Pool boiling Heat Transfer Characteristics of Low GWP Refrigerants on Enhanced tube used in Flooded Evaporator for Turbo-Chiller

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Abstract

Recently, researches on the finding a next-generation refrigerant that does not destroy the ozone layer or affect on global warming are growing globally. In this study, pool boiling heat transfer characteristics of low GWP refrigerants on plain and enhanced tube were investigated experimentally. The measurement of heat transfer coefficients was carried out through the Wilson plot method. The method was validated and used to characterize the inside heat transfer coefficient of the cooling water. Tests were performed for three different refrigerants (R-134a, R-1234ze(E), R-1233zd(E)) at two different saturation temperatures 4.4 ℃ and 26.7 ℃, heat flux was varied from 10 to 50 kW/m^2. Results were shown that the pool boiling heat transfer coefficients of R-1234ze(E) was obtained 5.0% ~ 11.5% lower than the those of R-134a. Heat transfer coefficients of R-1233zd(E) were 64.6% ~ 71.7% lower than those of R-134a. Compared with the heat transfer performance of the plain tube, the heat transfer enhancement ratios of enhanced tube were obtained 2.3 for R-134a, 2.1 for R-1234ze(E) and 1.8 for R-1233zd(E) respectively.

Keywords: Low GWP refrigerant; Pool boiling; Enhanced tube; Wilson plot; Flooded evaporator

1. Introduction

The revised 2014 Montreal Protocol proposed a stepwise reduction of the production and consumption of HFC refrigerants from 2015 to 2035. In compliance with the protocol, the production and consumption of refrigerants should be reduced to 85% of current levels until 2045. Thus, global refrigerant manufacturers have been competitively committed to developing low GWP refrigerants. [1] Currently, large chiller manufacturers have already launched products using low GWP refrigerants.

Recent experimental researches of pool boiling heat transfer performance on new refrigerants (e.g., R-1234ze(E), R-1233zd(E)), especially low GWP refrigerants, are discussed below.

Rooyen and Thome [2] conducted pool boiling heat transfer performance of R-134a, R-236fa and R-1234ze(E) with latest enhanced tubes (Turbo-B5 and Gewa-B5). Test was conducted at the saturation temperature of 5~25℃ and heat fluxes from 15 to 70 kW/m^2. The R-236fa showed lower heat transfer coefficient than R-134a for all temperatures and heat fluxes and R-1234ze(E) showed similar performance compared to R-134a. They showed the nearly constant heat transfer coefficient over the heat fluxes for Turbo-B5 or decrease of heat transfer coefficient was observed for Gewa-B5 with increase of heat fluxes. They proposed the pool boiling heat transfer correlation for enhanced tube based on the boiling mechanism in the near wall region. The film thickness equation was proposed by thermal properties and heat transfer coefficient was suggested as a function of film thickness and geometric factors. The proposed correlation showed that Gewa-B5 tube is superior to Turbo-B5 at

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low heat flux condition (20~40 kW/m^2) and Turbo-B5 tube showed better performance at high heat flux conditions (40~70 kW/m^2).

Nagata et al. [3] used R-1234ze(E), R-1234ze(Z) and R-1233zd(E) to test the pool boiling and condensation with plain tubes at evaporating temperatures of 10 ~ 60℃ and heat fluxes of 0.7 ~ 80 kW/m^2. They compared the pool boiling experiment results of low GWP refrigerants against Ribatski's equation, and reported good predictability. They attributed the differences in heat transfer between refrigerants to the thermodynamic physical properties based on the surface tension data [4], an important property in pool boiling heat transfer.

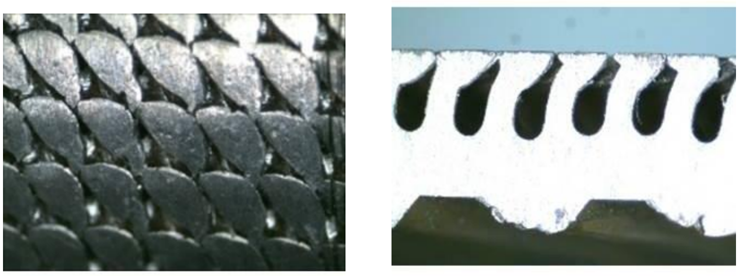
Evraam Gorgy [5] measured the pool boiling heat transfer coefficients of three refrigerants, i.e., R-1234ze, R-1233zd(E), and R-450A, in enhanced tubes, using Wilson plot method, at a saturated temperature of 4.44℃ and heat flux 10 ~ 110 kW/m^2. The heat transfer performance of R-1234ze(E) nearly paralleled that of R-134a, whereas R-450a showed 28% lower performance. R-1233zd(E), a substitute for R-123, showed better than R-123 by approximately 19%. The experimental data was represented as the function of heat flux ranging from 10 kW/m^2 to 60 kW/m^2.

As aforementioned, published literatures on pool boiling with low GWP refrigerants are not sufficient. Moreover, previous studies were focused on plain tubes only. Thus, articles on the enhanced tubes hardly exist. Within a short future, most refrigeration and HVAC products will displace the existing refrigerants and adopt low GWP refrigerants, which continuous research will be needed. For the present study, 2 low GWP(<100) refrigerants (R-1234ze(E) and R-1233zd(E)) were selected to verify the pool boiling heat transfer performance of the plain tube and the commercial enhanced tube.

2. Experiments

The enlarged photos of the tunnels and surfaces of the present tubes were shown in Fig. 1. An internally and externally unprocessed plain tube (d_o= 19.05 mm, t=1.3 mm) was pre-tested as a reference tube. The enhanced 1 tube was processed internally and externally, and appeared similar to the warped cross-section of the widely known commercial Turbo-B tube. The details of dimension were shown in Table 1, where indicated values are actual measurements.

A schematic drawing of the experimental apparatus was shown in Fig. 2. The test section had a stainless steel shell measuring 300 mm in inside diameter and 1,000 mm in length. The test tube was inserted at the center of the shell. As the heat source inside the tube, water was supplied from a thermostatic bath. Refrigerants filled the space outside the tube, or inside the shell, up to approximately 50 mm of the enhanced tube, forming a flooded type. Inside the shell, the gas and liquid temperatures were measured at 2 spots with an RTD sensor respectively, whose arithmetic mean values were used to calculate the saturated pressure and other physical properties. Major physical properties were calculated from Refprop 9.0 [6]. The gas refrigerant evaporating on the tube surface entered into the three ports located at the top of the shell and condensed in the condenser located at the top of the shell. The condensed liquid refrigerant was in turn supplied to the three ports installed at the lower part of the shell by natural circulation. Here, no separate refrigerant pump was constructed. At the center of the shell was an observation window installed to capture the phenomena occurring inside the shell and on the enhanced tube surface, e.g., bubble site density, growth and departure, with an ultrahigh-speed camera. A single tube was used in the experiment, with 1,000 mm (L) and unprocessed outside diameter (d_o) of the enhanced tube was used to calculate the heat transfer area (A_o=$\pi d_o L$). Table 1 shows the specifications of the enhanced tube as well as the plain tube used in the present study.



(a) external surface (b) cross-sectional view

Fig. 1. Enlarged view of the enhanced tube

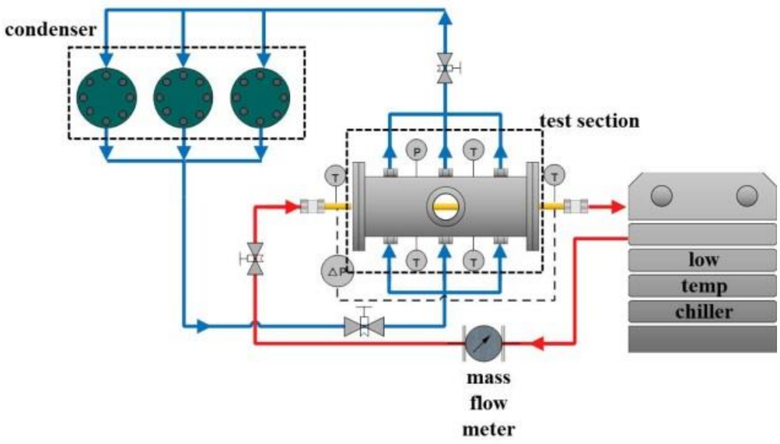


Fig. 2. Experimental apparatus

Table 1. Detail geometry of test tubes

	D_o [mm]	D_i [mm]	t [mm]	β [º]	N [-]	h_{groove} [mm]	FPI [-]	h_{fin} [mm]
Plain tube	19.05	16.65	1.2	-	-	-	-	
Enhanced tube	18.99	16.73	1.12	45	38	0.34	55	0.61

3. Data reduction

The heat transfer rate supplied to the test section, overall heat transfer coefficient and log mean temperature difference (LMTD) were calculated as in the equation (1) and (2). Here, Wilson plot method [7] was used to experimentally yield the tube-side heat transfer coefficient (h_i) with the Dittus-Boelter expression in equation (3). The U_oA_o experimentally calculated was used to find the pool boiling heat transfer coefficients on the outer surface of the tubes.

$$Q = \dot{m}C_p(T_{wi} - T_{wo}) = UA_o\Delta T_{LM} \tag{1}$$

$$\frac{1}{U_oA_o} = \frac{1}{h_iA_i} + \frac{1}{h_oA_o} + R_{wall} \tag{2}$$

$$Nu_i = \frac{h_id_i}{k_i} = CRe^nPr^{0.3} \tag{3}$$

The Wilson plot method, which was used in this study, is widely applied when the heat transfer coefficients inside and outside the tube are unknown. Here, the method is not elucidated in detail as it has been described in many previous studies [9][10].

First, the plain tube was used to conduct the Wilson plot. The result was used to derive the in-tube heat transfer coefficient, which was compared against the Dittus-Boelter correlation equation [8], a well-known correlation equation for the fully developed turbulence in a smooth tube. The heat transfer coefficient of the enhanced tube with processed inner grooves was 45% higher than that of the plain tube.

Experimental uncertainties were analyzed with the method by Kline and McClintock [11]. According to Kline and McClintock [11], when a parameter 'R' is a function of measured variables (x_1; x_2; . . . ; x_n), the uncertainty on the parameter 'w_R' is obtained from the following equation (4). Here, 'w_1' is the uncertainty on variable 'x_1', 'w_2' is the uncertainty on variable 'x_2', etc. From the equation (1), parameters causing the experimental errors can be chosen. Any flow and temperature errors lead to heat flux (q") errors, which can be represented as the equation (5). After all, the h_o error can be calculated with h_i and U_o variables using the equation (6).

$$w_R = \left[\left(\frac{\partial R}{\partial x_1}w_1\right)^2 + \left(\frac{\partial R}{\partial x_2}w_2\right)^2 + \cdots\left(\frac{\partial R}{\partial x_n}w_n\right)^2\right]^{1/2} \tag{4}$$

$$w_{q''} = \left[\left(\frac{\partial q''}{\partial \dot{m}} w_{\dot{m}}\right)^2 + \left(\frac{\partial q''}{\partial \Delta T} w_{\Delta T}\right)^2\right]^{1/2} \tag{5}$$

$$w_{h_o} = \left[\left(\frac{\partial h_o}{\partial U_o} w_{U_o}\right)^2 + \left(\frac{\partial h_o}{\partial h_i} w_{h_i}\right)^2\right]^{1/2} \tag{6}$$

Introducing the measurement uncertainties listed in Table 2, the uncertainty on the overall heat transfer coefficients ranged ±5.4–45.8%, and that on pool boiling heat transfer coefficients ranged ±11.4–46.9%, which increased as heat flux decreased. The error values were shown as error bar along with the data.

Table 2. Uncertainties of measuring instruments and the heat transfer coefficients

Parameter	Uncertainty
Temperature [K]	±0.1 K
Absolute Pressure [kPa]	±0.2% of full scale
Mass flow rate [kg/hr]	±0.1% of full scale
Tube-side HTC h_i [kW/m^{2}K]	±10.0%
Overall HTC U_o [kW/m^{2}K]	±5.4 ~ 45.8%
Pool boiling HTC h_o [kW/m^{2}K]	±11.4 ~ 46.9%

4. Results and discussions

4.1 Plain tube

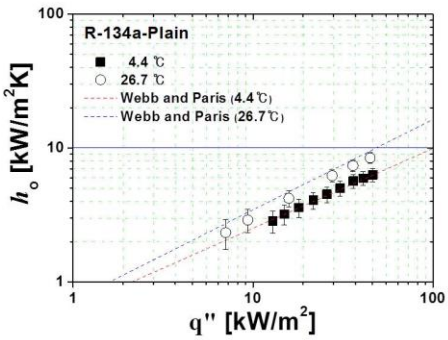
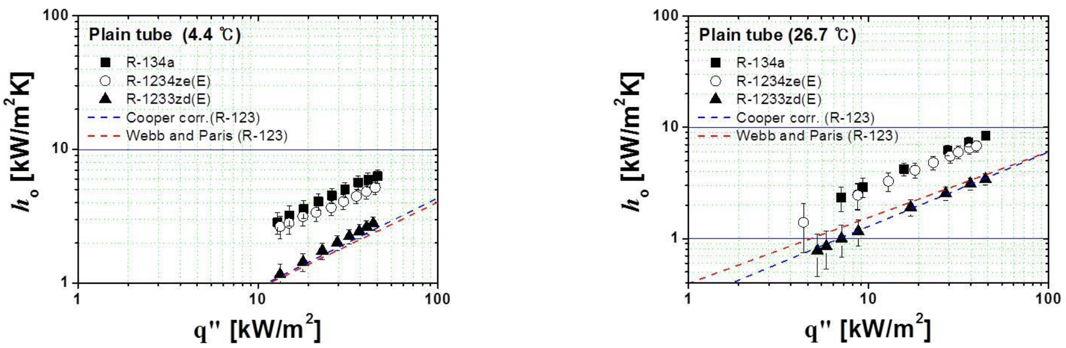


Fig. 3. Pool boiling heat transfer coefficients of the plain tube compared with Webb and Paris curve fit



(a) Tsat = 4.4 ℃ *(b) Tsat=26.7 ℃*

Fig. 4. Pool boiling heat transfer coefficients of the plain tube

Prior to testing the heat transfer performance of the low GWP refrigerants, a plain tube was used to pretest the existing refrigerant(R-134a), which was in turn used to verify the test unit built in this study. The R-134a was used at saturated temperatures 4.4℃ and 26.7℃ and heat flux 10~50 kW/m^2. Fig. 3 shows the heat transfer coefficient of the plain tube using R-134a. Also, the result was compared against the Webb and Paris [12]'s

Table 3 Representative properties of test refrigerants

refrigerants	M [g/mol]	P_{cr} [MPa]	σ [N/m]	P_{sat} [MPa]	ρ_l [kg/m^3]	ρ_g [kg/m^3]	P_r [-]
R-134a	102.0	4.06	0.0108	0.374	1280.1	16.8	0.09219
R-1234ze(E)	114.0	3.63	0.0116	0.254	1227.2	13.6	0.06985
R-1233zd(E)	130.5	3.62	0.0173	0.058	1311.2	3.4	0.01606
R-123	152.9	3.66	0.0177	0.040	1515.4	2.7	0.01086

experimental equation, which proved good predictability under both temperature conditions.

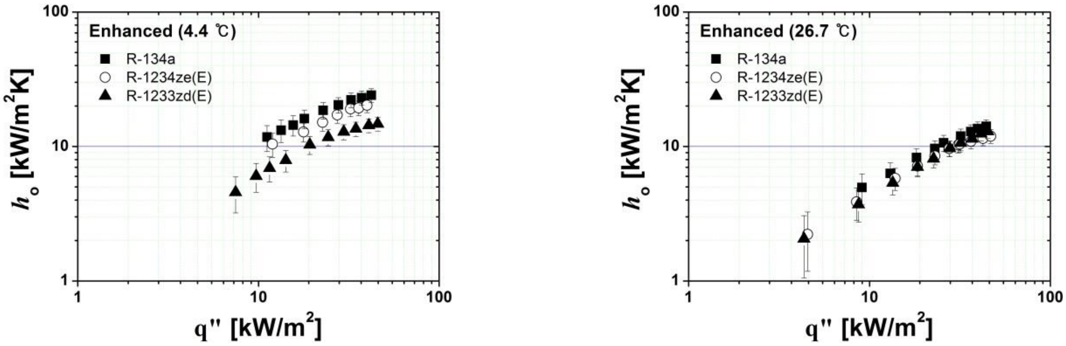
Fig. 4 shows the pool boiling heat transfer coefficients on the plain tube at T_{sat} = 4.4 and 26.7℃. R-134a showed the highest heat transfer coefficient. The heat transfer coefficients of R-1234ze(E) were slightly lower than those of R-134a, which seems attributable to the thermodynamic physical properties in the refrigerants. Table 3 outlines the physical properties of interest. The molar mass is presented in ascending order, which implies some points. The surface tension and the reducing pressure are presented in ascending and descending orders, respectively. Many earlier studies reported the higher the reducing pressure and the smaller the surface tension, the greater the thermal performance of refrigerants. In particular, the surface tension is known as a very important determinant of the behavior and departure of bubbles. As in the formula (7), Fritz [13] proposed a model based on Rayleigh's theory. This indicates the balance between buoyancy and surface tension in a thermodynamic equilibrium state. The heat transfer coefficient of R-1233zd(E) is remarkably smaller than those of the other two refrigerants, which seems attributable to the foregoing thermodynamic properties.

R-1233zd(E) is an alternative refrigerant replacing R-123. Yet, the present study did not experiment on R-123's heat transfer, but used the existing correlations for data comparison as marked with the dotted line in Fig. 4. Gorgy and Eckels [14] used plain and enhanced tubes to experiment on the performance of R-123 and R-134a, and reported Cooper correlation equation's good predictability of R-123 in a plain tube. Also, the data were compared against the experimental equation for plain tubes suggested by Webb and Paris [12]. R-1233zd(E) almost paralleled R-123 in terms of heat transfer coefficients with the difference falling within the error range. Fig. 4 (b) shows the data at 26.7℃, indicating a similar trend to the data at 4.4℃.

4.2 Enhanced tube

Pool boiling heat transfer coefficients of the enhanced tube were shown in Fig. 5. As with the plain tube, heat transfer coefficients of R-134a proved highest, whilst those of R-1234ze(E) and R-1233zd(E) were 15.2% and 41.2% lower than that of R-134a. Notably, in the enhanced tube compared with the plain tube, the heat transfer coefficient of R-1233zd(E) was quite high, which seems to be related with the shape of the tube surface where boiling occurred.

Kim and Choi [15] reported optimal gap sizes varied with refrigerants, with the high-pressure refrigerant R-134a's optimal gap size being larger than that of the low-pressure refrigerant R-123. Also, Nakayama [16] proposed an optimal pore size using R-11 and liquid nitrogen, and reported the optimal pore size of liquid nitrogen characterized by the high reducing pressure was proved larger than that of R-11. The low-pressure refrigerant R-1233zd(E) seems to account for the higher heat transfer coefficient of the enhanced tube. These findings are based on limited data, which need be clarified through further studies. At 26.7℃, the heat transfer coefficients of R-134a, R-1234ze(E), R-1233zd(E) showed nearly similar to each other.



(a) T_{sat} *= 4.4 ℃* *(b)* T_{sat}*=26.7 ℃*

Fig. 5. Pool boiling heat transfer coefficients of the enhanced tube

5. Conclusions

The present study was focused on the low GWP refrigerants of R-1234ze(E) and R-1233zd(E) among other alternative refrigerant candidates and experimented on two types of tubes, i.e., a plain and enhanced tube, in terms of pool boiling heat transfer performance. The test was conducted under the experimental condition at saturated temperatures 4.4℃ and 26.7℃ and heat flux of 10~50 kW/m^2. Wilson plot method was used to derive heat transfer coefficients of tube side, which were compared against previous empirical equations. R-1234ze (E) showed a slightly lower performance than that of R-134a. The heat transfer coefficients of R-1233zd(E) were largely similar to those of R-123 calculated from existing correlation. Compared with the heat transfer performance of the plain tube, the heat transfer enhancement ratios of enhanced rube were obtained 2.3 for R-134a, 2.1 for R-1234ze(E) and 1.8 for R-1233zd(E).

Acknowledgements

This research was supported by the Technology Innovation Program (Grant No. 10052926) by the Ministry of Trade, Industry & Energy

References

[1] Goetzler W, Sutherland T, Rassi M, Burgos J. Research & development roadmap for next-generation low global warming potential refrigerants. US DOE 2014.
[2] Van Rooyen E, Thome J. Pool boiling data and prediction method for enhanced boiling tubes with R-134a, R-236fa and R-1234ze(E). Int. J. Refrig. 2013;36(2):447-455.
[3] Nagata R, Kondou C, Koyama S. Comparative assessment of condensation and pool boiling heat transfer on horizontal plain single tubes for R1234ze(E), R1234ze(Z), and R1233zd(E). Int. J. Refrig. 2016;63:157-170.
[4] Kondou C, Nagata R, Nii N, Koyama S, Higashi Y. Surface tension of low GWP refrigerants R1243zf, R1234ze(Z), and R1233zd(E). Int. J. Refrig. 2015;53:80-89.
[5] Gorgy E. Nucleate boiling of low GWP refrigerants on highly enhanced tube surface. Int. J. Heat Mass Transf. 2016;96:660-666.
[6] REFPROP, NIST Refrigerant Properties Database 23, Gaithersburg, MD20899, Ver. 9.1
[7] Wilson EE, A basis for rational design of heat transfer apparatus, Trans. ASME J. Heat Transf. 1915;37:47-70.
[8] Dittus FW, Boelter LMK. : Univ. Calif. (Berkeley) Pub. Eng. 2 (1930) 443.
[9] Ferna´ndez-Seara J, Uhı´a FJ, Sieres J, Campo A, general review of the Wilson plot method and its modifications to determine convection coefficients in heat exchange devices, Applied Therm. Eng. 2007;27:2745-2757.
[10] Kim NH, Lee EJ, Byun HW, Evaporation heat transfer and pressure drop of R-410A in flattened smooth tubes having different aspect ratios, Int. J. Refrig. 2013;36:363-374.
[11] Kline SJ, McClintock F. Describing uncertainties in single-sample experiments, Mech. Eng. 1953;75 (1):3-8.
[12] Webb RL, Paris C. Nucleate pool boiling data for five refrigerants on plain, integral-fin, and enhanced tube geometries, Int. J. Heat Mass Transf. 1992; 35(8):1893-1904.
[13] Fritz,W, Berechnung des Maximal Volumens von Dampfblasen (Maximum volume of vapor bubbles). Physikalishe Z. 1935;36 :379-384.

[14] Gorgy E, Eckels S. Average heat transfer coefficient for pool boiling of R-134a and R-123 on smooth and enhanced tubes (RP-1316), HVAC&R Research 2010;16(5):627-676.
[15] Kim NH, Choi KK. Nucleate pool boiling on structured enhanced tubes having pores with connecting gaps, Int. J. Heat Mass Transf. 2001;44 (1):17-28.
[16] Nakayama W, Daikoku T, Kuwahara H, Nakajima T. Dynamic model of enhanced boiling heat transfer on porous surfaces. I. Experimental investigation, Trans. ASME J. Heat Transf. 1980;102 (3):445-450.