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Condensation heat transfer coefficients of HFC245fa on a horizontal plain tube †

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Abstract

Condensation heat transfer coefficients (HTCs) of HCFC22, HCFC123, HFC134a and HFC245fa are measured on a horizontal plain tube 19.0 mm outside diameter. All data are taken at the vapor temperature of 39℃ with a wall subcooling temperature of 3-8℃. Test results show the HTCs of newly developed alternative low vapor pressure refrigerant, HFC245fa, on a smooth tube are 9.5% higher than those of HCFC123, while they are 3.3% and 5.6% lower than those of HFC134a and HCFC22, respectively. Nusselt's prediction equation for a smooth tube underpredicts the measured data by 13.7% for all refrigerants, while a modified equation yielded 5.9% deviation against all measured data. From the view point of environmental safety and condensation heat transfer, HFC245fa is a long-term good candidate to replace HCFC123 used in centrifugal chillers.

Keywords: Condensation heat transfer; Heat transfer coefficients; HFC245fa; Ozone layer depletion

1. Introduction

Chlorofluorocarbon (CFCs) were used as working fluids for residential and commercial refrigeration and air-conditioning equipment for more than 50 years since their introduction in 1930s. These useful fluids, however, were regulated and eventually phased out since Molina and Rowland [1] discovered in 1974 that chlorine atoms in CFCs destroy the stratospheric ozone layer. In 1997, the Kyoto protocol was proposed to reduce the green house warming, which calls for the energy efficiency improvement in all energy conversion devices including refrigeration equipment [2]. 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 equipment, the indirect warming effect is more than 95% of the total warming. To increase the heat exchanger performance, research has to be carried out with new alternative refrigerants [3, 4].

As the living standard gets improved in many countries, people demand more comfortable indoor environment and thus central air-conditioning has become an essential component of modern society. For this application, centrifugal chillers are employed in most of the large commercial buildings.

In the past, CFC11 was used predominantly in centrifugal chillers. After the advent of the Montreal protocol in 1987, HCFC123 and HFC134a have been used in newly manufactured chillers replacing CFC11 [5]. It is known that HCFC123 is energy efficient and has a similar low operating pressure to the conventional CFC11. Since HCFC123 still contains ozone-depleting chlorine, however, it must be replaced by some alternative refrigerants for the protection of the stratospheric ozone layer. On the other hand, HFC134a has a much higher operating pressure and hence the chiller system can be made compact. The entire system, however, has to be redesigned due to a significant change in vapor pressure. At the same time, HFC134a was classified as one of the greenhouse gases by Kyoto protocol of 1997. It is also known that inherent thermodynamic efficiency of HFC134a is lower than that of CFC11 and HCFC123. For these reasons, it is necessary to develop environmentally friendly chillers with low vapor pressure refrigerant of low global warming and no ozone depletion.

Under this situation, Honeywell introduced HFC245fa to replace HCFC123 and HFC134a in centrifugal chillers. HFC245fa has no ozone depletion potential and its global warming potential and atmospheric life time are 30% and 50% lower than those of HFC134,a respectively [6]. Because of these good environmental properties, the Ebara company in Japan has developed commercial chillers with HFC245a for the past few years and is marketing them in Japan [7]. Also,

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some chiller manufacturers in the US have undertaken similar research and development activities to utilize HFC245fa in their new chillers.

For the past decade, some research activities to use HFC245fa in refrigeration and air-conditioning system have been undertaken. Thermal stability tests were performed by Angelino and Invernizzi [8] in an attempt to use HFC245fa to replace CFC11 and HCFC123 in low pressure centrifugal chillers.

In the development of new chillers, external condensation heat transfer coefficients (HTCs) are necessary. Most of the condensation heat transfer studies, however, were confined to such fluids as water, n-pentane n-butane, CFCs, HCFCs, and HFCs [9-11]. Recently, Jung et al. [12] and Hwang et al. [13] measured the external condensation HTCs of CFC11, CFC12, and HCFC22 and their alternatives. Also Jung et al. [14, 15] measured the external condensation HTCs of flammable refrigerants of R1270, R290, R600a, R600, RE170, R32 and their mixtures. In 2008, Park and Jung [16] measured the external condensation HTCs of R22, R410A, R407C, R134a on two enhanced tubes at three saturation temperatures.

As revealed in the literature survey, there have been no heat transfer data of HFC245fa available in the public domain. In this study, external condensation HTCs of HFC245fa, HCFC22, HCFC123, and HFC134a are measured to provide basic condensation heat transfer data to chiller industry for efficient design of environmentally friendly centrifugal chillers.

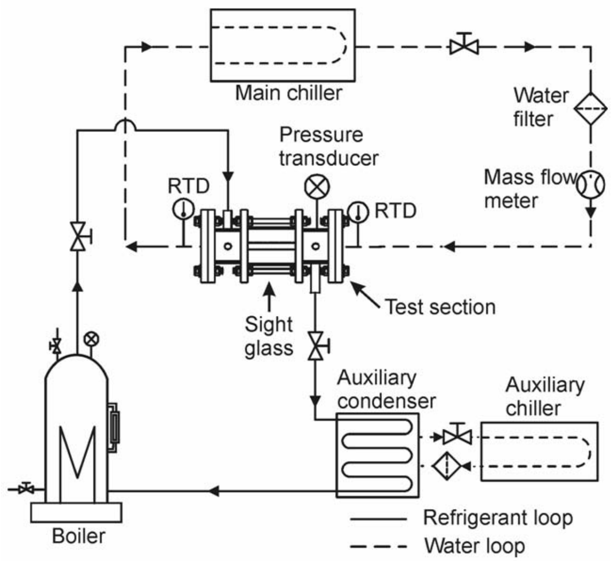

Fig. 1. Schematic diagram of the condensation heat transfer experimental apparatus.

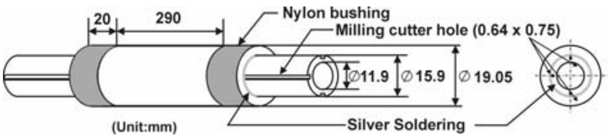

Fig. 2. Detailed description of test tube.

2. Experiments

2.1 Overall description of experimental apparatus

Fig. 1 shows the schematic diagram of the experimental apparatus. The facility is composed of the refrigerant and cooling water loops. The refrigerant vapor supplied to the test section was generated by an immersion heater of 3.5kW in the boiler located at the bottom of the apparatus. The vapor generated was fed to the main test section through a connecting pipe and condensed via counter-current heat exchange with the cooling water flowing inside a test tube. The condensate as well as the uncondensed vapor went into a large capacity auxiliary condenser and were cooled there and finally returned to the bottom of the boiler. The cooling water for the test section and for the auxiliary condenser was supplied by two independent external chillers that were capable of controlling the temperature with an accuracy of 0.1℃ as shown in Fig. 1.

The main test section was made of an 80 mm ID stainless steel pipe with a 110 mm long sight glass installed in the middle to observe the condensation phenomenon. Both ends of the test section were flanged for easy mounting of the test tube. When the cooling water flows inside the test tube and absorbs heat from the vapor, heat may flow from the water to the flanges at both ends of the test section where they touch the test tube. To prevent this from happening, at both ends of the test section nylon bushings of low thermal conductivity (Monomer cast nylon, 20 mm×1.5 mm) were tightly fastened on the tube so that heat transfer may occur only on the test tube.

As shown in Fig. 2, a copper tube of 15.9 mm OD and 2.0 mm thick was prepared with 0.64 mm wide slits located 90° apart at the top, side, and bottom of the tube. These slits were prepared by a milling cutter on the tube in a longitudinal direction from one end to the other. Then, this plain tube was tightly inserted into the test tube with stainless steel wires of 0.6 mm diameter placed into the slits across the entire length of the tube. And then, these two tubes were silver soldered together and the wires were pulled out at the final stage. Through this procedure, slits accommodating 0.5 mm TCs could be made at locations roughly 1.0 mm beneath the surface as illustrated in Fig. 2. If the tubes are not well soldered together, then the temperatures will vary quite significantly in the longitudinal direction resulting in erroneous data. Hence, a consistent way of joining the two tubes was tried many times and thus soldered tubes were cut into many sections to make sure that the silver solder flowed well into every space in between the tubes. Finally, the best tubes showing good repeatability were selected for the tests. With this procedure, any commercial tube with a smooth inner surface can be tested without altering the surface condition.

2.2 Measurements

One of the most important parameters in determining HTCs accurately is the measurement of the heat transfer rate to the cooling water. The heat transfer rate in the main test section, Q, was determined by a simple energy balance equation as follows:

$$Q = \dot{m}_w C_{pw} (T_{wo} - T_{wi}) \tag{1}$$

where $\dot{m}_w$, C_{pw}, T_{wo}, T_{wi} are the mass flow rate (kg/s), specific heat (kJ/kg · K), and temperatures of the cooling water at the outlet and inlet of the tube (℃), respectively.

In this study, a mass flow meter of 0.2% accuracy utilizing the Coriolis force principle was used to measure the flow rate of the cooling water. The cooling water, supplied by an external chiller, was passed through a filter for the removal of the contaminants before entering into the test section.

With a low mass flow rate, the temperature rise of the cooling water across the tube would be quite high, resulting in a severe temperature gradient along the test tube. To prevent this from happening and to maintain uniform temperature on the surface as much as possible, the mass flow rate should be fairly large. For this case, the temperature rise of the water would be at about 1-2℃ and hence the temperature difference between the outlet and inlet of the tube should be measured accurately. Otherwise, a 10-20% error can easily occur. To tackle this difficulty, a set of resistance temperature detectors (RTDs) of 0.01℃ accuracy was employed to measure the temperature difference directly.

Five TCs were in the vapor space along the tube for vapor temperatures, which were placed 20.0 mm away from the test tube. The vapor temperature inside the test section, T_{sat}, was assumed to be the average of these temperatures. Stainless steel sheathed copper-constantan TCs of 0.5 mm OD were inserted into the slits of the test tube and placed in the middle of the tube longitudinally for surface temperatures. A coil spring of 8.0 mm OD and 6.0 mm ID was inserted in the cooling water passage of the test tube in order to decrease the thermal resistance in the cooling water side. All TCs used in the measurements were calibrated before their use by using an RTD of 0.01℃ accuracy. Finally, a pressure transducer of 0.2% accuracy was calibrated and used to measure the pressure of the main test section.

2.3 Test procedures

Condensation heat transfer is affected greatly by the non-condensable gases present in the system and hence degassing is very important for the precise measurements. For this purpose, refrigerant was charged to the system and boiled and purged a few times through a valve located on top of the main test section. This was done until the measured vapor temperature and the saturation temperature at the measured pressure agreed with each other within 0.1℃.

The experimental procedure was as follows:

(1) Nitrogen was charged to the refrigerant loop up to 2000 kPa with some halogenated refrigerants to check by a halogen detector if there was any leak.
(2) A vacuum pump was turned on for about 2 hours to take out all gases.
(3) The refrigerant for the test was charged to the boiler up to 100 mm higher than the top of the heater, and the electricity to the heater was provided.
(4) The main test section was maintained at saturated temperature through the use of an external chiller and the non-condensable gases were purged.
(5) Desired wall subcooling was maintained by controlling the mass flow rate, and the temperature of the cooling water and all variables were recorded under steady-state. The wall subcooling was varied from 3-8℃ for a given fluid.
(6) The refrigerant was changed and the same procedures were repeated.

Table 1. Experimental condition.

Test refrigerants	HCFC22, HCFC123, HFC134a, HFC245fa
Sat. vapor temp.	39℃
Flow rate (coolant)	60 g/s
Wall subcooling	3℃ ~ 8℃ (at intervals of 1℃)
Heat transfer area	0.01736 m^2

Table 1 shows the overall experimental conditions of this study.

2.4 Data reduction

The local condensation HTC was determined by the following equation.

$$h = \frac{Q / A}{(T_{sat} - T_{wall})} \tag{2}$$

where h, Q, A, T_{sat}, T_{wall} are the HTC (W/m^2 · K), heat transfer rate (W), heat transfer area (m^2), vapor and surface temperatures (℃), respectively. For the plain tube tested, the nominal area based on the outside diameter was used as the area in Eq. (2).

Since there would be a temperature drop from the actual surface to the wall thermocouple locations, a 1-D steady-state conduction equation, Eq. (3), was applied to determine its magnitude.

$$T_{wall} = T_t + \frac{Q}{2\pi L}\left[\frac{\ln(r / r_t)}{k_{tube}}\right] \tag{3}$$

where T_t, L, r, r_t, k_{tube} are measured temperature by a wall thermocouple (℃), length of the tube (m), radius of the tube (m), the distance from the center of the tube to the thermocouple (m), thermal conductivity of the tube (W/m · K) respectively.

Since the plain tube was made of copper, the temperature

Table 2. Saturation properties of tested refrigerants at 39℃.

Refrigerants	P_{sat} (kPa)	ρ_f (kg/m^3)	ρ_g (kg/m^3)	C_{Pf} (kJ/kg·K)	h_{fg} (kJ/kg)	k_f (W/m·K)	μ_f (μ Pa·s)
HCFC22	1497	1133	64.5	1.332	167.76	0.077	140.3
HCFC123	149	1427	9.3	1.037	165.39	0.073	356.3
HFC134a	990	1151	48.7	1.493	164.07	0.075	163.5
HFC245fa	242	1300	13.7	1.355	181.78	0.084	342.0

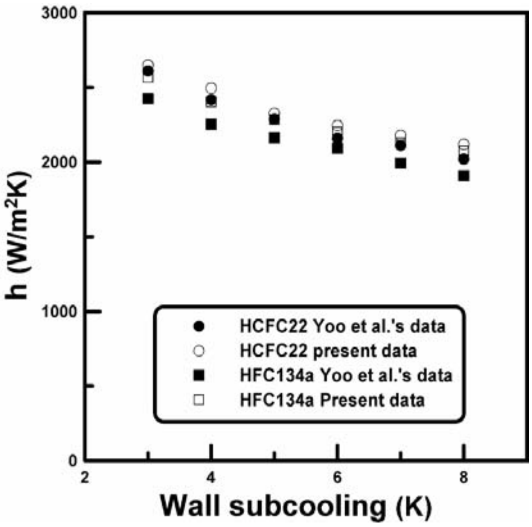

Fig. 3. Comparison of HCFC22 and HFC134a with Yoo et al.'s data [19].

compensation term, $(T_{wall} - T_t)$, in Eq. (3) was very small, typically less than 0.1℃. Therefore, this term did not have any significant effect on the HTCs and the measured wall temperatures were used directly in the calculation of HTCs.

The measurement uncertainties were estimated by the method suggested by Kline and McClintock [17]. In general, the measurement uncertainties increased as the wall subcooling decreased. The uncertainties were estimated to be 1.8-5.5% for the wall subcooling range applied in this study.

3. Results and discussion

In this study, external condensation HTCs of four refrigerants of HCFC22, HCFC123, HFC134a and HFC245fa were measured at the vapor temperature of 39℃ on a horizontal plain tube of 19.0 mm outside diameter with the wall subcooling of 3-8℃. Table 2 lists some relevant thermophysical properties of these refrigerants at 39℃, which were obtained by NIST REFPROP program [18].

3.1 Experiment with horizontal plain tube

First, experiments with HCFC22 and HFC134a were performed to check the reliability of the test method and facility. Fig. 3 shows a comparison between the present data and the data for HCFC22 and HFC134a taken by Yoo et al. [19]. Average deviations between Yoo et al. [19]'s and present data for

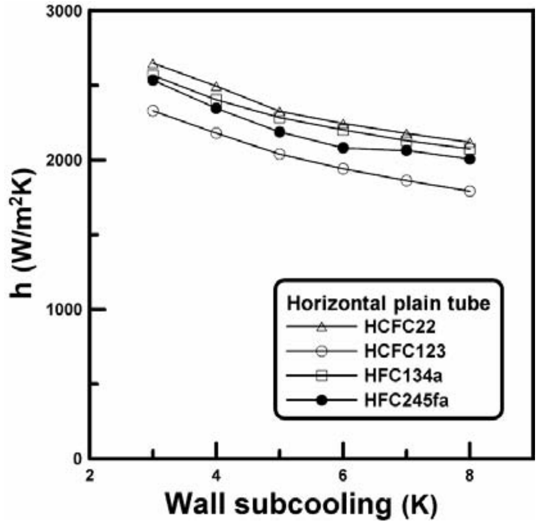

Fig. 4. Condensation HTCs as a function wall subcooling on a plain tube.

HCFC22 and HFC134a are 3% and 6%, respectively. From this result, the reliability of the present experimental work is indirectly confirmed.

Fig. 4 shows the external condensation HTCs of HCFC 22, HCFC123, HFC134a and HFC245fa. As seen in Fig. 4, the external condensation HTCs increased as the vapor pressure of the refrigerants increased at the same temperature. Also, the external condensation HTCs decreased as the wall subcooling increased. This is a typical trend observed in horizontal condensation heat transfer due to the increase in film thickness with an increase in wall subcooling.

Condensation HTCs of HFC245fa are 9.5% higher than those of HCFC123, while they are 3.3% and 5.6% lower than those of HFC134a and HCFC22, respectively. As seen in Nusselt's correlation [20], Eq. (4), external condensation HTCs are proportional to the heat of condensation and liquid thermal conductivity and they are indirectly proportional to the liquid viscosity.

$$h_{Nusselt} = 0.725\left[\frac{\rho_f(\rho_f-\rho_g)gk_f^3h_{fg}}{\mu_f \Delta TD}\right]^{1/4} \tag{4}$$

As can be seen in Table 2, heat of condensation and liquid thermal conductivity of HFC245fa are 10% and 15% higher than those of HCFC123 while the liquid viscosity of HFC245fa is 4% lower than that of HCFC123. Accordingly, one can expect that condensation HTCs of HFC245fa would be higher than those of HCFC123 from the consideration of thermophysical properties alone. As for HCFC22 and HFC134a, their viscosities are half the viscosity of HFC245a with 10% decrease in heat of condensation. Thus, one can expect that condensation HTCs of HCFC22 and HFC134a would be higher than those of HFC245a and Fig. 4 reflects this trend. From the view point of condensation heat transfer, even though HCFC123 and HFC134a are presently used in

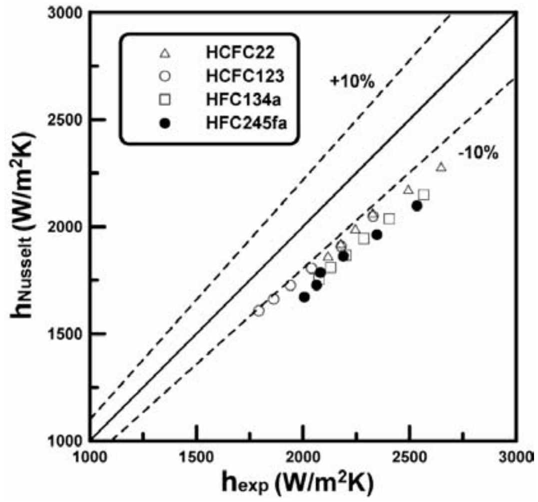

Fig. 5. Comparison of the present data with Nusselt's equation [20].

centrifugal chillers, more environmentally friendly refrigerant of HFC245fa may replace them successfully without a significant change in condenser size.

3.2 Comparison with correlations

Fig. 5 shows the comparison between the measure data and prediction correlation by Nusselt [20], Eq. (4), for HCFC22, HCFC123¸ HFC134a and HFC245fa. As seen in Fig. 5, the correlation underpredicted the measured data for HCFC22, HCFC123¸ HFC134a and HFC245fa by 12.2%, 11.4%, 15.3% and 16.0%, respectively. Nusselt's correlation was derived based upon an assumption that condensation film is laminar [21]. But in reality, the condensation film formed on the outer tube surface is usually wavy and turbulent. Thus, it is normal that measured HTCs are 10-15% higher than those of the prediction, as also seen in Kim et al. [22], Wanniarachchi et al. [23], and Marto et al. [24].

In our laboratory, external condensation HTCs of more than 15 refrigerants have been measured consistently for the past 15 years. In fact, Jung et al. [14] measured condensation HTCs of CFCs, HCFCs, HFCs, and hydrocarbons which can be used in refrigeration and air-conditioning equipment as a pure fluid or as one of the components of mixed refrigerants. Based upon the measured data, Jung et al. [14] increased the constant in Nusselt's correlation by 9.0% to account for the turbulent flow and suggested the following correlation for design engineers.

$$h_{modified\ Nusselt} = 0.79\left[\frac{\rho_f(\rho_f-\rho_g)gk_f^3h_{fg}}{\mu_f\Delta TD}\right]^{1/4} \quad (5)$$

Fig. 6 shows the comparison of the modified correlation and the measured data. The average deviation between the modified correlation and the measured data was less than 6.0% for all refrigerants including a new environmentally friendly refrigerant of HFC245fa.

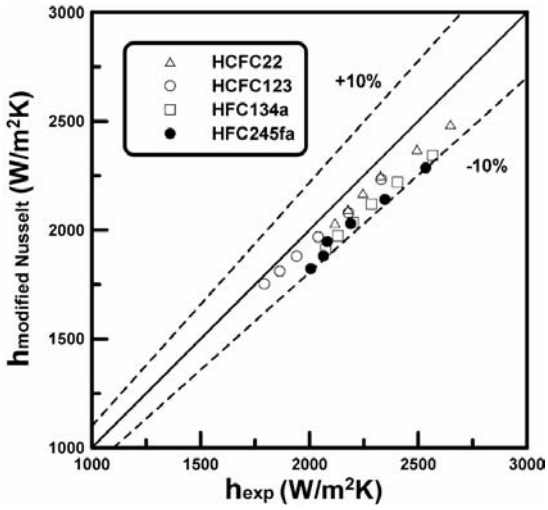

Fig. 6. Comparison of the present data with a modified Nusselt's equation.

4. Conclusions

In this study, external condensation heat transfer coefficients of HCFC22, HCFC123 ¸ HFC134a and a new alternative refrigerant of HFC245fa are measured on a horizontal smooth tube at 39℃ with the wall subcooling of 3-8℃. From the experimental data, following conclusions can be drawn.

(1) The external condensation HTCs are a strong function of vapor pressure and they increased as the vapor pressure of the refrigerants increased at the same temperature. Also, the external condensation HTCs decreased as the wall subcooling increased.
(2) The condensation HTCs of HFC245fa are 9.5% higher than those of HCFC123, while they are 3.3% and 5.6% lower than those of HFC134a and HCFC22, respectively.
(3) For all refrigerants, Nusselt's correlation underpredicted the measured data with an average deviation of 13.7%.
(4) From the viewpoint of condensation heat transfer, HFC245fa is a good alternative low pressure refrigerant which may replace HCFC123 in centrifugal chillers.

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Nomenclature

A : Heat transfer area (m^2)
C : Specific heat (kJ/kg·K)
D : Diameter (m)
g : Gravitational acceleration (m/s^2)
h : Heat transfer coefficient ($W/m^2\cdot K$)
h : Heat of condensation (kJ/kg)

k : Thermal conductivity (W/m·K)
L : Length (m)
$\dot{m}$: Mass flow rate (kg/s)
P : Pressure (kPa)
Q : Heat transfer rate (W)
r : Radius (m)
T : Temperature (K)
ΔT : Wall subcooling (K)

Greek letters

ρ : Density (kg/m^3)
μ : Viscosity (Pa·s)

Subscripts

exp : Experimental
f : Saturated liquid
fg : Change between liquid and gas phases
g : Saturated vapor
modified Nusselt : Modified Nusselt Eq.
Nusselt : Nusselt
p : Constant pressure
sat : Saturated
t : Thermocouple
tube : Tube
w : Cooling water
wi : Cooling water inlet
wo : Cooling water outlet
wall : Wall

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