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Condensing heat transfer and pressure drop characteristics of hydrocarbon refrigerants

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Abstract

This paper presents the experimental results of condensing heat transfer coefficients and pressure gradients of HC refrigerants (e.g. R-1270, R-290 and R-600a) and R-22 in horizontal double pipe heat exchangers, having two different internal diameters of 12.70 mm and 9.52 mm (OD), respectively. Both the local condensing heat transfer coefficients and pressure drops (inside the tube) of hydrocarbon refrigerants were higher than R-22. The average condensing heat transfer coefficient increased with the mass flux. The experimental heat transfer coefficients agreed with the correlations of Shah, Travis and Cavallini–Zecchin's to within ±20%. These results can be useful in the design of new heat exchangers involving hydrocarbon refrigerants for future air-conditioning systems.

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

1. Introduction

Traditional refrigerants CFCs and HCFCs are being gradually replaced by HFC refrigerants internationally. Although HFCs have zero ozone depletion potential (ODP), those suffer from high global warming potential (GWP), and hence they are not particularly attractive from the environmental view point.

New alternative refrigerants should not only have low ODP but should also have low GWP, be safe, be reliable, be less flammable, and be economical for being used in the existing facilities [1,2]. Under these circumstances, hydrocarbon refrigerants (HC's—e.g. propylene, propane, iso-butane, etc.) are being examined vigorously as alternative refrigerants due to their low cost, ease of availability and better mixing properties with general lubricants. But the developed countries like US have not yet adopted them due to their flammability except Europe [3]. According to James and Missenden [4], in the case of household refrigerators, the hydrocarbon refrigerant charge is so small (about half of the general CFC) that the possibility of explosion due to flammability is practically negligible.

However, before these refrigerants can be accepted by the refrigeration industry internationally, fundamental heat and mass transfer characteristics of these refrigerants need to be investigated for the optimal design of the heat exchangers and thereby, the refrigeration systems. Recently, the authors had performed an extensive study on the evaporation heat transfer and pressure drop characteristics of hydrocarbon refrigerants inside smooth tubes and published the results in a sister paper [5]. However, as a part of the global project, this paper presents the physics of condensation heat transfer and corresponding pressure drops of hydrocarbon refrigerants in smooth

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Nomenclature

C_p	specific heat at constant pressure (kJ/kg K)	*Subscripts*	
d	diameter (m)	avg	average
G	mass flux (kg/m^2 s)	c	condenser
h	heat transfer coefficient (kW/m^2 K)	i	inner
i	enthalpy (kJ/kg)	in	inlet
i_{fg}	latent heat of vaporization (kJ/kg)	loc	local
k	thermal conductivity (kW/m K)	o	outer
$\dot{m}$	mass flow rate (kg/h)	out	outlet
n	number of local sub-sections of the tube	r	refrigerant
$\dot{q}$	heat flux (kW/m^2)	sub	condenser sub-section
$\dot{Q}$	heat transfer rate (kW)	w	source water
T	temperature (K)	wi	inside tube wall
x	quality		
Greek symbols			
Δ	difference		
Δz	sub-section length (m)		

tubes. Although there are some studies available in the open literature [6–8] that deal with the fundamental aspects of numerical heat transfer, the information on the condensation heat transfer dealing with hydrocarbon refrigerants is practically absent. Therefore, the current study fills in this void by presenting the condensing heat transfer characteristics of hydrocarbon refrigerants. In doing so, the paper presents the fundamental experimental heat transfer data on the condensation heat transfer and pressure drop of hydrocarbon refrigerants, namely R-1270 (propylene, 99.5% purity), R-290 (propane, 99.5% purity), R-600a (iso-butane, 99.5% purity); and compares it against R-22 in smooth tubes.

2. Experimental apparatus and method

2.1. Experimental apparatus

Fig. 1 shows the schematics of the experimental apparatus including basic air-conditioning and refrigerating system consisting of a compressor, a condenser, an expansion valve, an evaporator and a peripheral device such as an oil separator, a receiver, an accumulator and so on. The system also consists of two main flow loops: a refrigerant loop and a secondary heat source water circuit involving either evaporation or condensation loop. In the test section of the experiment, the condenser is a double-tube type heat exchanger divided into two sections, which are inner tube and annular region. The inner tube and the annular section are applied respectively to the refrigerant and the secondary fluid flow.

The heat exchanger (test section) is shown in Fig. 2. The outer diameter of the inner copper tubes is respectively 12.7 mm (ID: 10.92 mm) and 9.52 mm (ID: 8 mm), outer diameter of the outer copper tube is 22.22 mm. The heat exchanger is equally divided into 8 small sub-sections of 675 mm length. The shape of the refrigerant tube through the U-bend is double-tube type with identical bending to avoid a detour. As shown in Fig. 2, water flows countercurrently in the annulus test section of the double-tube heat exchanger, while refrigerant is condensed inside the test tube.

Fig. 3 shows the temperature measurement locations of the refrigerant, cooling water and inner wall of heat exchanger. Each of these sub-sections are instrumented with four insulated type T thermocouples of 0.3 mm diameter, one at the top, two at the two sides and one at the bottom. Two pressure gauges were installed at the inlet and outlet of the heat exchanger to measure the refrigerant pressure drop in the inner tube.

The test conditions are summarized in Table 1. Here, R-600a has a small mass flux range due to the larger specific volume than other refrigerants.

2.2. Experimental method

In this paper, four refrigerants, namely R-290 (propane, purity 99.5%), R-600a (iso-butane, purity 99.5%) and R-1270 (propylene, purity 99.5%), were investigated to evaluate their condensing heat transfer characteristics against R-22. The data (temperature of refrigerant, heat source water and outer wall) was measured along the length of the heat exchanger. In addition, flow rates of both the refrigerant and the heat source, and the pressures at the inlet and outlet of the heat exchanger were measured. All temperatures were measured with T-type thermocouple having measurement uncertainty of ±0.1 (°C). Bourdon-type pressure gauges were used to measure pressures. Micromotion mass flow-meter measured the refrigerant mass flow rates to within ±1% at condenser

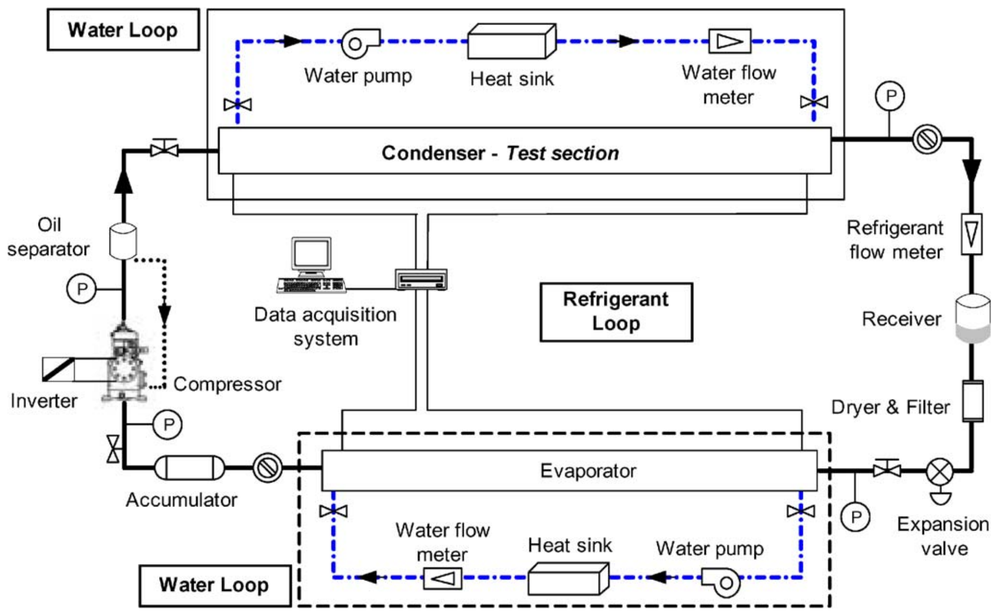

Fig. 1. Schematic diagram of experimental apparatus.

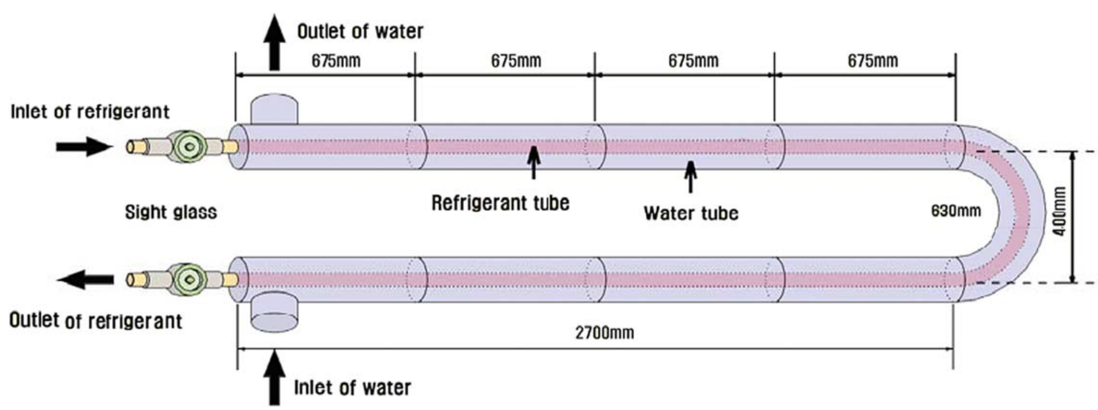

Fig. 2. Test section of the condenser.

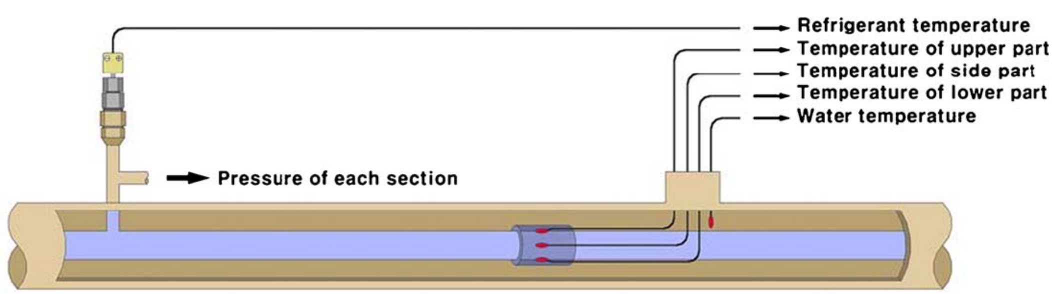

Fig. 3. Temperature and pressure measurement sensors in the test set-up.

outlet. An orifice flow-meter measured the heat source water flow rate to within ±1% at the inlets of evaporator and condenser. The experiments were performed in steady state flow conditions, and were repeated for varying conditions of flow rate and temperature. Detective signals for checking the data were processed through a computer controlled data logger. The thermo-physical properties of R-22 and R-1270, R-290, R-600a (alternative refrigerants) were calculated using REFPROP (version 6.0) a thermo-physical property calculation program developed by NIST (National Institute of Standards and Technology).

Table 1
Experimental test conditions

Parameters	Range
Refrigerant	
Working fluid	R-22, R-1270, R-290, R-600a
Condensing temperature (K)	308–318
Inner tube diameter (OD, mm)	12.70, 9.52
Mass flux (kg/m^2 s)	R-600a: 62–150
	The others: 150–300
Cooling water	
Inlet temperature (K)	305
Mass flow rate (kg/h)	700

2.3. Formulation of heat transfer analysis

The amount of heat exchanged in the condenser can be given by

$$\dot{Q}_{\mathrm{cw}} = \dot{m}_{\mathrm{cw}} \cdot c_{p,\mathrm{cw}} \cdot (T_{\mathrm{c,out}} - T_{\mathrm{c,in}}) \tag{1}$$

$$\dot{Q}_{\mathrm{cr}} = \dot{m}_{\mathrm{cr}} \cdot (i_{\mathrm{c,in}} - i_{\mathrm{c,out}}) \tag{2}$$

The local condensing heat transfer coefficient toward circumferential direction of sub-section of the tube can be defined as

$$h_{\mathrm{c,loc}} = \frac{\dot{q}_{\mathrm{c}}}{T_{\mathrm{c,wi}} - T_{\mathrm{cr}}} \tag{3}$$

where $T_{\mathrm{c,wi}}$ is the inner wall temperature of the inner tube and the heat flux, $\dot{q}_{\mathrm{c}}$ can be given by

$$\dot{q}_{\mathrm{c}} = \frac{\dot{Q}_{\mathrm{cr}}}{\pi \cdot d_{\mathrm{i}} \cdot \Delta z} \tag{4}$$

$$T_{\mathrm{c,wi}} = T_{\mathrm{w}} - \frac{\dot{Q}_{\mathrm{cr,sub}} \cdot \ln \frac{d_{\mathrm{o}}}{d_{\mathrm{i}}}}{2\pi \cdot k_{\mathrm{w}} \cdot \Delta z} \tag{5}$$

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

where $T_{\mathrm{w,top}}$, $T_{\mathrm{w,side}}$ and $T_{\mathrm{w,bottom}}$ are the tube wall temperatures, measured respectively at the top, side and bottom of the tube. The average condensing heat transfer coefficient $h_{\mathrm{c,avg}}$ can be expressed as

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

where the refrigerant quality x is given by

$$x_{\mathrm{c,out}} = x_{\mathrm{in}} - \frac{\dot{Q}_{\mathrm{cr,sub}}}{m_{\mathrm{cr}} \cdot i_{\mathrm{fg}}} \tag{8}$$

where i_{fg} is the latent heat of refrigerant and $\dot{Q}_{\mathrm{cr,sub}}$ is the heat transfer rate (kW) at one sub-section of the condenser.

The uncertainties of the measured and calculated parameters are estimated following Moffat [9] and Holman [10] and are given in Table 2.

Table 2
Parameters and estimated uncertainties

Parameter	Uncertainty
Measured quantities	
Temperature (°C)	±0.1 °C
Pressure (kPa)	±2 kPa
Pressure drop (kPa)	±0.2 kPa
Water flow rate (kg/s)	±1%
Refrigerant flow rate (kg/s)	±1%
Calculated quantities	
Mass flux (kg/m^2 s)	±1%
Vapor quality	±7.3%
Heat flux (kW/m^2)	±7.2%
Heat transfer coefficient (kW/m^2 K)	±8.8%

3. Results and discussion

3.1. Condensing heat transfer

To scrutinize the reliability of the experimental set-up, the heat balance between the refrigerant and the heat source water in the condenser was examined and the result is shown in Fig. 4. The heat capacity $\dot{Q}_{\mathrm{cw}}$ (calculated by Eq. (1)) is plotted on the x-axis, while the condenser refrigerant capacity $\dot{Q}_{\mathrm{cr}}$ (calculated by Eq. (2)) on y-axis of Fig. 4. It is evident from the figure that the two values agree with each other to within ±10% for all refrigerants and tube diameters.

Fig. 5 shows the variation of the local condensing heat transfer coefficient against refrigerant quality. As the refrigerant condenses, its quality progressively decreases, increasing the thermal resistance of the liquid component in the two-phase flow and decreasing the heat transfer coefficient. On qualitative basis, the local heat transfer rate of HC's refrigerants is almost identical to R-22, however, on the quantitative basis, it is over 20% higher than R-22 for both the tube diameters.

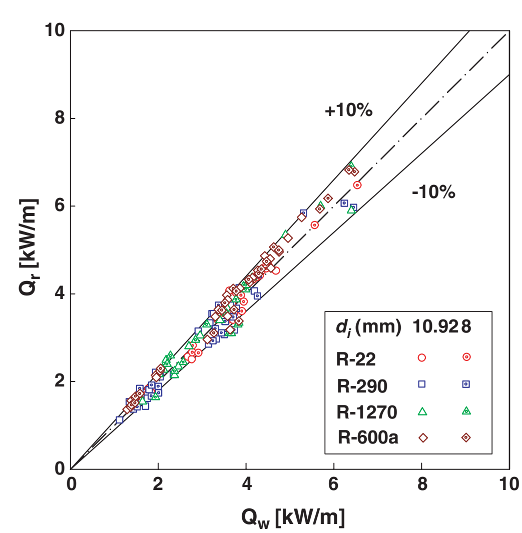

Fig. 4. Heat balance in the condenser.

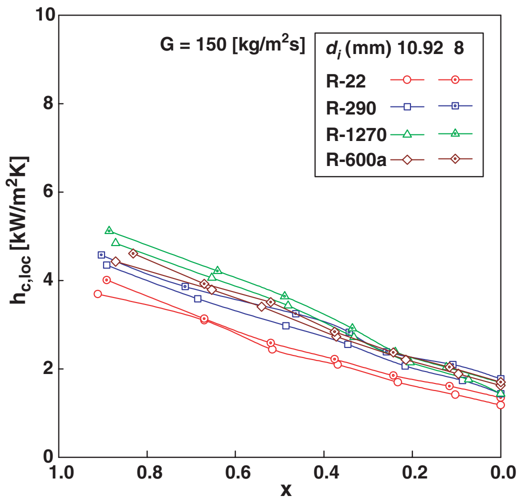

Fig. 5. Local condensing heat transfer coefficients.

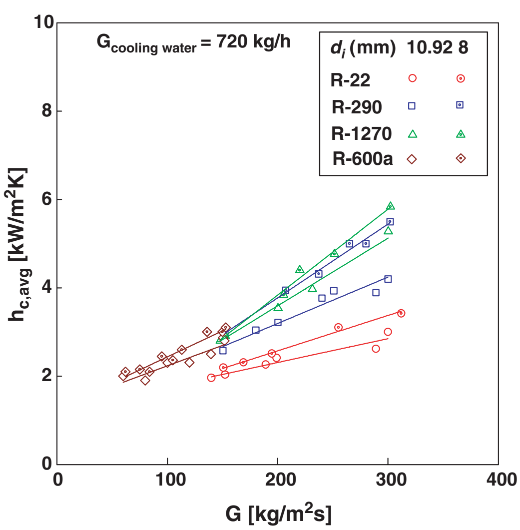

Fig. 6. Variation of average condensing heat transfer coefficients vs. mass flux.

The average condensing heat transfer coefficient is shown in Fig. 6 against refrigerant mass flux. The average condensing heat transfer coefficients of HC's refrigerants are higher (R-1270 performing the best followed by R-600a and R-290) than R-22 at approaching high-mass flow velocity. Turbulence is more pronounced in smaller diameter tube (9.52 mm) than the larger diameter tube (12.70 mm) and hence its condensing heat transfer coefficient is higher. In comparison to R-22, the average condensing heat transfer coefficient for R-290, R-600a and R-1270 is respectively 37.8%, 31.3%, and 36.1% higher for 12.7 mm diameter tube; and 36.3%, 36.2% and 40.9% higher for 9.52 mm diameter tube.

3.2. Comparison with other correlations

In the design of the heat exchanger, the non-dimensional heat transfer correlations such as Shah, Traviss and Cavallini–Zecchin, are crucial to determine the size or shape of the heat exchanger. The experimental condensing heat transfer coefficient has been compared against the Shah, Traviss and Cavallini–Zecchin correlations, respectively in Figs. 7–9, where the agreement is consistently found to be within ±20%.

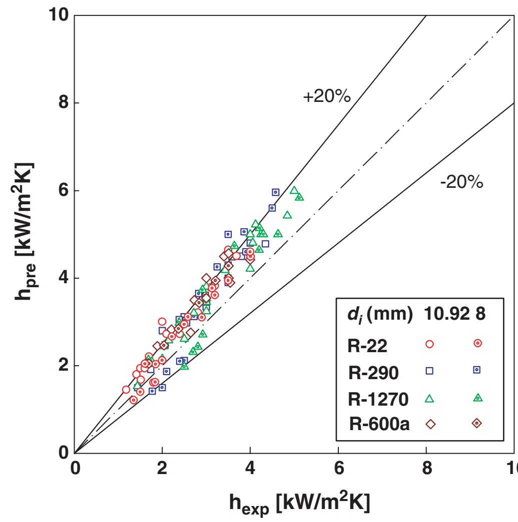

Fig. 7. Comparison of experimental heat transfer coefficient with Shah's correlation.

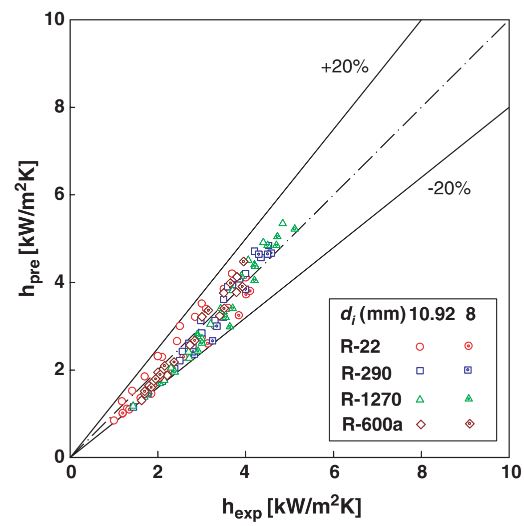

Fig. 8. Comparison of experimental heat transfer coefficient with Traviss's correlation.

3.3. Pressure drop

In Fig. 10, the average pressure drop of R-22, R-290, R-600a and R-1270 is compared with refrigerant quality at mass flux of 150 (kg/m^2 s). The highest pressure drop occurs at a point of refrigerant quality of 0.3 around the bent section of the pipe. The pressure drop of HC's refrigerants is higher than R-22, because of lower vapor density

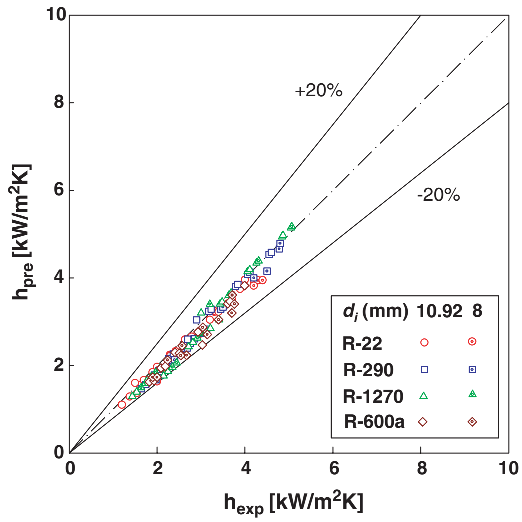

Fig. 9. Comparison of experimental heat transfer coefficient with Cavallini–Zecchin's correlation.

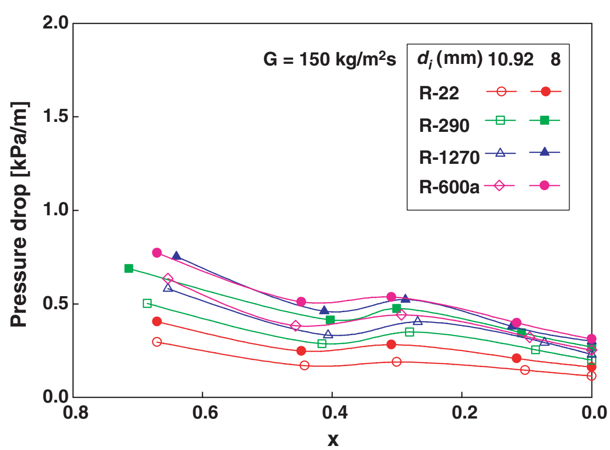

Fig. 10. Pressure drop per unit length vs. refrigerant quality.

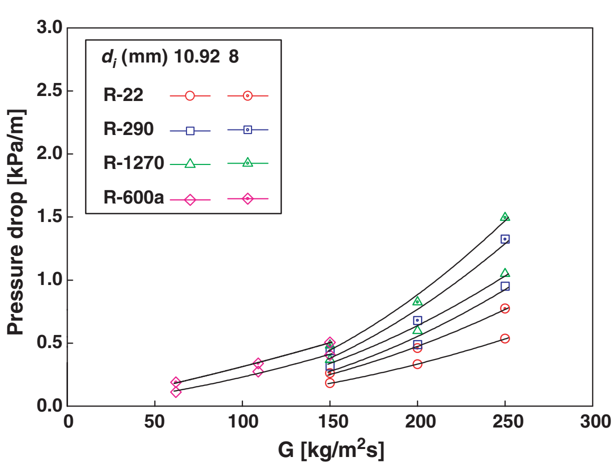

Fig. 11. Average pressure drop per unit length vs. refrigerant mass flux.

of HC refrigerants. The results are similar to those of Wijaya and Spatz [11] and Torikoshi and Ebisu [12].

Fig. 11 shows the pressure drop per unit length against mass flux ranging between 50 and 250 (kg/m^2 s). Compared to R-22, the average pressure drop of HC's refrigerants is approximately 50% and 61.3% higher for 12.7 mm and 9.52 mm outer diameter, respectively.

4. Conclusions

This study reports the condensing heat transfer and pressure drop data for R-22 and HC refrigerants that would be useful in the future designs of heat exchangers involving HC refrigerants. The local condensing heat transfer coefficient of all HC refrigerants were higher in smaller diameter tube and further, were generally higher by at least 31% than conventional R-22. The general trend indicated that the average condensing heat transfer coefficient increased with an increase of mass flux in smaller tube, and HC's have higher improvement rate than R-22. However, HC refrigerants suffer from the problem of higher pressure drops by at least 50% than R-22.

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