Journal of
Mechanical
Science and
Technology

Journal of Mechanical Science and Technology 21 (2007) 804~813

Condensation Heat Transfer Coefficients of HCFC22, R410A, R407C and HFC134a at Various Temperatures on a Plain Horizontal Tube

Ki-Jung Park [b], Dongsoo Jung [a,*]

[a]Department of Mechanical Engineering, Inha University, 253, Yonghyundong, Namgu, Inchon 402-751, Korea
[b]Graduate School, Inha University, 253, Yonghyundong, Namgu, Inchon 402-751, Korea

(Manuscript Received June 21, 2006; Revised March 26, 2007; Accepted March 26, 2007)

Abstract

In this study, external condensation heat transfer coefficients (HTCs) of HCFC22, R410A, R407C, and HFC134a were measured on a smooth horizontal tube at 30, 39, and 50℃ with the wall subcooling of 3-8℃. The results showed that condensation HTCs decreased for all fluids tested with an increase in temperature. This is due mainly to such properties as the saturated liquid density and liquid thermal conductivity. These properties decrease as the temperature increase and accordingly HTCs decrease. The condensation HTCs of R410A are 9.2~19.7% higher than those of HCFC22 while those of R134a are 2.5~10.2% lower than those of HCFC22. Condensation HTCs of R407C, non-azeotropic mixture, are 29.4~34.3% lower than those of HCFC22. Overall, the HTCs of R407C are much lower than those of HCFC22, HFC134a and R410A due to the mass transfer resistance in a diffusion vapor film. Condensation HTCs of HCFC22 and HFC134a are higher than those calculated by Nusselt' s equation by 7.7~11.8% and 4.0~11.1% respectively. On the other hand, HTCs of R407C measured on plain tube, however, are not well predicted by these well-known prediction correlations due to the introduction of mass transfer resistance associated with non-azeotropic mixtures.

Keywords: Condensation heat transfer; Alternative refrigerant; HCFC22; R410A; R407C; HFC134a

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1. Introduction

Chlorofluorocarbon (CFCs) have been 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, have been regulated and eventually phased out since Molina and Rowland (1974) discovered that chorine atoms in CFCs destroy the stratospheric ozone layer in 1974. In 1997, 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 (GECR, 1997).

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 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 (Marto and Nunn, 1981; Bergles, 1985).

HCFC22 has been used predominantly in residential air conditioners. Even though the Montreal

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[*] Corresponding author. Tel.: +82 32 860 7320, Fax.: +82 32 868 1716
E-mail address: dsjung@inha.ac.kr

protocol signed for the complete phase out of HCFC22 by 2020, all European countries have already stopped using HCFC22 and the US will not use HCFC22 in new equipment from 2010. Therefore, HCFC22 will be replaced by certain refrigerants discussed by HCFC22 Alternative Refrigerants Evaluation Program. Unlike CFC11 and CFC12, no pure refrigerant has been identified to replace HCFC22. Binary and ternary mixtures of R410A(50%HFC32/ 50%HFC125) and R407C(23%HFC32/25%HFC125/ 52%HFC134a) have been identified as the promising candidates for HCFC22 and some products charged with these alternatives have been on the market recently (Muir, 1994). R407C is a non-azeotropic refrigerant mixture(NARM) whose gliding temperature difference(GTD) is roughly 6℃. Its vapor pressure is similar to that of HCFC22 and hence it is expected that R407C may be used in existing equipment without major changes. Since it is a NARM, however, fractionation may occur in the case of leak in the system. Also, the heat transfer degradation associated with NARMs may cause performance degradation of heat exchangers when R407C is adopted (ARI, 1992-1997; Cavallini, 1996).

On the other hand, R410A is a near-azeotropic mixture with a GTD of less than 0.2℃. Its vapor pressure is roughly 50% higher than that of HCFC22 and hence the capacity increases significantly with R410A. Due to the high pressure, compressors may need to be redesigned completely and also the heat exchangers need to be optimized to accommodate lower volumetric flow rates associated with the use of R410A. Even though a simple thermodynamic cycle analysis shows that the cycle efficiency of R410A is somewhat lower than that of HCFC22, the actual energy efficiency of R410A is higher than that of HCFC22 due to the improved compressor efficiency and reduced energy losses in some components of the refrigeration system. At present, it seems that R410A will be adopted in the new systems while R407C will be used in the existing systems (ARI, 1992-1997; Cavallini, 1996).

For commercial air-conditioning equipment, HFC-134a was also identified as a possible candidate to replace HCFC22. Hwang et al. (1999) showed that the condensation heat transfer coefficients (HTCs) of HFC134a were 20% higher than those of CFC12. Honda et al. (2002) studied the effect of two- and three-dimensional surface geometry in horizontal tubes using HFC134a.

In 1982, Goto and Fujii (1982) studied the liquid condensate formation for CFC12/CFC114 and CFC-114/CFC11 mixtures at various concentrations while Fujii et al. (1990) studied the condensation heat transfer of binary mixtures of methanol/water, ethanol/water, and methanol/ethanol. In 1986, Hijikata et al. (1986) measured condensation HTCs of CFC114/ CFC113 and CFC11/CFC113 on a flat plate and a finned plate and observed that condensation heat transfer resistance was affected greatly by a diffusion vapor film when the wall subcooling is small while that is much affected by a liquid condensate film when the wall subcooling is large. In 1994, Wang et al. (1994) predicted condensation HTCs of HCFC22/ HFC152a numerically as a function of composition and wall subcooling while in 1996 Signe et al. (1996) reported condensation HTCs of HFC23/HFC134a on a horizontal plain tube and GEWA-SC enhanced tube. These studies all show that the condensation phenomenon for mixtures is quite different from that for pure fluids and HTCs of mixtures are lower than those of pure fluids constituting the mixtures.

As for the tube bundle tests with mixtures, Gabrielii and Vamling (1997) measured overall HTCs on a full scale test plant with a horizontal shell and tube condenser charged with HCFC22 and R407C and showed that overall HTCs with R407C decreased up to 70% as compared to those with HCFC22. Honda et al. (1999a, 1999b) examined the condensation phenomenon of zeotropic mixture of HCFC123/HFC134a flowing downward on a staggered bundle of horizontal low-finned tubes with four fin geometries and showed that HTCs of mixture were considerably lower than those of either HFC134a or HCFC123.

As seen in the literature survey, very few studies have been reported for the condensation heat transfer of refrigerant mixtures on a plain tube. Especially, not much data is available for R407C and R410A on horizontal tubes. In order to apply R407C and R410A successfully in commercial equipment, however, their condensation HTCs need to be measured with their heat transfer characteristics analyzed subsequently. Especially, this is important in the light of global environmental issues requiring high energy efficiency equipment. In order to develop heat transfer correlation, condensation HTCs have to be measured at various temperatures. Therefore, in this study condensation HTCs of HCFC22, R407C, R410A, HFC-134a are measured on a plain horizontal tube at condensation temperatures of 30, 39, 50℃.

2. Experiments

2.1 Overall description of experimental apparatus

Figure 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 the immersion heater of 3.5 kW in the boiler that was 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°C as shown in Fig. 1.

The main test section was made of a 80mm id stainless steel pipe with a 110mm 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 were not well soldered together, then the temperatures would 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.

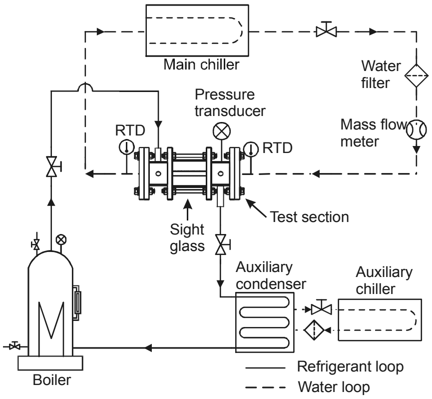

Fig. 1. Schematic diagram of the condensation heat transfer experimental apparatus.

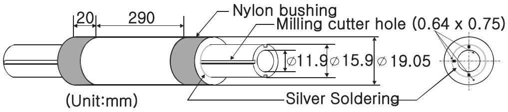

Fig. 2. Detailed description of test tube.

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:

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

where $\dot{m}_w$, Cp_w , T_{wo} , T_{wi} are the mass flow rate, specific heat, 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 a 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°C 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°C 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_g, 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 using an RTD of 0.01°C accuracy. Finally, a pressure transducer of 0.2% accuracy was calibrated and used to measure the pressure of the main test section.

2.3 Experimental procedure and condition

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°C.

The experimental procedure was as follows:

(1) Nitrogen was charged to the refrigerant loop up to 1500 kPa with some halogenated refrigerants to check by an halogen detector if there was any leak.

(2) A vacuum pump was turned on for a few 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 temperature of the cooling water and all variables were recorded under steady-state. The wall subcooling was varied from 3~8°C for a given fluid.

(6) The refrigerant was changed and the same procedures were repeated.

Table 1 show the overall experimental condition of this study.

Table 1. Experimental condition.

Test refrigerants	HCFC22, R410A, R407C, HFC134a
Sat. vapor temp.	30℃, 39℃, 50℃
Flow rate(coolant)	60 g/s
Wall subcooling	3℃~8℃(at intervals of 1℃)
Heat flux	3 kW
Heat transfer area	0.017356 m^2

2.4 Heat transfer coefficients

A local condensation HTC was determined by the following equation.

$$h = \frac{Q}{A(T_g - T_s)} \tag{2}$$

where h, Q, A, T_g, T_s are the HTC(W/m^{2}K), heat transfer rate(W), area(m^2), vapor and surface temperatures(°C) respectively. For the plain tube tested, the nominal area based on the outside diameter was used as the area in equation (2).

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

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

where T_s, T_t, L, r_{tube}, r_t, k_{tube} are the actual surface temperature(°C), measured temperature by a wall thermocouple(°C), 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 compensation term, $(T_s - T_t)$, in equation (3) was very small, typically less than 0.1°C. 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 errors were estimated by the method suggested by Kline and McClintock (1953). In general, the measurement error increased as the wall subcooling decreased. For all tubes, the error was estimated to be 4.9~5.3% and 2.0~2.3% for the wall subcooling of 3°C and 8°C respectively.

As time progresses, fouling(or oxidization) will be developed on the surface of the test tube. This fouling increases the thermal resistance and hence reduces the HTCs of the tube. Therefore, in this study, whenever the tube was changed, the surface was cleansed with acetone and thoroughly dried. Also repetition tests were performed a few times with an interval of one month and it was observed that HTCs were all within 5% of the average value for the same tube and same fluid. These repetition tests were performed for all three test tubes considered in this study.

3. Results and discussion

3.1 Effect of condensation temperatures

Figure 3 shows the condensation HTCs of HCFC22 at three condensation temperatures as a function of wall subcooling. Most of the previous data were obtained at 39℃ but in this study, more data were obtained to see the effect of temperature on condensation HTCs. For HCFC22, HTCs decreased as the condensation temperature increased. With 10℃ increase in temperature, 5% decrease in HTCs was observed at various subcoolings as shown in Fig. 3. In condensation, saturated liquid density(ρ_f), the difference in liquid and vapor densities($\rho_f - \rho_g$), liquid thermal conductivity(k_f) affect HTCs greatly. As shown in Table 2, these properties decrease as the temperature increase. Therefore, condensation HTCs decrease accordingly.

Figures 4~6 show the condensation HTCs of R410A, R407C, and HFC134a at three temperatures with various wall subcoolings. For these alternative fluids, HTCs also decrease as the temperature increases. HTCs of these fluids at 30℃ were 4.0%,

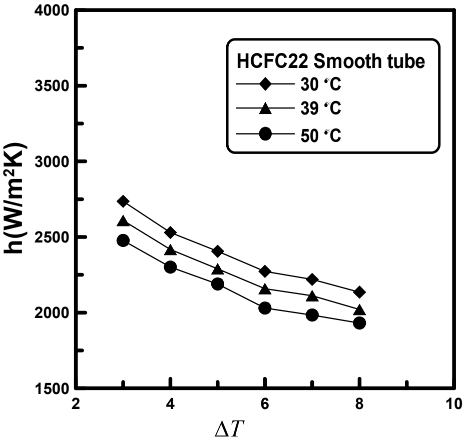

Fig. 3. Condensation heat transfer coefficients of HCFC22 on a smooth tube as a function of wall subcooling.

Table 2. Saturation properties of experimental refrigerants at 30℃, 39℃, 50℃.

Refrigerants	Temperature	P_{sat} (kPa)	ρ_f (kg/m^3)	ρ_g (kg/m^3)	Cp_f (kJ/kg·K)	h_{fg} (kJ/kg)	k_f (W/m·K)	μ_f (Pa·s)
HCFC22	30℃	1192	1171	50.70	1.281	177.68	0.081	156.7
	39℃	1497	1133	64.47	1.332	167.78	0.077	141.1
	50℃	1943	1082	85.95	1.419	154.20	0.072	123.1
R407C	30℃	1175	1116	50.80	1.563	176.90	0.085	143.2
	39℃	1499	1073	66.15	1.641	165.30	0.080	126.9
	50℃	1985	1014	91.09	1.779	148.80	0.018	108.3
R410A	30℃	1877	1035	76.11	1.750	178.40	0.094	133.0
	39℃	2352	985	99.54	1.897	161.80	0.088	99.2
	50℃	3053	912	140.00	2.227	137.20	0.080	82.8
HFC134a	30℃	770	1187	37.54	1.446	173.09	0.079	185.8
	39℃	989	1151	48.68	1.493	164.09	0.075	165.6
	50℃	1318	1102	66.27	1.566	151.80	0.070	143.1

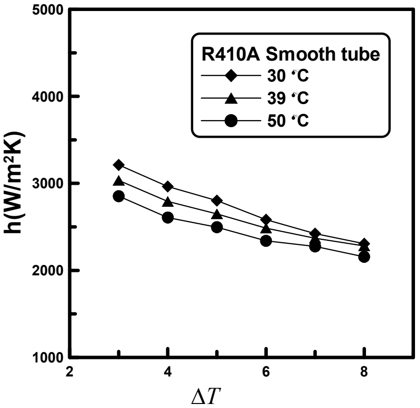

Fig. 4. Condensation heat transfer coefficients of R410A on a smooth tube as a function of wall subcooling.

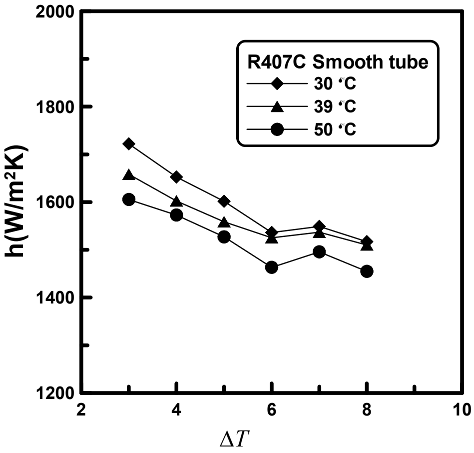

Fig. 5. Condensation heat transfer coefficients of R407C on a smooth tube as a function of wall subcooling.

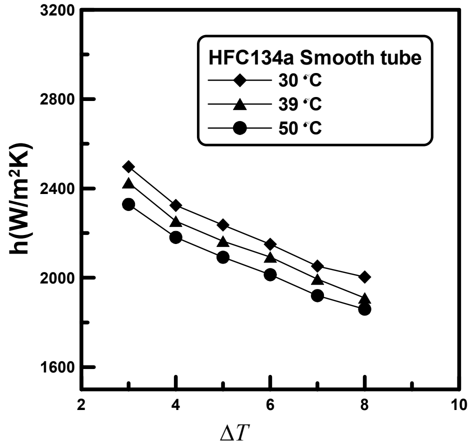

Fig. 6. Condensation heat transfer coefficients of HFC134a on a smooth tube as a function of wall subcooling.

1.9%, 3.2% higher than those measured at 39℃. On the other hand, HTCs of these fluids at 50℃ were 6.0%, 3.0%, 3.6% lower than those measured at 39℃.

3.2 Effect of mixtures

The fact that HTCs of R407C are much lower than those of other fluids is due to its non-azeotropy causing a substantial effect on condensation phenomenon. Figure 7 illustrates temperature and pressure distribution near the wall including a liquid/vapor interface when a non-azeotropic binary mixture condenses. For this mixture the less volatile component whose boiling point is higher begins to condense before the more volatile and thus a diffusion vapor film is formed(Region B in Fig. 7). Therefore, in region B the concentration of the less volatile component decreases toward the wall and so does its partial pressure, P_{lv}. Consequently, the partial pressure of the more volatile component whose boiling point is lower, P_{mv} , increases toward the wall in region B to maintain the same total pressure. Due to this unique phenomenon which is not present in pure fluids, an additional vapor diffusion film is formed between the liquid layer and bulk vapor with a concentration gradient developed within it. Ultimately, this vapor diffusion film plays a role as a thermal resistance retarding the heat transfer as explained by Hijikata et al. (1986), Colburn and Drew (1937), and Collier and Thome (1994). This would be true also of multi-component mixtures such as R407C. From this one can see that the condensation of non-azeotropic mixtures such as R407C is affected not only by thermal resistance due to liquid film, region A in Fig. 7, but also by thermal resistance due to diffusion vapor film, region B in Fig. 7.

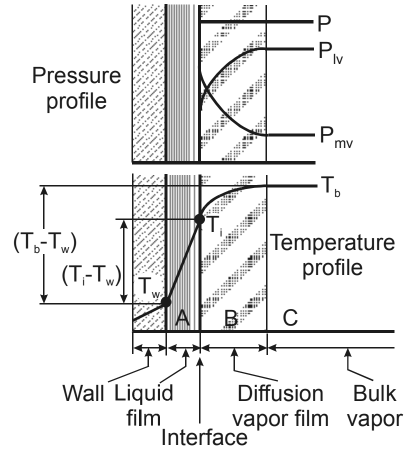

Fig. 7. Temperature and pressure distribution during the condensation of a binary mixture.

Due to the formation of diffusion vapor film, the saturation temperature at the interface is reduced from T_b to T_i and consequently the temperature driving potential for the conduction in liquid film decreases from ($T_b - T_w$) to ($T_i - T_w$) resulting in a reduction in HTCs. The reduction in HTCs is determined by the magnitude of the concentration gradient in the diffusion vapor film for which the GTD is a good indicator. As the non-azeotropy increases, so does the GTD. As mentioned before, the GTD of R410A is less than 0.2℃ and hence R410A is expected to behave as if it were a pure fluid. The GTD of R407C, however, is close to 6℃ and hence HTCs of R407C

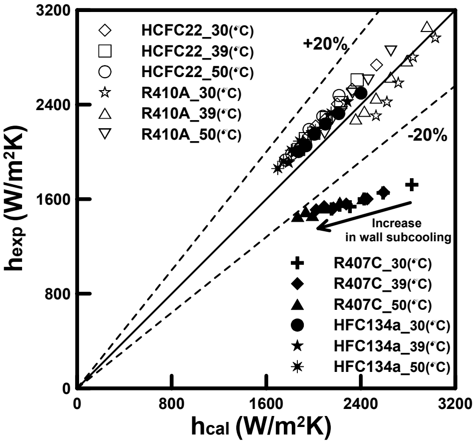

Fig. 8. Comparison of the measured HTCs with the calculated ones on a smooth tube.

are expected to suffer from the mass transfer resistance and test results confirm this.

3.3 Comparison with theoretical prediction

Plain tube data obtained in this study are compared against a prediction equation to check their validity for pure fluids of HCFC22 and HFC134a and near-azeotrope R410A as well as to see the heat transfer reduction associated with non-azeotropic mixture R407C. For this purpose, theoretical HTCs were calculated using Nusselt's condensation heat transfer correlation for smooth horizontal tube. Nusselt's correlation, Equation (4), was derived under the assumption that the condensing vapor is pure fluid and the flow is laminar and no shear stress occurs at the liquid/vapor interface with no variation of liquid properties in the condensate.

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

Figrue 8 shows the comparison between the predicted and measured condensation HTCs of 4 fluids at 30℃, 39℃, 50℃. For reference, Table 2 lists some of the physical properties of the refrigerants tested at 30℃, 39℃, 50℃ (REFPROP 6.0, 1998).

Measured HTCs of HCFC22 and HFC134a are 7.7~11.8% and 4.0~11.1% higher than the prediction respectively while those of R410A are similar to the prediction. For pure HCFC22 and near-azeotrope R410A, the difference between the measured data and prediction is caused mainly by the assumption used in deriving the Nusselt's equation that the condensate film is laminar (Holman, 1992). In fact, however, the condensate film is wavy even at low Reynolds numbers causing a convection current and also reducing the film thickness, which was observed through the sight glass (Jung et al., 1999). The fact that measured HTCs on a plain tube are up to 12% higher than those calculated by the Nusselt's equation was also observed by other researchers (Jung et al., 1999; Wanniarachchi et al., 1986; Marto et al., 1990; Kim et al., 1995). This comparison indirectly confirms that the present data for the plain tube are reliable.

Measured HTCs of R407C were 15.6~28.9% lower than those by the prediction. This kind of reduction in HTCs of non-azeotropic mixtures, as shown with the present data for R407C, was also observed by Signe et al. (1996) who the measured condensation HTCs of HFC23/HFC134a mixture on a plain tube at 40℃ with a wall subcooling of 1~20℃. The GTDs of this mixture are 9.5 and 14.0℃ at HFC23 mass fractions of 0.05 and 0.08 respectively and condensation HTCs were reduced by 7~35% and 28~55% at these compositions respectively as compared to those of HFC134a. They also showed that the reduction in HTCs increased with an increase in GTD of the mixture.

For R407C, measured HTCs approach the prediction as the wall subcooling increases as indicated in Fig. 8. This is due to the fact that the rate of condensation of the more volatile component increases rapidly with an increase in subcooling and hence the concentration difference between the bulk vapor and interface decreases. In other words, the vapor diffusion film plays no longer an important role as a thermal barrier. Hence under this situation, the thermal resistance due to the liquid layer plays a dominant role as with pure fluids. Thus, measured data for NARMs will approach the prediction developed for pure fluids as the wall subcooling increases. This phenomenon was also observed by Signe et al. (1996).

3.4 Relative heat transfer performance among alternatives

Figure 9 shows the relative heat transfer improvement of HCFC22 alternatives. HTCs of R410A are 9.2~19.7% higher than those of HCFC22 at the same temperatures. HTCs of R407C and HFC134a are 29.4~34.3% and 2.5~10.2% lower than those of HCF-

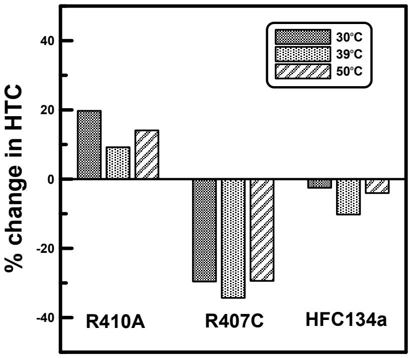

Fig. 9. Deviation of condensation heat transfer coefficients at various temperatures as compared HCFC22.

C22. Therefore, from the viewpoint of condensation heat transfer, R410A showed the best performance.

4. Conclusions

In this study, condensation HTCs of HCFC22, R410A, R407C, and HFC134a were measured on a smooth horizontal tube at 30, 39, and 50℃. Nusselt's correlation was used to verify the measured data for pure fluids and to see the degree of discrepancy for mixtures. Based on this research, following conclusions can be drawn.

(1) As the condensation temperature increased, condensation HTCs decreased for all fluids tested. This is due mainly to such properties as the saturated liquid density(ρ_f), the difference in liquid and vapor densities($\rho_f - \rho_g$), liquid thermal conductivity(k_f). These properties decrease as the temperature increase and accordingly HTCs decrease.

(2) Condensation HTCs of R410A, near-azeotropic mixture, are 9.2~19.7% higher than those of HCFC22. R410A may become an environmentally friendly solution to HCFC22 from the standpoint of condensation heat transfer showing an improvement over HCFC22.

(3) Condensation HTCs of R407C, non-azeotropic mixture, are 29.4~34.3% lower than those of HCFC22. Overall, the HTCs of R407C are much lower than those of HCFC22, HFC134a and R410A due to the mass transfer resistance in a diffusion vapor film.

(4) Condensation HTCs of R134a are 2.5~10.2% lower than those of HCFC22.

(5) Condensation HTCs of HCFC22 and HFC134a are higher than those calculated by Nusselt's equation by 7.7~11.8% and 4.0~11.1% respectively. On the other hand, HTCs of R407C measured on plain tube, however, are not well predicted by these well-known prediction correlations due to the intro-duction of mass transfer resistance associated with non-azeotropic mixtures. A more elaborate model ac-counting for the liquid/vapor interfacial phenomenon might be needed to predict the condensation HTCs of non-azeotropic refrigerant mixtures.

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}K]
h_{fg} : Heat of evaporation [kJ/kg]
k : Thermal conductivity [W/m·K]
L : Length [m]
$\dot{m}$: Mass flux [kg/s]
P : Pressure [kPa]
Q : Heat transfer rate [W]
r : Radius [m]
T : Temperature [^{o}C or K]
ΔT : Temperature difference [^{o}C or K]

Greek symbols

μ : Dynamic viscosity [Pa · s]
ρ : Density [kg/m^3]

Subscripts

b : Bulk vapor
cal : Calculated
exp : Experimental
f : Saturated liquid phase
g : Saturated vapor phase
i : Interface
lv : Less volatile component
mv : More volatile component
Nu : Nusselt
o : Nominal outside
p : Constant pressure
s : Surface
sat : Saturated
t : Thermocouple
tube : Tube
w : Cooling water
wi : Cooling water inlet
wo : Cooling water outlet

Acknowledgement

This work was supported by INHA UNIVERSITY Research Grant.

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