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Evaporating heat transfer and pressure drop of hydrocarbon refrigerants in 9.52 and 12.70 mm smooth tube

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Abstract

Experimental results of heat transfer characteristic and pressure gradients of hydrocarbon refrigerants R-290, R-600a, R-1270 and HCFC refrigerant R-22 during evaporating inside horizontal double pipe heat exchangers are presented. The test sections have one tube diameter of 12.70 mm with 0.86 mm wall thickness, another tube diameter of 9.52 mm with 0.76 mm wall thickness was used for this study. The local evaporating heat transfer coefficients of hydrocarbon refrigerants were higher than those of R-22. The average evaporating heat transfer coefficient increased as the mass flux increased. It is showed the higher values in hydrocarbon refrigerants than R-22. Comparing the heat transfer coefficient of experimental results with that of other correlations, the obtained results from the experiments had coincided with most of the Kandlikar's correlation. Hydrocarbon refrigerants have higher pressure drop than R-22 in 12.7 mm and 9.52 mm. This results form the study can be used in the case of designing heat transfer exchangers using hydrocarbons as the refrigerant for the air-conditioning systems.

Keywords: Hydrocarbon refrigerant; Evaporation; Heat transfer coefficient; Pressure drop; Natural refrigerant

1. Introduction

Due to the environmental problems by CFCs and HCFCs, the development of new alternative refrigerants with the high efficient machine which can reduce energy consumption has been becoming an urgent issue [1,2]. HFCs or non-azeotropic refrigerant mixtures [3] has been being regarded as alternative refrigerants. However, HFC's can make acids and toxic substances when they are resolved in a compound into their forming elements by sunlight [4], and though, they have zero ODP (ozone depletion potential), but they have high GWP (global warming potential). Besides of that fact, it is hard to treat non-azeotropic refrigerant mixtures efficiently and is difficult to reproduce the primary constant composition due to its variation caused by leakage for repairing. So, new alternative refrigerants having no

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Nomenclature

BO	boiling number, q/Gi_{lg}
C_p	specific heat at constant pressure (kJ/kg K)
CO	convection number, $((1-x)/x)^{0.8}(\rho_v/\rho_l)^{0.5}$
d	diameter (m)
F_{fl}	fluid dependent parameter
G	mass velocity (kg/m^{2}s)
h	heat transfer coefficient (kW/m^{2}K)
i	enthalpy (kJ/kg)
i_{lg}	latent heat of vaporization (J/kg)
k	thermal conductivity (kW/m K)
m	mass flow rate (kg/h)
n	number of local tube
q	heat flux (kW/m^2)
Q	heat capacity (kW)
Re	Reynold number, $\rho uD/\mu$
S	suppression factor
T	temperature (K)
x	quality

Greek symbols

Δ	difference
μ	dynamic viscosity (Pa s)
ρ	density (kg/m^3)

Subscripts

avg	average
CBD	convective boiling dominant
e	evaporator
eq	equivalent
i	inner
in	inlet
l	liquid
loc	local
NBD	nucleate boiling dominant
o	outer
out	outlet
r	refrigerant
tp	two phase
v	vapor
w	source water

poisonous characteristics, no flammability and should be similar to conventional refrigerant in terms of thermodynamic property are required.

Under these circumstances, additional and active studies regarding the so-called ''natural refrigerants'' have been under way. Especially HC's refrigerants are examined positively as an alternative refrigerant for (H)CFC because it is easily available and its GWP and ODP are almost close to zero. But, the developed countries like US and Japan have not adapted them except for Europe due to flammability of HC's. However, according to James [5], in case of the household refrigerators, the possibility of explosion by flammability can be negligible since the HC's charge quantity is about half of general CFC refrigerant's one. Besides, if some simple safety device (e.g. ventilation system or leakage detector) is installed, it can overcome that problem in the large size air-conditioning and refrigerating system. But, the researches for performance of the refrigeration and air-conditioning systems using the HC's as a refrigerant are not enough, especially, the study on characteristics of evaporating heat transfer is the one of those.

Kandlikar [6] introduced a general correlation about fluid boiling in the vertical·horizontal tube. Kwon [7] experimented regarding the characteristics of evaporating heat transfer using R-290, R-410A and compared with those of R-22. According to his report, evaporating heat transfer coefficient of R-290 was higher than that of R-22 or R-410A, but the research on evaporating heat transfer of natural refrigerants is still ridiculously rare.

In this scenarios, the purpose of this paper is to obtain basic data for the purpose of designing the evaporator that uses HC's refrigerants and is to compare experimentally, the evaporating heat transfer characteristic and the pressure drop of R-1270 (propylene), R-290 (propane), R-600a (iso-butane) taking R-22 as base at the smooth tube.

2. Experimental apparatus and method

2.1. Experimental apparatus

Fig. 1 shows the schematic of the experimental apparatus including basic air-conditioning and refrigerating system consisted of compressor, condenser, expansion valve, evaporator and peripheral device. The system also consists of two main flow loops: a refrigerant loop and heat source water for evaporating or condensing loop. In the test section of the experiment, the evaporator is a double-tube type heat exchanger divided into three sections, which are inner tube, outer tube and annular section.

The heat exchanger (test section) is shown in Fig. 2. The inner diameter of the inner tube (copper) is 10.92 mm, 8 mm, and outer and inner diameters of the outer tube (copper) are 19.94 and 22.22 mm respectively.

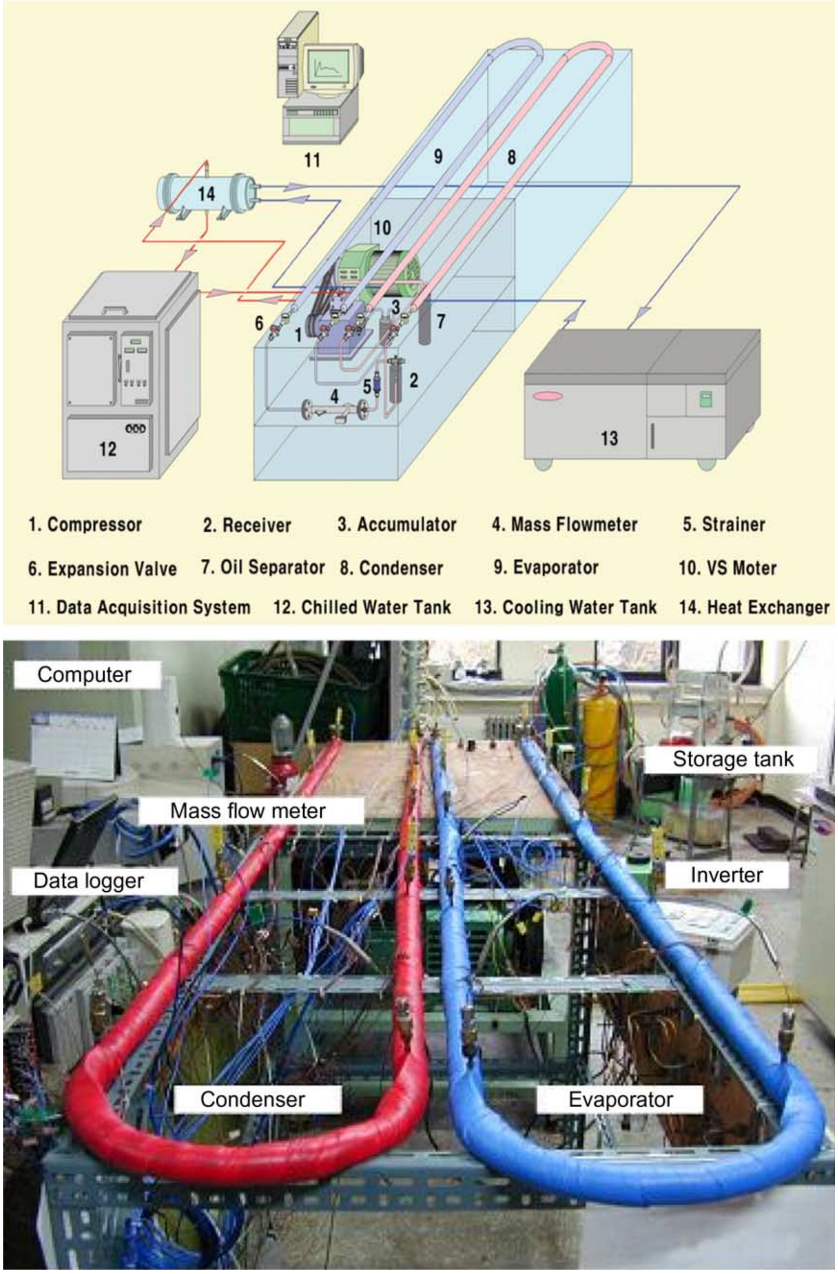

Fig. 1. Schematic diagram of experimental apparatus.

The heat exchanger is divided into eight small subsections equally, each has 675 mm length, and the shape of a refrigerant tube through the U-bend is double-tube type with identical bending used to avoid a detour. As seen from Fig. 2, inside the double-tube heat exchanger, water flows countercurrently in the test section annulus, while refrigerant is evaporated inside the test tube.

Fig. 3 shows that the temperatures of the refrigerant, cooling water and inner wall of heat exchanger are measured in the heat exchanger as stated above. Each of these subsections are instrumented with four insulated type T thermocouples of 0.3 mm diameter, one at the top, two at the both sides and one at the bottom. The pressure gages installed at the inlet and outlet of the heat exchanger can measure the pressure drop of the refrigerant in the inner tube.

The test conditions are summarized in Table 1.

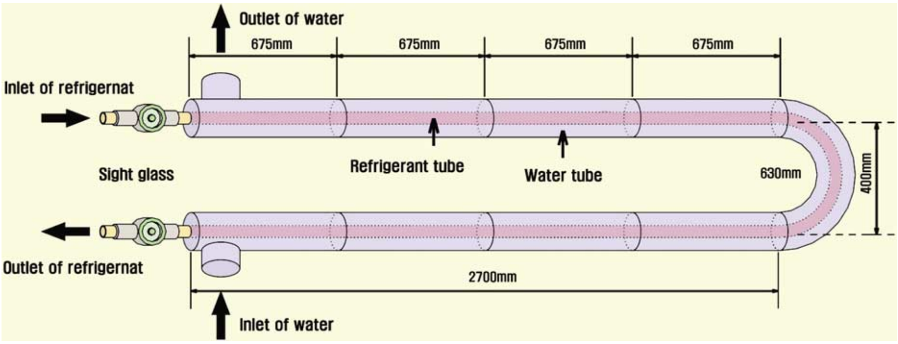

Fig. 2. Test section of the evaporator.

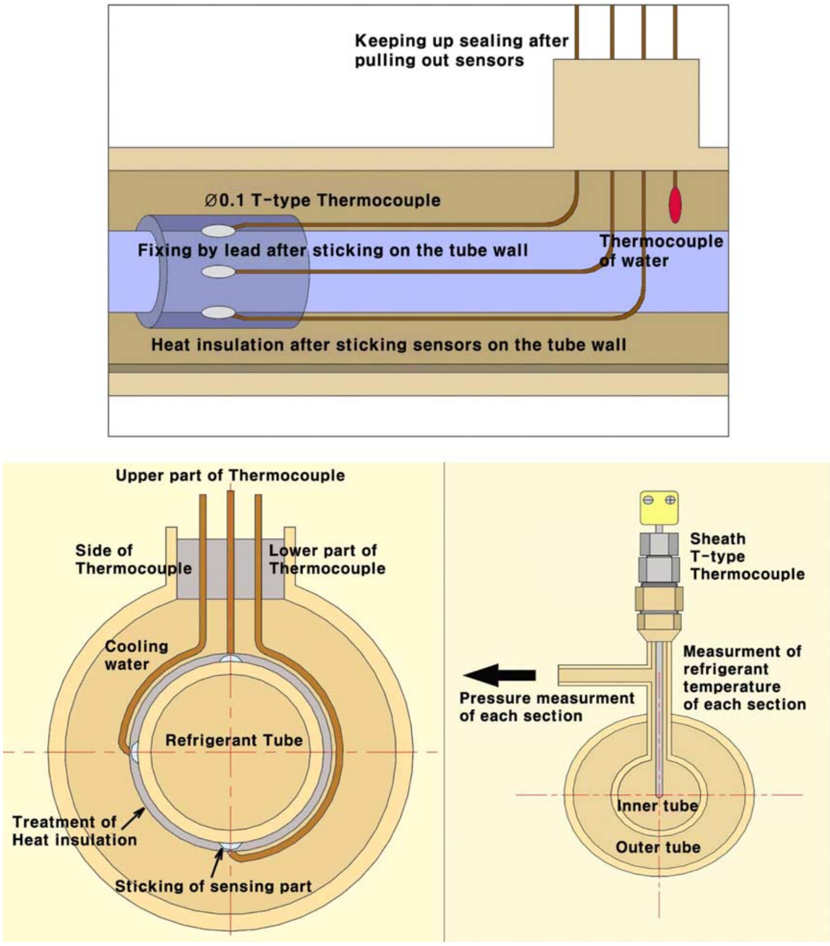

Fig. 3. Setting of temperature sensor.

2.2. Experimental method

In this paper, we used R-22 (restricted refrigeration), R-290 (propane, purity 99.5%), R-600a (iso-butane, purity 99.5%) and R-1270 (propylene, purity 99.5%) as working fluids. To examine the evaporating heat transfer characteristics, the data (temperature of refrigerant,heat source water and outer wall) are measured at

Table 1
Experimental conditions

Parameters	Range
Refrigerant	
Working fluid	R-22, R-1270, R-290, R-600a
Evaporating temperature (K)	263–283
Inner tube diameter (mm)	12.70, 9.52
Mass flux (kg/m^{2}s)	50–200
Chilled water	
Inlet temperature (K)	287
Mass flow rate (kg/h)	240–480

the heat exchanger. In addition, flow rate of refrigerant and heat source, the pressure between inlet and outlet of heat exchanger are measured as well.All the temperatures are measured by T-type thermocouple that has ±0.1% error range, and we used Bourdon-type pressure gauges installed 12 pieces throughout all sections for checking the pressure. The accurate mass flow meter is installed at the outlet of condenser, and an orifice flow-meter is set to measure heat source water flow rate at the inlet of evaporator and condenser, respectively. The flow meter is accurate within 1% of full scale. The experiment was performed on steady state after conditions control, and repeated with two given conditions changing flow rate and temperature.

2.3. Data reduction

Defect signals for checking data are processed with the computer through the data logger. The thermo-physical properties of R-22 and R-1270, R-290, R-600a (alternative refrigerants) are calculated by REFPROP(version 6.0) a thermo-physical property calculation program developed by NIST (National Institute of Standards and Technology). We can use the following equations to analyze the test data, using the above mentioned properties.

The amount of heat exchange at the evaporator is given as:

$$Q_{\mathrm{ew}} = m_{\mathrm{ew}} \cdot c_{p,\mathrm{ew}} \int_{T_{\mathrm{c,in}}}^{T_{\mathrm{c,out}}} \mathrm{d}t \tag{1}$$

$$Q_{\mathrm{er}} = m_{\mathrm{er}} \cdot (i_{\mathrm{e,in}} - i_{\mathrm{e,out}}) \tag{2}$$

where Q_{ew} is the heat amount from water to refrigerant and Q_{er} is the heat amount from refrigerant to water. m_{ew} and m_{er} are the heat source water mass flow rate [kg/h] and the refrigerant mass flow rate [kg/h] separately. $T_{\mathrm{e,in}}$ and $T_{\mathrm{e,out}}$ are the temperature [K] of heat source water at the inlet and outlet on the evaporator. $i_{\mathrm{e,in}}$ and $i_{\mathrm{e,out}}$ are the enthalpy differences [kJ/kg] between inlet and outlet, $c_{p,\mathrm{ew}}$ the specific heat [kJ/kg K] of chilled water respectively.

In case of the evaporating process, we needed to calculate a heat transfer coefficient toward circumferential direction of the tube, since it has many influences on the system, and is defined as follows:

$$h_{\mathrm{e,loc}} = \frac{q_{\mathrm{e}}}{T_{\mathrm{e,wi}} - T_{\mathrm{er}}} \tag{3}$$

where $h_{\mathrm{e,loc}}$ is the local heat transfer coefficient[kW/m^{2}K] at the subsection of the evaporator and q_{e} is heat flux [kW/m^2] shown in Eq. (4). T_{er} and $T_{\mathrm{e,wi}}$ are refrigerant temperature [K] and inner wall temperature at the inner tube. $T_{\mathrm{e,wi}}$ is given in Eq. (5).

$$q = \frac{Q_{\mathrm{ew}}}{\pi \cdot d_{\mathrm{i}} \cdot \Delta z} \tag{4}$$

$$T_{\mathrm{e,wi}} = T_w - \frac{Q_{\mathrm{e,sub}} \cdot \ln \frac{d_{\mathrm{o}}}{d_{\mathrm{i}}}}{2\pi \cdot k_{\mathrm{w}} \cdot \Delta z} \tag{5}$$

where Q_{ew} is the heat amount [kW] calculated by Eq. (1), d_{i} and Δz are the inner diameter[m] of inner tube and the length[m] of subsection. T_{w} is the average temperature [K] measured from one at the top, two at the side and one at the bottom, at outer wall of inner tube given Eq. (6). $Q_{\mathrm{e,sub}}$ measured by the experiment is the exchange heat amount [kW] at the subsection of the evaporator. d_{o} and k_{w} are the outer diameter[m] of inner tube and the thermal conductivity [kW/m K] of copper tube separately.

$$T_{\mathrm{w}} = \frac{T_{\mathrm{w,top}} + 2T_{\mathrm{w,side}} + T_{\mathrm{w,bottom}}}{4} \tag{6}$$

where $T_{\mathrm{w,top}}$, $T_{\mathrm{w,side}}$ and $T_{\mathrm{w,bottom}}$ are the temperature [K] measured at the top, side and bottom, respectively.

To express the average evaporating heat transfer coefficient $h_{\mathrm{e,avg}}$ [kW/m^{2}K], we also could write:

$$h_{\mathrm{c,avg}} = \frac{1}{x_{\mathrm{in}} - x_{\mathrm{out}}} \int_{x_{\mathrm{out}}}^{x_{\mathrm{in}}} h_{\mathrm{c,loc}}\, \mathrm{d}x = \sum \frac{h_{\mathrm{c,loc}}}{n} \tag{7}$$

where x_{in} and x_{out} are the quality at the inlet and outlet of the evaporator subsection, $h_{\mathrm{e,loc}}$ is the local heat transfer coefficient [kW/m^{2}K] calculated by Eq. (3). n is the number of the subsection. The refrigerant quality x is given to Eq. (8) as following the quality $x_{\mathrm{e,out}}$ at the outlet of the evaporator subsection is also given to Eq. (9).

$$x = \frac{\Delta i_{\mathrm{sub}}}{i_{\mathrm{fg}}} \tag{8}$$

$$x_{\mathrm{e,out}} = x_{\mathrm{in}} + \frac{\pi \cdot d_{\mathrm{i}}}{m_{\mathrm{er}} \cdot i_{\mathrm{fg}}} \cdot \int_{z_{\mathrm{in}}}^{z_{\mathrm{out}}} q_{\mathrm{e}}\, \mathrm{d}z \tag{9}$$

where Δi_{sub} is the enthalpy difference between inlet and outlet of the subsection, i_{fg} is the latent heat of refrigerant. z_{in} and z_{out} are the inlet and outlet of a section, respectively. q_{e} is the heat flux [kW/m^2] at the evaporator and is calculated by Eq. (4), $x \cdot \int_{z_{\mathrm{in}}}^{z_{\mathrm{out}}} q_{\mathrm{e}}\,\mathrm{d}z$ is the

Table 2
Parameters and estimated uncertainties

Parameter	Uncertainty
Measured quantities	
Temperature (°C)	±0.1 °C
Pressure (MPa)	±0.002 MPa
Pressure drop (kPa)	±0.2 kPa
Water flow rate (kg/s)	±1%
Refrigerant flow rate (kg/s)	±1%
Calculated quantities	
Mass velocity (kg/m^{2}s)	±1.03%
Vapor quality	±7.26%
Heat flux (kW/m^2)	±7.19%
Heat transfer coefficient (kW/m^{2}K)	±8.76%

cumulative total heat mount at the subsection from the heat exchange inlet.

The uncertainties of the measured and calculated parameters are estimated by following the procedures described by Moffat [8] and Holman [9]. The experimental uncertainties associated with the measurement devices and sensors. The method is based on a combining of all the uncertainties primary experimental measurements. The results are tabulated in Table 2.

3. Results and discussion

3.1. Evaporating heat transfer

To scrutinize the reliability of the experimental setup, we examined the heat balance between refrigerant and heat source water in the evaporator and the result is shown in Fig. 4. Fig. 4 reveals that the heat capacity Q_w calculated by Eq. (1) is given in X-direction and the heat capacity Q_r calculated by Eq. (2) is in a Y-direction.

In case of HC's refrigerants, the range of error is produced almost equal values of around ±20% regardless of refrigerant types and tube diameter used in the experiment.

Fig. 5 shows the local evaporating heat transfer coefficient with respect to the change of quality on refrigerant variation. It is increasing continuously with refrigerant quality. Besides, it decreases rapidly for the identical mass-flux, but over 0.85 quality. This means that the increased gaseous refrigerant comes out with the completion of evaporation of liquidized refrigeration over 0.85 quality and causes the drop of heat transfer. It is reported that the local heat transfer rate of HC's refrigerants is almost identical with that of R-22 in a qualitative tendency, but is 13.35% higher in an average than that with a diameter of 12.70 mm and is 13.73% also higher for 9.52 mm outer diameter in a quantitative difference.

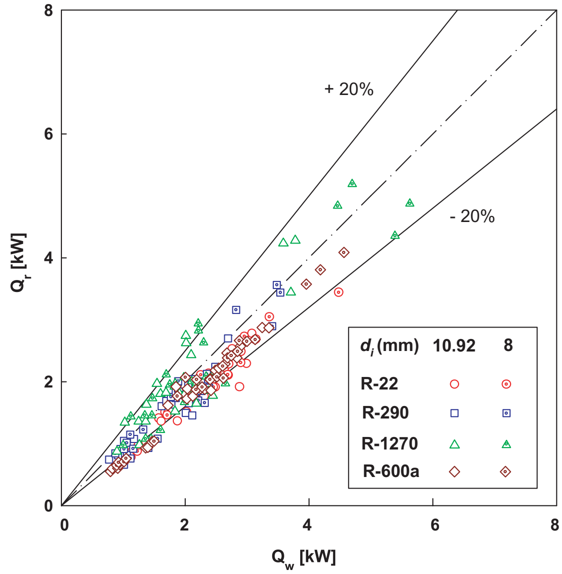

Fig. 4. Heat balance in the evaporator.

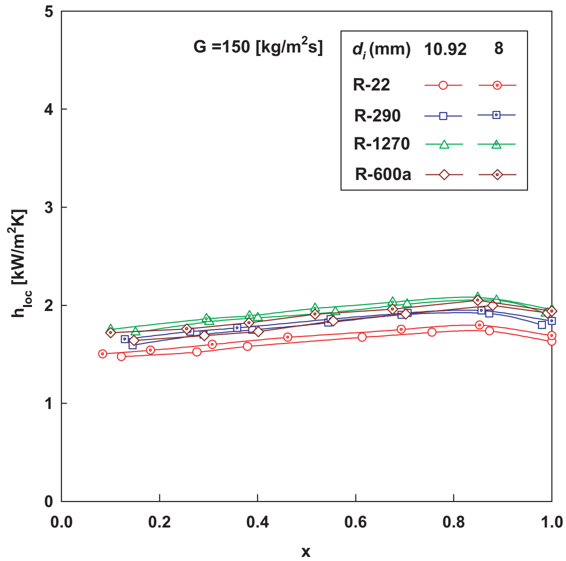

Fig. 5. Local evaporating heat transfer coefficients.

The average evaporating heat transfer coefficient is shown in Fig. 6 with respect to a refrigerant mass flux. It increases as the mass flux increases irrespective of a refrigerant variation. If we observe the data in terms of refrigerant classification, the average evaporating heat transfer coefficients of HC's refrigerants is higher than those of CFCs, and appeared in the order of R-1270, R-600a, R-290 with respect to the approaching of the high-mass flow velocity. Turbulence happens

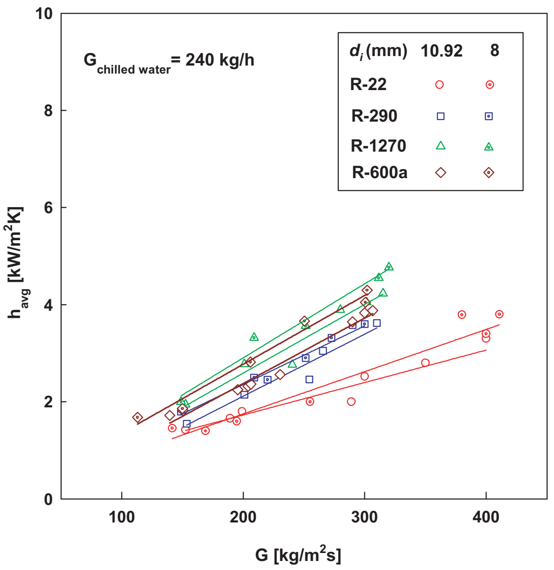

Fig. 6. Average evaporating heat transfer coefficients.

more often for 9.52 mm outer diameter than 12.70 mm that's why the evaporating heat transfer coefficient is showing higher value for 9.52 mm. In comparison with R-22, the average evaporating heat transfer coefficient for R-290 is approximately 18.98% higher, R-600a is 18.27% higher and R-1270 is 32.38% higher, respectively for 12.70 mm inner tube. In case of 9.52 mm inner tube, R-290 is approximately 19.96% higher, R-600a is 18.57% higher and R-1270 is 34.23% higher, respectively.

3.2. Comparison with other correlations

A Comparison with other correlations is imperative to predict the heat transfer coefficient. In the design of the heat exchanger, the non-dimensional heat transfer correlations are used crucial factors to determine the size or shape of the heat exchanger, and the representative correlations such as Shah [10], Gungor-Winterton [11] and Kandlikar were used for this purpose.

Here, the Kandlikar correlation is given to Eqs. (10)–(12).

$$h_{\mathrm{tp}} = \text{the larger of } h_{\mathrm{NBD}} \quad \text{and} \quad h_{\mathrm{CBD}} \tag{10}$$

where

$$h_{\mathrm{NBD}} = h_l[0.6683\mathrm{CO}^{-0.2} + 1058\mathrm{BO}^{0.7}F_{\mathrm{fl}}] \tag{11}$$

and

$$h_{\mathrm{CBD}} = h_l[1.1360\mathrm{CO}^{-0.9} + 667.2\mathrm{BO}^{0.7}F_{\mathrm{fl}}] \tag{12}$$

where the Dittus–Boelter correlation [12] is used to calculate the heat transfer coefficient h_1 for the single-phase liquid only. The variable F_{fl} is a fluid-dependent parameter and is 2.20 for R-22. The magnitude of F_{fl} for hydrocarbon refrigerants is not available in the literature. Hence, we used Froster–Zuber correlation [13] to find out pool boiling data of experimental fluid with suppression factor S, defined in Eq. (13). This is the reflection of the fact that as forced-convection effect grows and thickness of thermal boundary layer decreases, distribution of nucleate boiling is greatly restricted. This concept, at first, was proposed by Chen [14], and was developed by Collier and Thome [15] for practical use later on.

$$S = \frac{1}{1 + 2.56 \times 10^{-6} Re_{\mathrm{eq}}^{1.17}} \tag{13}$$

where Re_{eq} is equilibrium Reynolds number that can be calculated through Eq. (14).

$$Re_{\mathrm{eq}} = G_{\mathrm{eq}} \cdot d_{\mathrm{in}}/\mu_l \tag{14}$$

where G_{eq} is equivalent mass velocity that can be calculated through Eq. (15).

$$G_{\mathrm{eq}} = G \cdot [(1-x) + x(\rho_{\mathrm{l}}/\rho_{\mathrm{v}})^{1/2}] \tag{15}$$

Figs. 7–9 show the comparison with the evaporating heat transfer coefficient and the results. The results are well matched with above mentioned correlations regardless of a type of refrigerants and tube diameter around 20% error range.

3.3. Pressure drop

In Fig. 10, the average pressure drop of R-22, R-290, R-600a and R-1270 is compared with respect to the

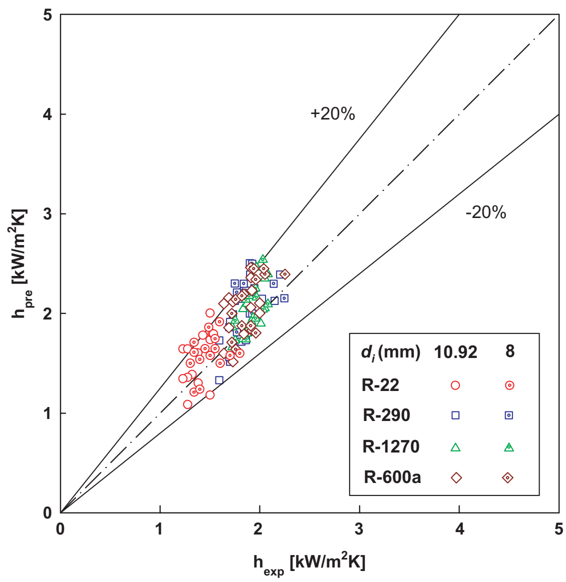

Fig. 7. Comparison of heat transfer coefficient with Shah's correlation.

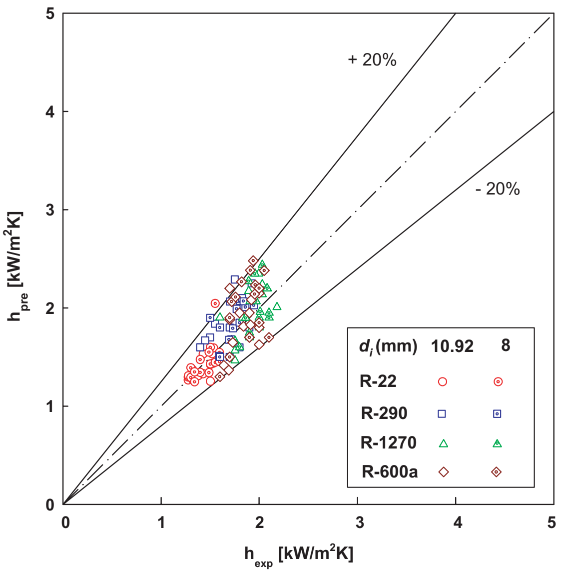

Fig. 8. Comparison of heat transfer coefficient with Kandlikar's correlation.

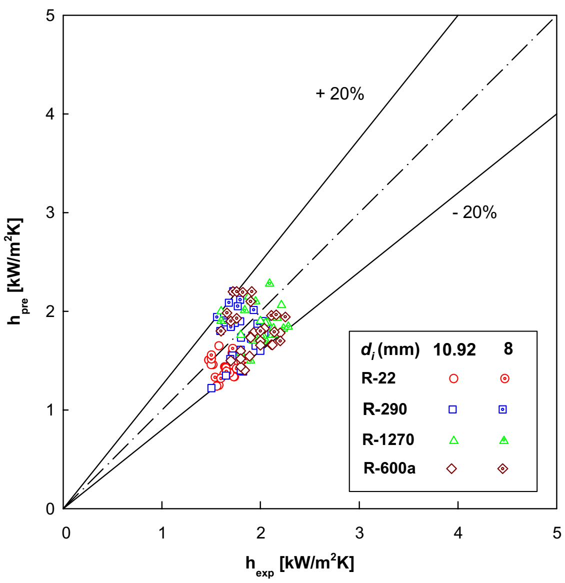

Fig. 9. Comparison of heat transfer coefficient with Gungor–Winterton's correlation.

quality for 150 [kg/m^{2}s] mass flux. The highest value of pressure drop is shown at 0.6 quality point in which the bent pipe section (at the evaporator) is located. After 0.8 quality point, pressure drop decreased gradually and it seemed that the friction loss caused by thinned liquid film is diminished at the annular flow section. In comparison with R-22, the average pressure drop of HC's refrigerants is approximately 67.7% higher.

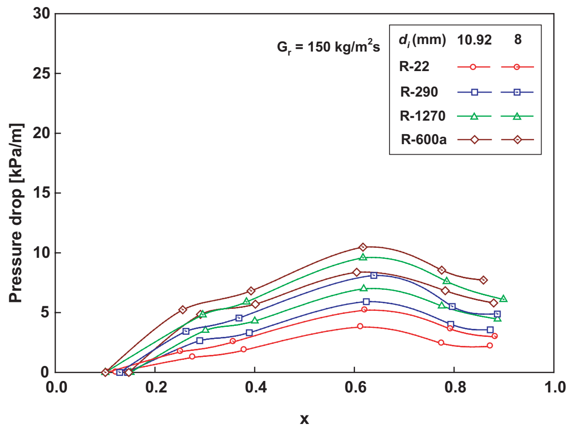

Fig. 10. Pressure drop vs. quality.

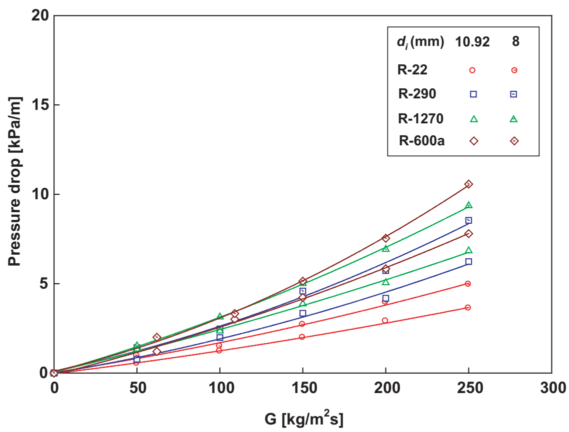

Fig. 11. Average pressure drop vs. mass flux.

Fig. 11 shows the change of pressure drop for 50–250 [kg/m^{2}s] mass flux and, in comparison with R-22, the average pressure drop of HC's refrigerants is approximately 47.18% and 45.42% higher for 12.70 mm and 9.52 mm outer diameter respectively.

4. Conclusions

In connection with the above results, we have projected the following conclusions for natural refrigerant on HC's that is expected to be the alternative refrigerant of R-22 with environment friendly vision.

The local evaporating heat transfer coefficient of HC's is higher than that of conventional R-22. R-1270 showed the highest average evaporating heat transfer coefficient among all the HC's refrigerants.

In comparison to R-22, HC's refrigerants have similar or better ability and are also environmentally friendly other than flammability. Hence, we can claim that they can be used as the new alternative refrigerants (naturally) of R-22 in the future.

The correlation of Kandlikar highly seemed to conform to the obtained experimental results.

It turned out that the pressure drop of HC's refrigerants is greater than that of R-22 through our experiment. Accordingly, we need further study to reduce loss caused by pressure drop and to get more accurate results.

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