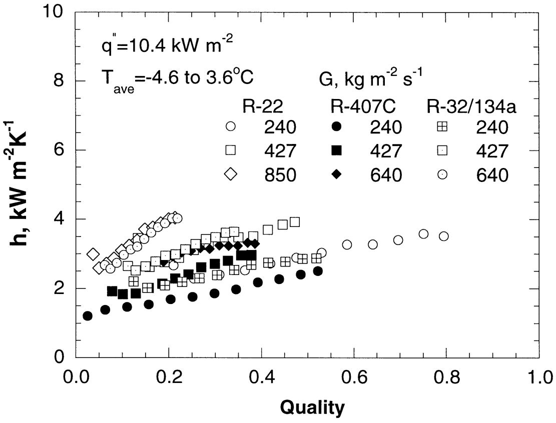

Fig. 5. Variations of heat transfer coefficients with respect to mass flux for R-22, R-32/134a, and R-407C.

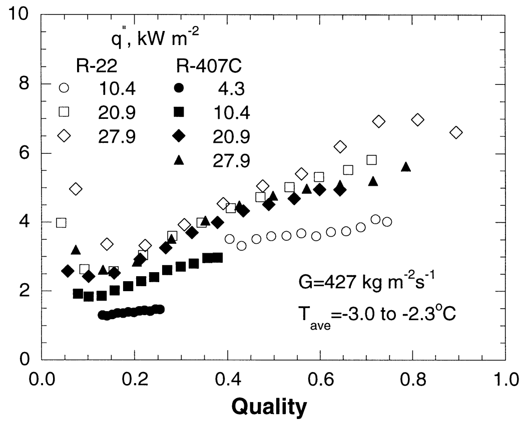

Fig. 6. Variations of heat transfer coefficients with respect to heat flux for R-22 and R-407C.

[16]. However, in the high quality region where the convective evaporation is dominant, the influence of heat flux on heat transfer is reduced, and for higher heat flux above a certain level, the influence becomes quite minor. The partial boiling region is also shown in the figure, where both the forced convection and nucleate boiling are comparable and dependent upon heat flux. The fact that the heat transfer coefficient is not so sensitive to the heat flux in the convection-dominant region was also observed by Ross et al. [9] and Jung et al. [1].

In Fig. 7, the heat transfer coefficients of R-32/134a with respect to the composition of R-32 are illustrated at the condition of mass flux of 427 kg m^{-2} s^{-1}, heat flux of 20.9 kW m^{-2}, and test section average temperature from −1.3 to −0.5°C. As pointed out previously, the non-linear mixture properties and mass transfer resistance due to concentration gradient near the interface are the main causes for the heat transfer degradation of refrigerant mixtures [1,9]. Fig. 7 shows the heat transfer degradation for refrigerant mixtures with the heat transfer coefficients of pure components which are comprised in the mixture. It is observed that the heat transfer coefficients of refrigerant mixtures are lower than the linearly interpolated values (solid line in the figure) between the pure components. The maximum heat transfer degradation occurs approximately between 50 and 75% of the composition of R-32 for both quality ranges.

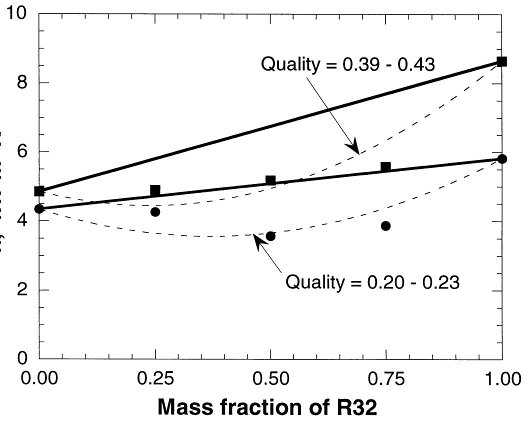

Fig. 7. Heat transfer coefficients for several compositions of R-32 in R-32/134a mixture for G = 427 kg m^{-2} s^{-1}; $T_{ave} = -1.3$ to -0.5°C; $q'' = 20.9$ kW m^{-2}.

4. Correlation

4.1. Correlation for pure refrigerants

In two phase flow inside tubes with heat addition, the mass quality and void fraction increase as the refrigerant evaporates. The liquid film thickness becomes thinner and the flow velocity increases in this process due to the density difference between the liquid and vapor phase. The heat transfer characteristics are dependent on the quality, which represents different flow regimes. As stated previously, the heat flux and mass flux have influences on heat transfer characteristics of refrigerants. Considering these factors, the following three dimensionless parameters such as Bo, Co, and X_{tt} are chosen for the correlation for pure refrigerants.

Both the enhancement factor and the suppression factor of the superposition model are used to make a correlation for the refrigerants in this study. The basic form of the correlation is shown in Eq. (3).

$$h_{tp} = Eh_{DB} + Sh_{SA} \tag{3}$$

where h_{DB} represents the correlation by Dittus and Boelter for pool boiling [22], and h_{SA} implies the correlation by Stephan and Abdelsalam for single phase heat transfer [3]. h_{DB} and h_{SA} are expressed in Eqs. (4)–(5b).

$$h_{DB} = 0.023 Re_l^{0.8} Pr_l^{0.4} \tag{4}$$

$$h_{SA} = 207 \frac{k_l}{bd} \left(\frac{q'' bd}{k_l T_s} \right)^{0.674} \left(\frac{\rho_v}{\rho_l} \right)^{0.581} Pr_l^{0.533} \tag{5a}$$

$$bd = 0.0146 \beta \left[2\sigma / g(\rho_l - \rho_v) \right]^{0.5} \quad \text{with } \beta = 35° \tag{5b}$$

The enhancement factor is represented by Eq. (6), and the form of suppression factor, S, is selected as in Eq.

(7).

$$E = c_1 Bo^{c_2} X_{tt}^{c_3} \tag{6}$$

$$S = c_4 Co^{c_5} \tag{7}$$

Constants, c_1–c_5 in Eqs. (6) and (7) are obtained by an iteration process to minimize the errors between the heat transfer coefficients calculated from the above correlation and the experimental results. These constants are listed in Table 2.

4.2. Correction of the correlation for refrigerant mixtures

By considering the effect of heat flux and the temperature difference between the dew point and bubble point, Thome and Shakir [15] introduced a mixture correction factor. It varies from 1.0 at very low heat fluxes, and to a lower value for the boiling range at large heat fluxes. In fact, this factor has been shown to work for a very wide variety of binary and multicomponent mixtures. However, the effect of heat flux on heat transfer coefficient has already been considered in h_{SA} (see Eq. (5a)) and in Bo represented by Eq. (6), a simple form of correction factor for mixtures are proposed in this study. This correction factor for mixtures, F_M is used with the above factors of S and E in Eqs. (6) and (7) with the properties of refrigerant mixture. The heat transfer coefficients for zeotropic or azeotropic refrigerant mixtures are calculated using the following Eqs. (8) and (9).

$$h_{tp} = Eh_{DB} + F_M S h_{SA} \tag{8}$$

$$F_M = \frac{1}{1 + c_M\{[C_{pl}(T_{dew} - T_{bub})]/h_{fg}\}} \tag{9}$$

In Eq. (9), $T_{dew} - T_{bub}$ is always positive, and c_M has the positive value. Thus, F_M is always less than 1, which represents the heat transfer degradation for zeotropic refrigerant mixtures. This equation can be applicable to pure refrigerants, since F_M becomes 1 for pure refrigerants of which both dew point and bubble point temperatures are the saturation temperature at a given pressure. The value of c_M is represented in Table 2.

Table 2
Constants in Eqs. (6), (7) and (9)

c_1	c_2	c_3	c_4	c_5	c_M
49.971	0.383	−0.758	0.909	0.301	0.039

Table 3
Comparison of experimental data in this study with the correlations in the open literature[a]

Refrigerants	No. of data	Gungor and Winterton [18]			Gungor and Winterton [19]			Jung et al. [2]			Kandlikar [17]			This study		
		Bias	AAD	RMS	Bias	AAD	RMS	Bias	AAD	RMS	Bias	AAD	RMS	Bias	AAD	RMS
R-22	551	−27.9	28.3	31.2	−18.2	21.9	25.4	−33.4	34.8	38.3	−8.5	15.9	19.7	−2.8	10.5	13.5
R-134a	363	−26.7	27.0	30.0	−18.2	20.5	24.0	−34.6	35.3	38.8	−14.8	20.1	24.3	−10.8	12.2	15.3
R-32	262	−17.2	21.6	24.5	−4.0	18.0	21.8	−17.5	24.6	28.6	8.7	17.7	24.3	16.3	18.3	24.2
R-32/134a	1137	−37.3	37.4	39.7	−26.8	27.1	30.0	−28.5	29.1	32.1	–	–	–	−1.4	11.8	15.4
R-407C	658	−43.6	43.9	46.0	−34.4	35.4	38.5	−41.2	41.2	44.3	–	–	–	−12.1	16.4	19.1
Total	2971	−33.9	34.5	37.6	−23.8	26.4	30.1	−32.0	33.2	36.9	−6.6	17.6	22.3	−3.6	13.2	16.9

[a] Bias error, %: $\frac{1}{N}\sum[(h_{meas} - h_{pred})/h_{pred}] \times 100$; absolute average of deviation (AAD), %: $\frac{1}{N}\Sigma|(h_{meas} - h_{pred})/h_{pred}| \times 100$; root mean square of deviation (RMS), %: $\sqrt{\frac{1}{N}\Sigma[(h_{meas} - h_{pred})/h_{pred}]^2} \times 100$.

Table 4
Comparison of experimental data in the open literature with the correlations

Data source	Refrigerants	No. of data	Gungor and Winterton [18]			Gungor and Winterton [19]			This study			Test condition
			Bias	AAD	RMS	Bias	AAD	RMS	Bias	AAD	RMS	
Wang et al. [12]	R-22	30	1.1	22.9	26.3	40.1	47.7	59.5	15.3	31.6	37.4	$q'' = 6\text{–}14$ kW m^{-2} $G = 100\text{–}300$ kg m^{-2} s^{-1} $P_{sat} = 600\text{–}680$ kPa Quality range = 0.1–0.8 Second fluid (water) heating (counter flow)
	R-407C	37	−50.9	50.9	53.0	−33.1	36.7	41.0	−21.8	34.7	40.8	
Murata and Hashizume [8]	R-123	29	4.7	19.5	23.5	27.6	29.0	38.7	−4.6	19.1	21.7	$q'' = 10, 30$ kW m^{-2} $G = 100\text{–}300$ kg m^{-2} s^{-1} $P_{sat} = 200$ kPa Quality range = 0.1–1.0 Electrically sheathed heating
	R-123/134a (90/10 mol%)	38	−13.3	22.4	25.7	6.3	18.6	23.2	−2.9	21.5	24.9	
Takamatsu et al. [11]	R-22/114 (51/49 mol%)	13	−19.8	20.9	25.2	8.6	11.8	13.4	4.0	13.7	20.0	$q'' = 1.9\text{–}60$ kW m^{-2} $G = 290$ kg m^{-2} s^{-1} $P_{sat} = 1140$ kPa Quality range = 0.0–1.0 Second fluid (water) heating (counter and parallel flow)
Total		147	−16.9	29.0	34.4	7.7	30.5	39.9	−3.6	25.7	31.5	

4.3. Comparison of experimental results with several correlations

For pure refrigerants, the correlations of Jung et al. [2], Gungor and Winterton [18,19], and Kandlikar [17] were used for comparison. The correlation of Jung et al. [2] and that of Gungor and Winterton [18,19] with Thome and Shakir correction factor for mixtures [15] were compared with experimental results for refrigerant mixtures. Comparison results are presented in Table 3

The correlation by Gungor and Winterton [18] was developed from a large data bank including water, refrigerants and ethylene glycol, and it has been widely used to predict the experimental heat transfer coefficients. However, their correlation overpredicted the experimental data in this study. Although the correlation by Kandlikar [17] predicted the heat transfer coefficients for pure refrigerants with fair accuracy, there was a shortcoming that the fluid dependent parameter should be determined for each working fluid. His correlation was not applied for refrigerant mixtures. For refrigerant mixtures, the correlation by Jung et al. [2] overpredicted the experimental heat transfer coefficients. Comparatively, the correlation by Gungor and Winterton [19] predicted the heat transfer coefficients well, but with an overprediction. The correlation of Jung et al. [2] was developed originally for binary mixtures and the ideal heat transfer coefficient in the correlation of Jung et al. [2] was slightly modified for ternary mixtures to indicate the harmonic average of each pure component's heat transfer coefficient. Thus, caution should be taken when using Jung et al.'s correlation for ternary mixtures.

Recently, many works have been carried out on evaporation heat transfer of HFC refrigerants. Electrical heating and secondary fluid heating method were used to transfer heat to the test section. Electrical heating method is more popular than the secondary fluid heating method because of the higher accuracy in determining the heat flux. However, secondary heating method gives relatively constant temperature condition, which is thought similar to the real situation. In Table 4, experimental conditions and heating methods from the literature are listed. It also shows comparison of test results from several researchers [8,11,12] with the correlation developed in this study. Comparatively better results were obtained for the correlation of this study than the correlations by Gungor and Winterton [18,19]. Fig. 8 represents the relative errors between the calculated values from the correlation of this study and the experimental results obtained in this study. The absolute average of deviation between the two is about 13.2%.

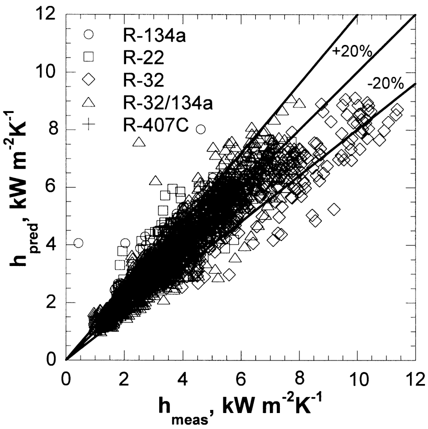

Fig. 8. Comparison between predicted and experimental heat transfer coefficients.

5. Concluding remarks

Evaporative heat transfer characteristics of HFC refrigerants, R-32, R-134a, R-32/134a, and R-407C, have been investigated in this study. Experimental results were compared with several correlations for evaporative heat transfer coefficients, and a new correlation was presented based on the experimental data in this study. This new correlation was used to compare other researchers' experimental data, which gives a slightly better result than the correlation by Gungor and Winterton [18,19]. The conclusions of this study are summarized as follows:

1. As the evaporation temperature increased, heat transfer coefficients became greater for the same mass quality. In the low quality region, dependence of heat transfer coefficients on evaporation temperature was observed for pure refrigerants. On the other hand, for refrigerant mixtures, influence of temperature in the low quality region on heat transfer coefficient was weak. For high quality region, the effect of evaporation temperature was minor for both pure refrigerants and refrigerant mixtures.
2. The influence of mass flux and heat flux was also investigated. The heat transfer coefficient increased as the mass flux increased. In the low quality region, the influence of heat flux on heat transfer coefficient was obvious, however, the influence was reduced for high heat flux above a certain magnitude in the high quality region.
3. A new heat transfer correlation was presented by using a superposition model to predict the experimental data for both pure refrigerants and refrigerant mixtures. A correction factor which represents the heat transfer degradation for zeotropic refriger-

ant mixtures was introduced. Absolute average deviation between the predicted and measured data was 13.2% based on 2971 data points obtained from the experiment for pure refrigerants and refrigerant mixtures.

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