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Flow condensation heat transfer characteristics of hydrocarbon refrigerants and dimethyl ether inside a horizontal plain tube

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Abstract

Flow condensation heat transfer coefficients (HTCs) and pressure drop of R22, propylene, propane, DME and isobutane are measured on a horizontal plain tube. The main test section in the experimental flow loop is made of a plain copper tube of 8.8 mm inner diameter and 530 mm length. The refrigerant is cooled by passing cold water through the annulus surrounding the test section. Tests are performed at a fixed refrigerant saturation temperature of 40 ± 0.2 °C with mass fluxes of 100, 200, and 300 kg/m^2 s and heat flux of 7.3–7.7 kW/m^2. The heat transfer and pressure drop data are obtained in the vapor quality range of 10–90%. Test results show that for a given mass flux the flow condensation HTCs of propylene, propane, DME and isobutane are higher than those of R22 by up to 46.8%, 53.3%, 93.5% and 61.6%, respectively. Also well-known correlations developed based upon conventional fluorocarbon refrigerants predict the present data within a mean deviation of 33%. Finally, the pressure drop increases as the mass flux and quality increase and isobutane shows the highest pressure drop due to its lowest vapor pressure among the fluids tested.

Keywords: Flow condensation; Heat transfer coefficients; Pressure drop; Natural refrigerants; Propylene; Propane; DME; Isobutane

1. Introduction

CFCs have been widely used as working fluids for refrigeration and air-conditioning equipment for more than 50 years since their introduction in 1930s due to their excellent thermodynamic and chemical properties and good stability. These useful fluids, however, have been regulated and eventually phased out by the Montreal protocol signed in 1987 due to a discovery that chlorine atoms in CFCs destroy the stratospheric ozone layer. Furthermore, in 1997, Kyoto protocol was proposed to reduce the global warming effects, which calls for the energy efficiency improvement in all energy conversion devices including refrigeration and air-conditioning equipment (Global Environmental Change Report, 1997). Due to a concern for severe global warming, even newly developed HFCs such as R134a are now being regulated since they have high greenhouse warming potentials (GWPs).

In order to comply with the global environmental issues effectively, conventional refrigerants have to be changed to environmentally safe ones. At the same time, the performance of heat exchangers in refrigeration and air-conditioning equipment has to be improved to reduce the indirect green house warming caused by the use of electricity generated mainly by the combustion of fossil fuels. In fact, for most of the refrigerating and air-conditioning equipment, the indirect warming constitutes considerably larger part, typically 90%, of the total warming. To increase the heat exchanger performance, more research has to be carried out with new environmentally friendly alternative refrigerants.

One of the possible solutions to avoid CFCs, HCFCs and HFCs is the use of natural refrigerants such as hydrocarbons. For the past few decades, flammable hydrocarbon refrigerants have been prohibited in normal refrigeration and air-conditioning applications due to a safety concern.

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These days, however, this trend is being relaxed because of the environmental mandate. Therefore, some of the flammable refrigerants have been applied to certain applications (Kruse, 1996; Jung et al., 2000). Isobutane (R600a) has dominated the European refrigerator/freezer sector for the past decade and is being used even in Japan and Korea while propane (R290) and propylene (R1270) are used for heat pumping applications in Europe (Int. Energy Agency's Heat Pump Center, 2002). It is well known that hydrocarbons offer low cost, availability, compatibility with the conventional mineral oil, and environmental friendliness (Kruse, 1996; Jung et al., 2000).

Recently, some environmentally friendly refrigerant mixtures such as R429A and R432A are introduced as alternatives for R134a and R22, which include dimethyl ether (DME, RE170) as one of the components. Hydrocarbons and DME have very low GWPs, typically less than 3, as compared to 1300 and 1700 of R134a and R22. Due to the climate change all over the world, refrigerants with low GWPs are in great demand these days and certainly these fluids are good candidates for future use.

Due to the worldwide trend for the environmental protection, heat transfer research has been carried out employing hydrocarbons and their mixtures. Wen and Ho (2005) performed flow boiling heat transfer experiments for R290, R600, and R290/R600 mixture and found out that these are better than R134a from the view point of heat transfer and suggested a new correlation. Wen et al. (2006) also performed flow condensation heat transfer experiments for R290, R600, and R290/R600 mixture in the serpentine small-tube bank and found out that these fluids are again better than R134a from the view point of heat transfer.

Cho et al. (2005) performed also flow boiling heat transfer experiments using R32/R290 mixture inside a 5.0 mm diameter tube. On the other hand, Choi et al. (2006) carried out boiling experiments in mini-channels using R290. Recently, Park et al. (2005) showed that R290 has better heat transfer characteristics than R22 and R134a in flow condensation inside an aluminum multi channel flat tube. Finally, El Hajal et al. (2003) and Thome et al. (2003) presented two phase flow pattern map and new heat transfer model for horizontal flow condensation using the data of conventional HCFC and HFC fluids.

As one can see from the literature survey, little information is available on the flow condensation heat transfer characteristics of hydrocarbons and DME. The objectives of this study are to measure flow condensation heat transfer coefficients and pressure drop of R22, propylene (R1270), propane (R290), DME (RE170) and isobutane (R600a) inside a 8.8 mm inner diameter horizontal tube and to analyze and compare them against some correlations.

2. Experiments

2.1. Experimental apparatus

Fig. 1 shows the flow condensation heat transfer experimental facility which is composed of a refrigerant condensation loop, water loop with the constant temperature bath, and water/ethylene glycol loop with the chiller. The refrigerant loop consisted of a magnetic type refrigerant pump, filter, mass flow meter, pre-heater, calming section, main test section, plate type condenser, and an accumulator.

The main test section was of a tube-in tube configuration. The inner one was a copper tube of 8.8 mm inner diameter while the outer one was made of a thick insulating plastic. The annular space between the inner and outer tubes was 2 mm and water was passed in a counter current manner through this space to remove condensation heat from the refrigerant as shown in Fig. 2. The main test section was equipped with plastic flanges at both sides for easy mounting and assembly. The plastics used as the outer tube and flanges prevented heat transfer between the test section and the environment. As seen in Fig. 2, the length of the main test section for condensation heat transfer was

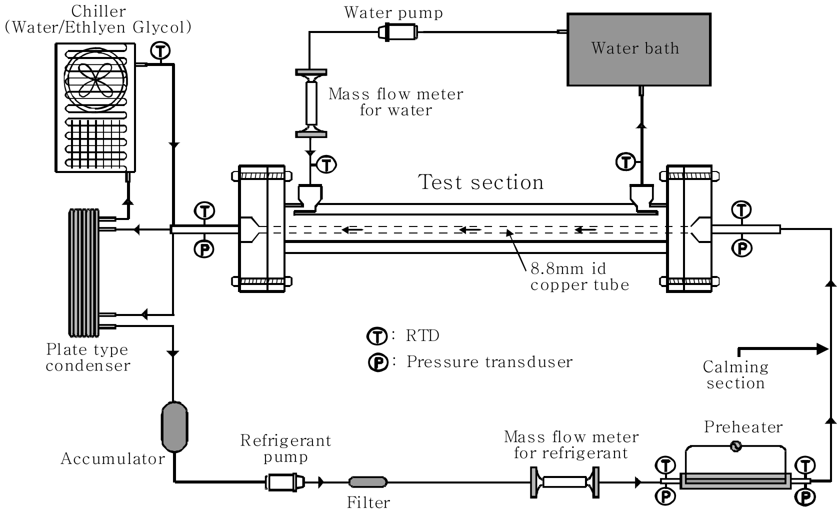

Fig. 1. Schematic diagram of test apparatus.

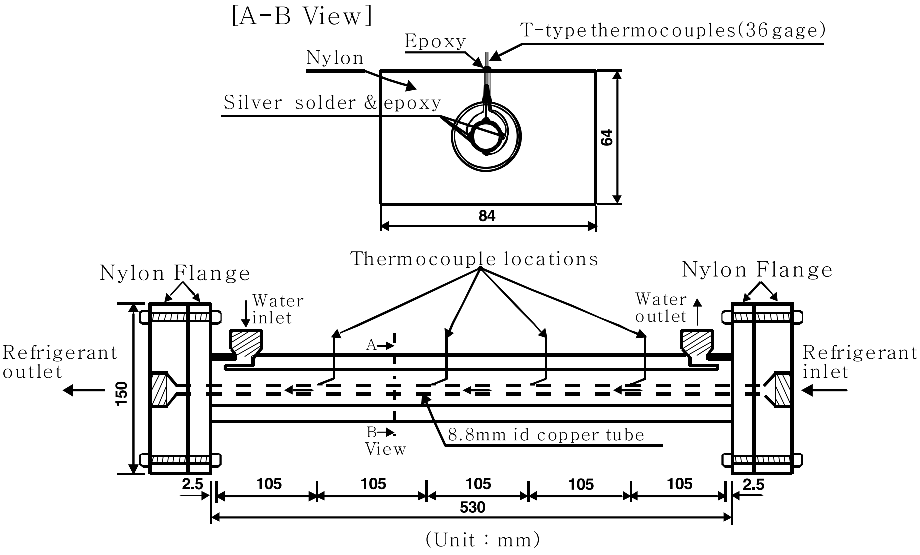

Fig. 2. Details of test section.

530 mm. T-type (copper–constantan) thermocouples were attached on the outside surface of the inner copper tube at four longitudinal locations to measure the wall temperatures directly. At each location, four thermocouples were attached to the surface 90° apart at top, bottom, and two sides of the tube. To attach thermocouples, small slits on test tubes were prepared carefully without damaging the inner surface of the test tube and fine thermocouples were first soldered to the surface and later covered by epoxy such that bare thermocouple wires were not extruded to touch the water flowing in the annulus.

To measure refrigerant temperatures at the inlet and outlet of the test tube, RTDs of 0.01 °C accuracy were used. Accurate pressure transducer and differential pressure transducer of 0.1% accuracy were used to measure the pressure at the inlet of the test section and pressure drop across the section respectively. All thermocouples used in this study were calibrated against a temperature calibrator of 0.01 °C accuracy while the pressure transducer was calibrated against a pressure calibrator of 0.1 kPa accuracy. The heat output from the condensing refrigerant was taken away by water flowing through the annulus of the test section. To determine the rate of heat transfer, the temperature difference in the water side across the test section was measured by a set of RTDs of 0.01 °C accuracy. Mass flow rates of both refrigerant and water were measured by precision mass flow meters of 0.2% accuracy. The power input to the pre-heater was measured by a digital power meter of 0.1% accuracy. All data were collected by a computer controlled data logger.

2.2. Experimental procedures and conditions

In this study, R22 and three hydrocarbons of propylene, propane, isobutane and DME were used as working fluids and measurements were taken at the condensation temperature of 40 ± 0.2 °C. For all working fluids tested, data were taken at three mass fluxes of 100, 200, and 300 kg/m^2 s under the average heat flux of 7.3–7.7 kW/m^2. These are the typical values found in residential air-conditioners.

Test procedures are as follows:

(1) A vacuum pump was turned on for 10 h to evacuate the refrigerant loop thoroughly.
(2) Proper amount of refrigerant was charged to the system and a chiller for the condenser and water bath for the test section were turned on.
(3) A refrigerant pump was turned on to deliver a desired mass flow rate. And then the pre-heater power, mass flow rate and temperature of water to the annulus of the test section were adjusted to desired values.
(4) When steady-state was achieved, data were taken for 30 min with an interval of 30 s at average refrigerant temperature in the test section of 40 °C. For a given refrigerant flow rate, the first data were usually taken at qualities close to 10–20%.
(5) Power to the pre-heater was increased to generate data at higher qualities for a given mass flux. Steady-state data were taken from the initial quality up to the maximum quality possible with a typical quality interval of 12%. Thus, for a given refrigerant flow rate, a number of experiments were carried out to cover the overall quality range of roughly 10–90%.

2.3. Data reduction

A pseudo local heat transfer coefficient for the test section was determined by Eq. (1).

$$h = \frac{Q_r}{(T_r - T_{wall})A} \tag{1}$$

where h, Q_r, T_r, T_{wall}, and A are the pseudo local heat transfer coefficient (W/m^2 K), heat transfer rate from refrigerant to water (W), refrigerant and wall surface temperatures (°C), and heat transfer area (m^2), respectively.

In flow condensation heat transfer experiments, heat is normally removed by a heat transfer fluid and hence it is very difficult to measure true local HTCs (Jung et al., 2003). The term 'pseudo local HTCs' means that the HTCs reported here are actually local average HTCs over a certain quality change across the test section. One can reduce the quality change across the test section for one measurement to make pseudo local HTCs reach close to true local HTCs. But for this, the length of the test section should be reduced greatly, which in turn would reduce the heat gain of the heat transfer fluid. Consequently, this would result in greater measurement uncertainties in HTCs due to the decreased temperature difference of water between the inlet and outlet. Therefore, for the past few decades, flow condensation heat transfer experiments have been carried out usually with quality changes of 10–30% across the test section (Jung et al., 2003). As mentioned, the present data were obtained with quality changes of less than 12% across the test section.

The heat transfer rate Q_r was determined by measuring the mass flow rate and temperature increase of water flowing inside the annulus of the test section as in Eq. (2)

$$Q_r = Q_w = \dot{m}_w C_{pw}(T_{w,\ out} - T_{w,\ in}) \tag{2}$$

where $Q_r, Q_w, \dot{m}_w, C_{pw}, T_{w,\ out}, T_{w,\ in}$ are the heat transfer rate on refrigerant and water sides (W), mass flow rate of water (kg/s), specific heat of water (J/kg K), and water temperatures at inlet and outlet of the test section (°C), respectively.

The average values of two fluid temperatures at the inlet and outlet of the tube and sixteen surface temperatures at four thermocouple locations were substituted for T_r and T_{wall} in Eq. (1) respectively for the determination of HTCs. For all pure refrigerants tested, the difference between the measured fluid temperature and the saturation temperature corresponding to the measured pressure was less than 0.1 °C.

Since a temperature drop between the actual inner wall and the wall thermocouple locations is expected due to conduction, a 1-dimensional steady-state conduction equation, Eq. (3), is applied to determine its magnitude

$$T_{wall} = T_{th} + \frac{Q}{2\pi L}\left[\frac{\ln(\mathrm{OD}/\mathrm{ID})}{k_{tube}}\right] \tag{3}$$

where T_{wall}, T_{th}, L, OD, ID, k_{tube} are the actual inner wall temperature (°C), measured temperature by a wall thermocouple (°C), length of the tube (m), outer and inner diameters of the tube (m), and thermal conductivity of the tube (W/m K), respectively.

Since the plain tube is made of copper, the temperature compensation term, $(T_{wall} - T_{th})$, in Eq. (3) is very small, typically less than 0.1 °C. Therefore, this term is not expected to play a significant role on the HTCs. Thus, in this study the measured wall temperatures are used directly in the calculation of HTCs.

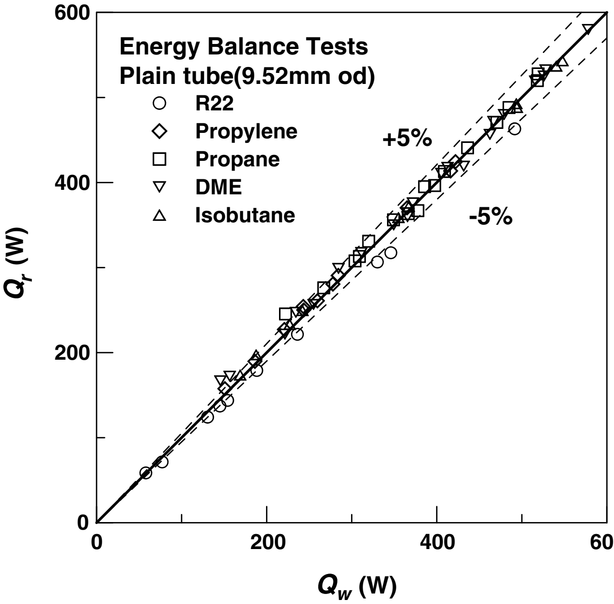

Fig. 3. Heat balance for single phase liquid flow tests.

The vapor quality at the entrance of the test section is calculated from an energy balance on the pre-heater section

$$x_{in} = \frac{Q_{pre} - \dot{m}_r C_{Pf}(T_{pre,\ out} - T_{pre,\ in})}{\dot{m}_r h_{fg}} \tag{4}$$

where $x_{in}, Q_{pre}, \dot{m}_r, C_{Pf}, T_{pre,\ out}, T_{pre,\ in}, h_{fg}$ are the quality entering the test section, heat transfer rate of the pre-heater (W), mass flow rate of refrigerant (kg/s), specific heat of refrigerant (J/kg K), refrigerant temperatures at the outlet and inlet of the pre-heater (°C), heat of evaporation (J/kg), respectively.

Before taking flow condensation data, single phase energy balance tests were carried out to check the validity of the instrumentation and test facility overall. Fig. 3 shows the test results. As one can see in this figure, the average deviation between the heat gain and heat loss of the refrigerant and water was less than 5% for all refrigerants tested. Also the single phase heat transfer coefficients thus measured agreed well with the well-known Dittus–Boeltor equation within a mean deviation of 5%. This indirectly validates the measurements.

Finally, the heat transfer measurement uncertainties were estimated by the method suggested by Kline and

Table 1
Measure parameters and their uncertainties

Parameter	Uncertainty
T_RTD (°C)	±0.01
T_Thermocouple (°C)	±0.1
Pressure (kPa)	±3.4
Pressure drop (kPa)	±0.2
Mass flow rate (kg/s)	±0.2%
Heat flux (kW/m^2)	1.1–1.5%
Heat transfer coefficient (kW/m^{2}K)	2.3–9.5%

McClintock (1953) and turned out to be less than 10% for the plain test tube. For reference, Table 1 lists the measured parameters and their measurement uncertainties.

3. Results and discussion

In this study, horizontal flow condensation heat transfer coefficients and pressure drop of R22, propylene, propane, DME, and isobutane were measured in a horizontal plain test tube at the saturation temperature of 40 ± 0.2 °C. For comparison of the data, R22 is used as a reference fluid in this report since it has been the dominant refrigerant in residential air-conditioners. It, of course, has to be phased out by the Montreal protocol due to ozone depleting chlorine in it. Propylene, propane, DME, and isobutane can be used either as a pure refrigerant or as one of the components of refrigerant mixtures such as R429A, R430A, R431A, R432A, R433A to replace R134a and R22. For reference, Table 2 lists some of the properties of working fluids tested in this study. In Table 2, P_{sat} is the saturated pressure, and ρ_{f}, ρ_{g}, h_{fg}, C_{pf}, k_{f}, μ_{f}, σ_{f} are the densities of saturated liquid and vapor (kg/m^3), heat of condensation (kJ/kg), and specific heat (kJ/kg K), thermal conductivity (mW/m K), viscosity (μPa s), and surface tension of the saturated liquid (N/m), respectively.

3.1. Effect of mass flux and quality on HTCs

Figs. 4–6 show the flow condensation HTCs as a function quality at three mass fluxes of 100, 200, and 300 kg/m^2 s respectively. Table 3 lists average heat transfer enhancement of three hydrocarbon refrigerants and DME as compared to R22. At the mass flux of 100 kg/m^2 s, the HTCs of propylene, propane, DME, isobutane are 5.8%, 4.7%, 32.0%, 19.5% higher than those of R22 respectively. On the other hand, at the mass flux of 200 kg/m^2 s, the HTCs of propylene, propane, DME, isobutane are 22.9%, 33.6%, 67.3%, 47.8% higher than those of R22, respectively. Finally, at the mass flux of 300 kg/m^2 s, the HTCs of propylene, propane, DME, isobutane are 46.8%, 53.3%, 93.5%, 61.6% higher than those of R22, respectively. From these results, one can see that for all refrigerants considered in this study the HTCs increase with an increase in mass flux and also the difference in HTCs between the individual refrigerant and R22 increases as well.

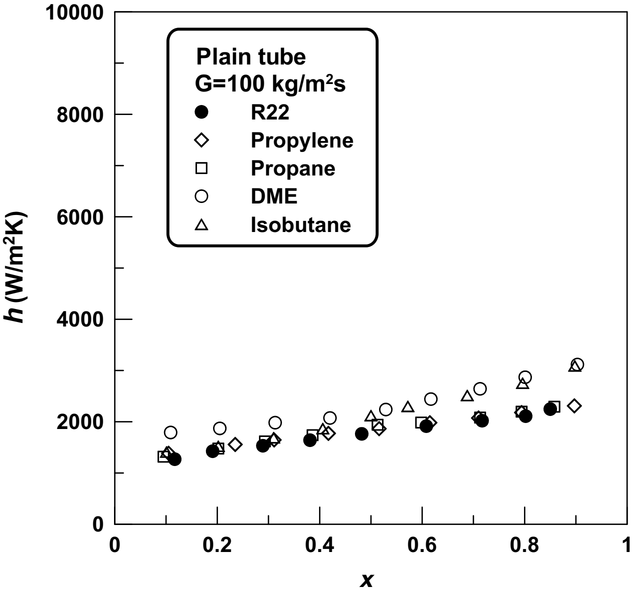

Fig. 4. HTCs of R22, propylene, propane, DME and isobutane at 100 kg/m^2 s in a plain tube.

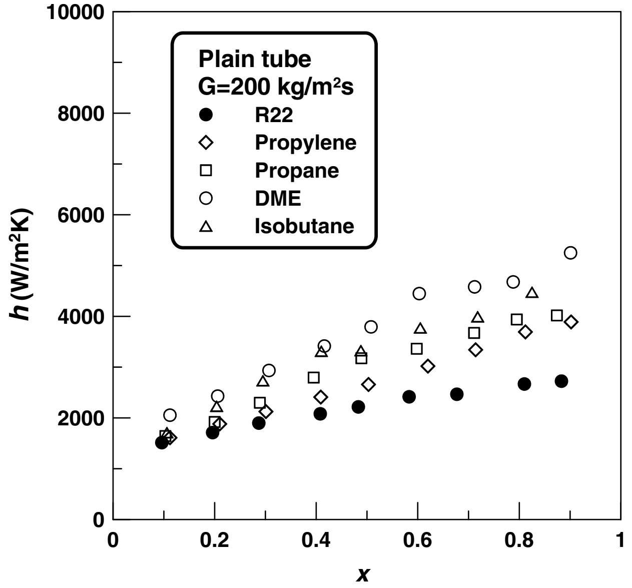

Fig. 5. HTCs of R22, propylene, propane, DME and isobutane at 200 kg/m^2 s in a plain tube.

In general, the HTCs of three hydrocarbon refrigerants and DME tested in this study were higher than those of R22 at all mass fluxes. This can be explained by observing the difference in physical properties of these refrigerants

Table 2
Physical properties of working fluids tested at 40 °C

Fluids	P_{sat} (kPa)	ρ_{f} (kg/m^3)	ρ_{g} (kg/m^3)	h_{fg} (kJ/kg)	C_{Pf} (kJ/kg K)	k_{f} (mW/m K)	μ_{f} (μPa s)	σ (N/m)	Φ	X_{tt} (at $x = 0.5$)
R22	1534	1129	66.2	166.56	1.339	76.60	138.67	0.00604	2.11	0.306
Propylene	1652	477	35.7	303.14	2.878	102.91	81.632	0.00486	4.23	0.338
Propane	1370	467	30.2	306.51	2.927	86.80	82.639	0.00521	3.83	0.318
DME	885	631	18.7	375.24	2.471	135.29	111.72	0.00892	4.24	0.219
Isobutane	531	530	13.7	311.40	2.554	83.69	128.23	0.00841	2.97	0.212

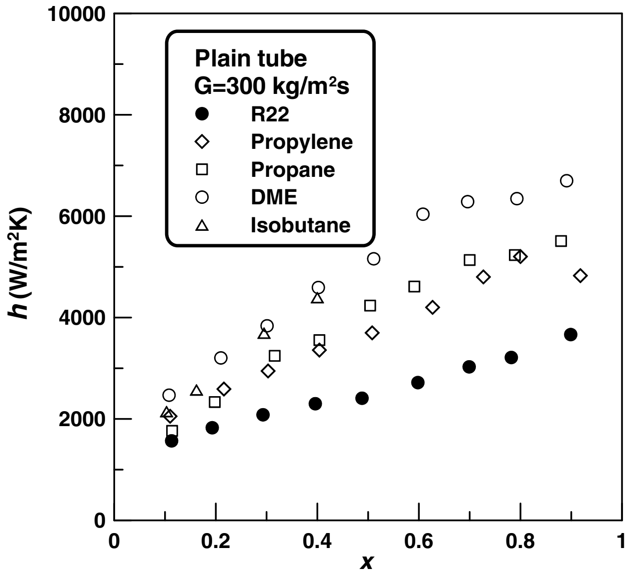

Fig. 6. HTCs of R22, propylene, propane, DME and isobutane at 300 kg/m^2 s in a plain tube.

Table 3
Average flow condensation heat transfer improvement of various fluids over R22 at various mass fluxes

G (kg/m^2 s)	100 (%)	200 (%)	300 (%)
Propylene	5.8	22.9	46.8
Propane	4.7	33.6	53.3
DME	32.0	67.3	93.5
Isobutane	19.5	47.8	61.6

and R22. Jung et al. (2003) presented a flow condensation heat transfer correlation based on measured data of seven well-known halocarbon refrigerants as follows:

$$h = 22.4h_{\mathrm{f}}\left(1+\frac{2}{X_{\mathrm{tt}}}\right)^{0.81}\left(\frac{Q}{h_{\mathrm{fg}}GA}\right)^{0.33}\left(\frac{k_{\mathrm{f}}}{D}\right) \tag{5}$$

where

$$h_{\mathrm{f}} = 0.023Re_{\mathrm{f}}^{0.8}Pr_{\mathrm{f}}^{0.4}$$

$$X_{\mathrm{tt}} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_{\mathrm{g}}}{\rho_{\mathrm{f}}}\right)^{0.5}\left(\frac{\mu_{\mathrm{f}}}{\mu_{\mathrm{g}}}\right)^{0.1}$$

where X_{tt}, Q, h_{fg}, G, A, k_{f}, D, Re_{f}, Pr_{f}, x, ρ_{f}, ρ_{g}, μ_{f}, and μ_{g} are the Martinelli parameter, heat transfer rate (kW), heat of condensation (kJ/kg), mass flux (kg/m^2 s), area (m^2), thermal conductivity of saturated liquid (W/m K), diameter of the tube (m), Reynolds and Prandtl numbers of the saturated liquid, quality, densities of saturated liquid and vapor (kg/m^3), and viscosities of saturated liquid and vapor (Pa s), respectively.

Basically, Eq. (5) is a modification of a general two phase flow heat transfer correlation form utilizing Dittus–Boelter equation for single phase liquid flow with Martinelli parameter, X_{tt} to consider two phase effect. As seen in Eq. (5), the flow condensation HTCs of a given fluid is directly proportional to the liquid property combination term, $\Phi = (C_{\mathrm{Pf}}/\mu_{\mathrm{f}})^{0.4}k_{\mathrm{f}}^{0.6}$, as suggested by Jung et al. (1989). As listed in Table 2, the values of the liquid property term of three hydrocarbon refrigerants and DME are much higher than that of R22. Especially, that of DME is even 100% higher than that of R22.

From Eq. (5), it is expected that the HTCs would be increased with a decrease in X_{tt}, a dimensionless parameter representing the ratios of the liquid and vapor's densities and viscosities as listed in Table 2. At quality of 0.5, X_{tt} of propylene and propane are 10% and 3% higher than that of R22 while those of isobutane and DME are 30% lower than that of R22. Consequently, one can expect that the HTCs of isobutane and DME would be higher than those of propylene and propane. In fact, this is well confirmed by the data shown in Figs. 4–6.

From the test data, one can safely conclude that the flow condensation HTCs of three hydrocarbons and DME are higher than those of R22 at all mass fluxes due to their excellent thermophysical properties and hence they can be used as long term alternative refrigerants from the view point of heat transfer.

3.2. Comparison with heat transfer correlations

Table 4 and Fig. 7 show the comparison of the present data against some of the well-known correlations by Akers et al. (1959), Cavallini and Zecchin (1974), Dobson and Chato (1998), Jung et al. (2003), Shah (1979), Soliman et al. (1968), and Traviss et al. (1973).

Table 4
Deviations of various correlations against the present flow condensation heat transfer data

Fluid	Akers et al. (1959)		Cavallini and Zecchin (1974)		Dobson and Chato (1998)		Jung et al. (2003)		Shah (1979)		Soliman et al. (1968)		Traviss et al. (1973)	
	Average	Mean	Average	Mean	Average	Mean	Average	Mean	Average	Mean	Average	Mean	Average	Mean
R22	−45.4	45.4	2.8	20.1	−2.9	20.7	−4.4	8.3	−9.0	18.3	−24.6	26.5	−0.3	20.6
Propylene	−20.1	20.1	50.3	50.3	40.2	41.5	15.7	16.7	36.5	37.6	13.9	19.3	46.1	46.9
Propane	−27.3	27.6	36.7	36.7	30.5	32.0	7.9	11.8	23.6	25.0	11.2	15.7	36.7	37.6
DME	−29.3	29.3	33.0	34.0	28.2	31.1	−3..3	11.5	17.0	20.1	−0.6	11.3	28.7	30.4
Isobutane	−33.9	33.9	24.2	25.0	21.5	23.9	3.3	9.5	10.8	13.7	5.5	13.9	24.4	25.9
All	−31.2	31.3	29.4	33.2	23.5	29.8	3.9	11.5	15.8	22.9	1.1	17.3	27.1	32.3

Average deviation $= \frac{1}{n}\sum_{1}^{n}\left[\frac{(h_{\mathrm{cal}}-h_{\mathrm{exp}})\times 100}{h_{\mathrm{exp}}}\right]$ Mean deviation $= \frac{1}{n}\sum_{1}^{n}\mathrm{ABS}\left[\frac{(h_{\mathrm{cal}}-h_{\mathrm{exp}})\times 100}{h_{\mathrm{exp}}}\right]$.

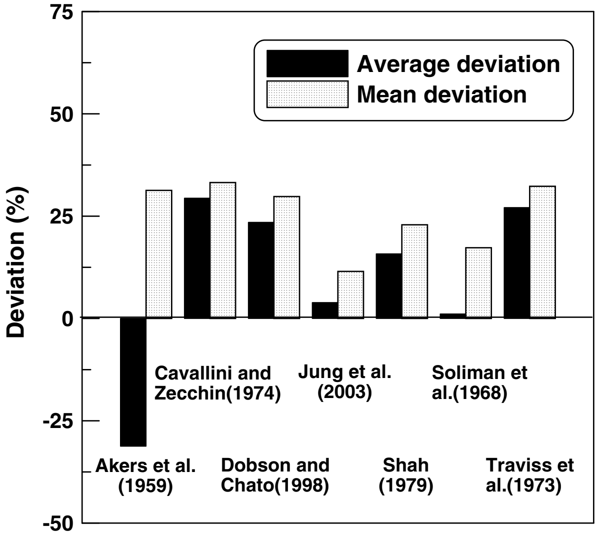

Fig. 7. Deviation of various heat transfer correlations against the present data.

All correlations predicted the data reasonably well within a mean deviation of 30%. Especially, the correlation by Jung et al. (2003) predicted the present data with a mean deviation of 11.5% as seen in Fig. 8. From this comparison, it can be concluded that previous correlations developed mainly based on conventional fluorocarbon refrigerants can be used to predict the HTCs of hydrocarbons and DME within 33% deviation. Considering the complexity of the flow condensation heat transfer phenomenon, 30% deviation is not a bad approximation to start with. Especially, that of Jung et al. (2003) is very good predicting both fluorocarbon and hydrocarbon refrigerants and DME data within 12% deviation.

3.3. Pressure drop

Fig. 9 shows the measured pressure drop as a function of quality at mass flux of 200 kg/m^2 s. As seen in Fig. 9, the pressure drop of all refrigerants increased as the quality and mass flux increased. One can also see that the pressure drop is inversely proportional to the vapor pressures of the refrigerants. Thus, isobutane, the lowest vapor pressure fluid among the fluids tested in this study, showed the highest pressure drop. A similar trend was also observed at mass fluxes of 100 and 300 kg/m^2 s.

Jung and Radermacher (1989) proposed a pressure drop correlation of an annular flow modifying Martinelli and Nelson (1948) correlation. As seen in Eq. (6) by Jung and Radermacher (1989), the pressure drop is directly proportional to the Martinelli parameter, X_{tt}. As listed in Table 2, X_{tt} decreases with a decrease in vapor pressure. Therefore, the pressure drop increases with a decrease in vapor pressure and the present data confirm this

$$\Delta P_{\mathrm{tp}} = \frac{2f_{\mathrm{fo}}G^2L}{D\rho_{\mathrm{f}}}\left[\frac{1}{x}\int_0^x \phi_{\mathrm{tp}}^2 dx\right] \tag{6}$$

where

$$\phi_{\mathrm{tp}}^2 = 12.82X_{\mathrm{tt}}^{-1.47}(1-x)^{1.8}$$

$$f_{\mathrm{fo}} = 0.046Re^{-0.2} \quad (5000 < Re < 200,000)$$

where ΔP_{tp}, $\mathrm{f_{fo}}$, L, ϕ_{tp} are the pressure drop of two phase flow (Pa), friction factor of total flow assumed as liquid, length of test tube (m), and pressure drop multiplier, respectively.

As for propylene and propane, their vapor pressures are similar to that of R22 but they showed higher pressure drop than R22. This is because the pressure drop is inversely proportional to the liquid density as seen in Eq. (6). As

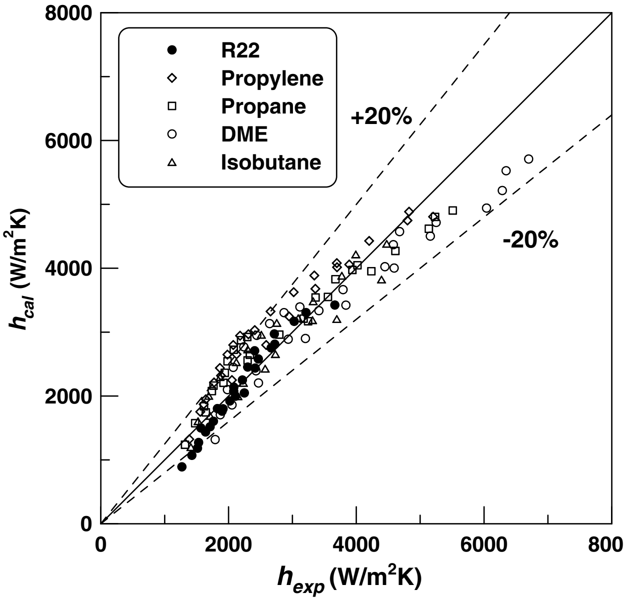

Fig. 8. Comparison of HTCs with Jung et al. (2003) correlation.

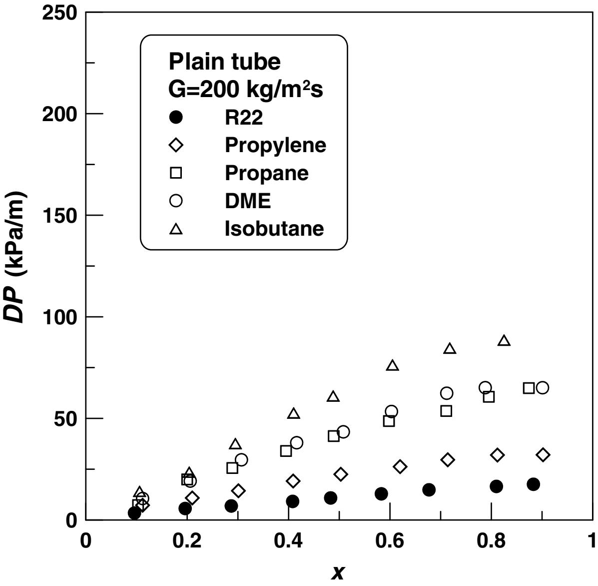

Fig. 9. Pressure drop of R22, propylene, propane, DME and isobutane at 200 kg/m^2 s in a plain tube.

listed in Table 2, the liquid densities of propylene and propane are 2.4 times lower than that of R22 and thus their pressure drop was higher than that of R22 under the same condition.

4. Conclusions

In this study, flow condensation HTCs and pressure drop of R22 and three hydrocarbon refrigerants of propylene, propane, isobutane, and DME are measured and from the test data, the following conclusions are drawn.

(1) For all refrigerants tested, the flow condensation HTCs increased as the quality and mass flux increased.
(2) Flow condensation HTCs of propylene, propane, DME, and isobutane were higher than those of R22 for all mass fluxes considered by up to 46.8%, 53.3%, 93.5%, and 61.6%, respectively.
(3) Previous correlations available in the literature predicted the present data within a mean deviation of 30%. Especially, the correlation of Jung et al. (2003) showed the best prediction capability with a mean deviation of 11.5%.
(4) For all refrigerants tested, the pressure drop increased as the quality and mass flux increased. In general, the pressure drop increased with a decrease in vapor pressure.

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