

External Condensation Heat Transfer Coefficients of Refrigerant Mixtures on a Smooth Tube

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Key words: Condensation heat transfer, Binary refrigerant mixture, Nonazeotropic mixture, HFC32, HFC134a, HCFC123

Abstract

In this study, condensation heat transfer coefficients (HTCs) of nonazeotropic refrigerant mixtures of HFC32/HFC134a and HFC134a/HCFC123 at various compositions were measured on a horizontal smooth tube. All data were taken at the vapor temperature of 39°C with a wall subcooling of 3~8 K. Test results showed that HTCs of tested mixtures were 11.0~85.0% lower than the ideal values calculated by the mass fraction weighting of the HTCs of the pure components. Thermal resistance due to the diffusion vapor film was partly responsible for the significant reduction of HTCs with these nonazeotropic mixtures. The measured data were compared against the predicted ones by Colburn and Drew's film model and a good agreement was observed within a deviation of 15%.

Nomenclature

<p>a : constant, $n_T C_{p_m}/h_g$</p> <p>A : heat transfer area [m^2]</p> <p>C_p : specific heat [$kJ/kg \cdot K$]</p> <p>D : tube diameter [m]</p> <p>f_{pi} : number of fins per inch</p> <p>GTD : gliding temperature difference [K]</p> <p>h : heat transfer coefficient [$W/m^2 \cdot K$] or enthalpy [kJ/kg]</p>	<p>h_g : gas film heat transfer coefficient [$W/m^2 \cdot K$]</p> <p>k : thermal conductivity [$W/m \cdot K$]</p> <p>L : tube length [m]</p> <p>\dot{m} : mass flow rate [kg/s]</p> <p>\dot{n} : condensation rate per unit area [$kg/m^2 \cdot s$]</p> <p>q : heat flux [W/m^2]</p> <p>Q : heat transfer rate [W]</p> <p>T : temperature [K]</p> <p>ΔT : temperature difference [K]</p> <p>x : liquid composition of more volatile component at interface</p> <p>y : vapor composition of more volatile component at interface</p>
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z : mass transfer rate, \dot{n}_i/\dot{n}_T

Subscripts

c : condensate

i : vapor/liquid interface

L : latent heat of condensation at vapor/liquid interface

m : average value at vapor/liquid interface

sat : saturated vapor

T : total amount of condensation

$tube$: heat transfer tube

v : sensible heat through vapor diffusion film

w : cooling water

$wall$: wall of heat transfer tube

wo : cooling water outlet

wi : cooling water inlet

1. Introduction

Condensers used in large refrigeration equipment, power plants, and chemical plants are often of the shell and tube type in which the vapor condenses on the outer surface of tube banks while the coolant flows through the inner passages of the tube banks.

For chillers, horizontal heat transfer tubes have been employed as the tube banks of the condenser with CFCs as refrigerants that have excellent thermodynamic, physical and chemical properties. These CFCs, however, have been phased out all over the world based upon the Montreal protocol.⁽¹⁾ Therefore these days, HCFCs and HFCs are used in conventional centrifugal chillers.

Colburn and Drew⁽²⁾ formulated basic equations for the heat and mass transfer of binary mixtures while Bell and Ghaly⁽³⁾ and Price and Bell⁽⁴⁾ proposed a model for the design of actual condensers with multi-component mixtures. Webb and Taylor,⁽⁵⁾ Webb⁽⁶⁾ predicted the composition and temperature at the vapor/

liquid interface and Goto and Fujii⁽⁷⁾ measured heat transfer coefficients (HTCs) of CFC12/CFC114 and CFC114/CFC11 mixtures on a horizontal tube and also examined the effect of air on the heat transfer. Hijikata et al.⁽⁸⁾ studied the condensation heat transfer characteristics of CFC113/CFC11 and CFC113/CFC114 mixtures on plain and finned tubes both experimentally and theoretically while Fujii et al.⁽⁹⁾ measured the HTCs of methanol/water, ethanol/water, and methanol/ethanol mixtures. Fujii et al.⁽¹⁰⁾ also divided the condensation phenomenon into 5 different types which are dependent upon the composition, vapor pressure, and heat transfer rate. On the other hand, Wang et al.⁽¹¹⁾ determined the HTC and interface temperature for HCFC22/HFC152a mixture by both experiments and numerical analysis and observed that the HTC increases with wall subcooling. Recently, Signe et al.⁽¹²⁾ measured HTCs of HFC134a/HFC23 and concluded that the vapor diffusion film is a very important factor in condensation heat transfer of mixtures. Gabrielli and Vamling⁽¹³⁾ studied the change of HTCs with HFC407C, HFC404A, HFC410B mixtures but could not predict the reduction of HTCs with nonazeotropic mixtures. On the other hand, Hwang et al.⁽¹⁴⁾ measured HTCs of HFC410A and HFC407C and concluded that they are acceptable as HCFC22 alternatives from the viewpoint of heat transfer.

As shown above, external condensation heat transfer data from horizontal tubes available in the literature are mainly of water, ethanol, methanol, n-pentane and CFCs and experimental data of pure and mixed HCFCs and HFCs are very rare. Hence in this study, HTCs of HFC32/HFC134a and HFC134a/HCFC123 mixture will be measured at different compositions and the results will be compared to the film model of Colburn and Drew⁽²⁾ in an attempt to examine the condensation heat transfer characteristics of nonazeotropic mixtures.

2. Experiments

2.1 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. Thus generated vapor 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 are capable of controlling the temperature within 0.1°C as shown in Figure 1.

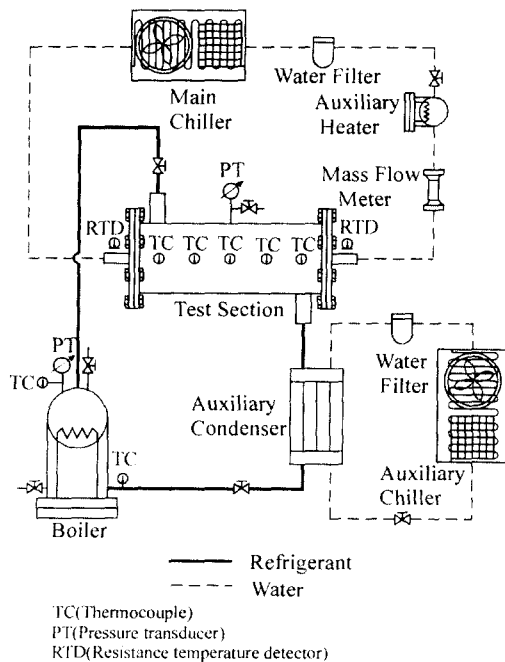


Fig. 1 Schematic diagram of the apparatus.

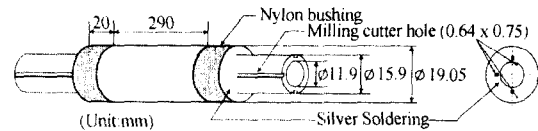


Fig. 2 Detailed description of test tube.

As illustrated in Figure 2, the main test section was made of a 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 (Momoner Cast Nylon, $20\text{ mm} \times 1.5\text{ mm}$) were tightly fastened on the tube so that heat transfer occurs only on the test tube. Finally, to reduce the heat loss to the surroundings, the whole apparatus including the test section was well insulated.

2.2 Experimental procedure

Condensation heat transfer is affected by the non-condensable gases present in the system and hence degassing is very important for the precise measurements of the condensation HTC. 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 tests 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 2,500 kPa with some halogenated refrigerants to check 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 39°C 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 varied from 3~8°C for a given fluid.

(6) Either the refrigerant or the test tube was changed and the same procedures of (1)~(5) were repeated.

More detailed information concerning the experiments is available elsewhere.⁽¹⁶⁾

2.3 Measurements

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

$$Q = hA(T_{sat} - T_{wall}) \quad (1)$$

$$Q = \dot{m}_w C_{p_w} (T_{wo} - T_{wi}) \quad (2)$$

Where \dot{m}_{cw} , $C_{p_{cw}}$, T_{cwo} , T_{cwi} are the mass flow rate, specific heat, and temperatures at the outlet and inlet of the tube respectively.

From these equations, the condensation HTC is calculated as follows:

$$h = \frac{\dot{m}_w C_{p_w} (T_{wo} - T_{wi})}{A(T_{sat} - T_{wall})} \quad (3)$$

The surface temperature, T_w , was determined by modifying the temperature read by a thermocouple located roughly 1.0 mm beneath the surface, T_t , via a 1-D steady-state conduction

equation to account for the temperature drop across the tube wall as follows:

$$T_w = T_t + \frac{(Q/A)}{2\pi L} \left[\frac{\ln(r_o/r_t)}{k_{tube}} \right] \quad (4)$$

Where T_t , L , r_o , r_t , k_{tube} are the measured temperature by a 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/mK) respectively.

All tubes tested in this study were made of copper and hence the temperature compensation term in the right hand side of equation (4) was small, typically less than 0.1°C and did not have any significant effect on the HTC's. Finally, the arithmetic average of the three local HTC's at the top, side, and bottom of the tube was taken for the data presented later.

The measurement errors were estimated by the method suggested by Kline and McClintock.⁽¹⁷⁾ In general, the measurement error increased as the wall subcooling decreased. For all tubes, the error was estimated to be 3.0~8.0%.

Finally, as time progresses, fouling (or oxidation) will be developed on the surface of the test tube. This fouling effect increases the thermal resistance and hence reduces the HTC's of the tube. Therefore, in this study, whenever the tube was changed, the surface was cleaned with acetone and thoroughly dried. Also repetition tests were performed a few times with an interval of one month and it was observed that HTC's were all within 5% for the same tube and same fluid. This was done for all three test tubes considered in this study.

3. Results and discussion

3.1 Experimental data on a plain tube

Table 1 lists the compositions and gliding

Table 1 Compositions and GTDs of tested nonazeotropic mixtures at $P=300$ kPa

	R32/R134a		R134a/R123	
	R32	GTD (K)	R134a	GTD (K)
Case 1	0.00	0.00	0.00	0.00
Case 2	0.39	5.04	0.08	9.76
Case 3	0.64	3.41	0.29	26.20
Case 4	0.84	1.83	0.55	22.94
Case 5	1.00	0.00	0.74	14.58
Case 6	-	-	0.86	7.64
Case 7	-	-	1.00	0.00

temperature differences (GTDs) at those compositions for two binary mixtures studied. Fig. 3 and Fig. 4 show the GTDs and HTC's of

HFC32/HFC134a on a plain tube as a function of composition. HTC's of HFC32/HFC134a mixture were lower than the ideal values calculated by mass fraction weighting of pure components' HTC's (dotted line in Fig. 4) by 15.7~20.5%, 21.0~24.3%, 11.0~16.9% for Cases 2, 3, and 4, respectively.

On the other hand, Fig. 5 and Fig. 6 show the GTDs and HTC's of HFC134a/HCFC123 on a plain tube as a function of composition. HTC's of HFC134a/HCFC123 mixture were lower than the ideal values (dotted line in Fig. 6) by 18.2~23.3%, 50.0~55.1%, 71.1~78.7%, 70.2~85.0%, 47.7~71.9% for Cases 2 to 6, respectively.

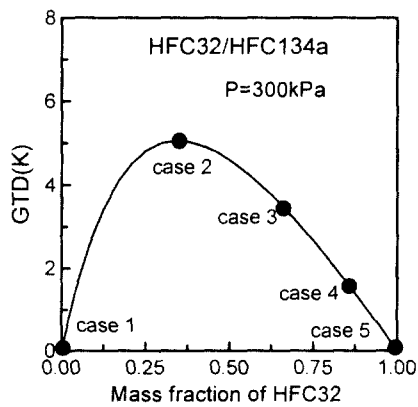


Fig. 3 GTDs of HFC32/HFC134a.

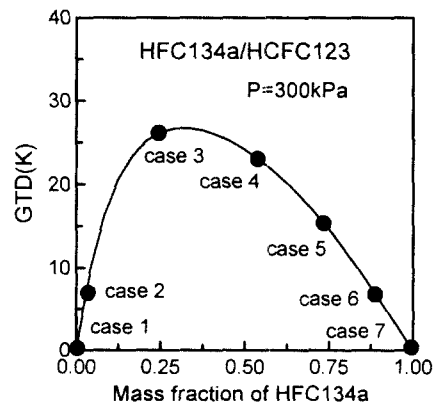


Fig. 5 GTDs of HFC134a/HCFC123.

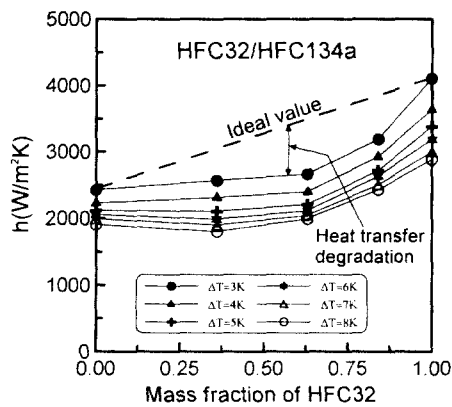


Fig. 4 Condensation HTC's of HFC32/HFC134a on a smooth tube.

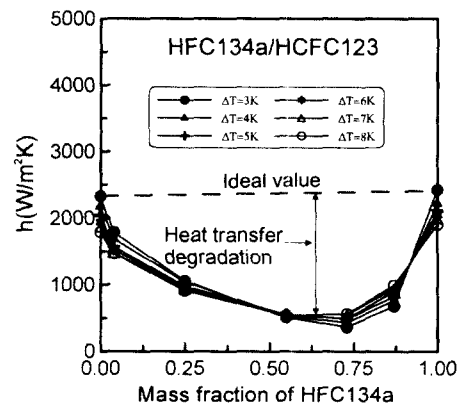


Fig. 6 Condensation HTC's of HFC134a/HCFC123 on a smooth tube.

The significant reduction in HTC's associated with these nonazeotropic mixtures is due to the fact that the condensation phenomenon of nonazeotropic mixtures is different from that of pure fluids. During the condensation of nonazeotropic mixtures, less volatile component begins to condense first at the interface rather than the more volatile component. Therefore, the partial pressure of the less volatile component decreases while that of the more volatile component increases to maintain equilibrium with the bulk vapor. Consequently, a diffusion vapor film is formed due to the composition difference between the interface and bulk vapor. Thus, the condensation heat transfer with nonazeotropic mixtures is influenced by the resistance due to liquid film as well as the resistance due to diffusion vapor film. Because of the increase in resistance to heat transfer, HTC's of nonazeotropic mixtures are degraded. This unique phenomenon is also observed by Hijikata et al.,⁽⁸⁾ Wang et al.,⁽¹¹⁾ Signe et al.⁽¹²⁾

In pool boiling heat transfer of nonazeotropic mixtures, the largest reduction in HTC's usually occurs at the composition where the GTD is the largest.⁽¹⁸⁾ This, however, is not the case in the condensation of nonazeotropic mixtures as shown in Fig. 4 and Fig. 6. This difference is thought to be caused by the effect of the diffusion vapor film. As shown in Fig. 7 and Fig. 8, condensation HTC's of HFC32/HFC134a whose GTDs are relatively small decrease with the wall subcooling as with the results of pure fluids. On the other hand, condensation HTC's of HFC134a/HCFC123 whose GTDs are relatively large, HTC's increase with the wall subcooling when the composition of HFC134a is greater than 55%. The fundamental condensation phenomenon of mixtures needs to be examined to explain this kind of difference. Therefore, the relationship between the film model of Colburn and Drew and experimental data is examined in the following analysis.

3.2 Comparison of Colburn and Drew's film model and experimental data

3.2.1 Theoretical background

The theoretical analysis of Colburn and Drew⁽²⁾ is given as follows: During the condensation of mixtures, both heat and mass transfers occur simultaneously. The heat flux carried away by the coolant, q_w is assumed to be composed of the heat flux by the cooling of the condensate q_c and the heat flux due to the latent heat of condensation q_L and finally the heat flux due to the sensible heat from the bulk vapor to the vapor/liquid interface q_v . In general, q_c and q_v are assumed to be small as compared to q_L and under this situation and the condensation heat transfer coefficient is calculated theoretically as follows:

The heat flux due to the latent heat of condensation at the interface, q_L , is obtained by condensation rate of each component, \dot{n}_i multiplied by its latent heat, h_{i-fg} as in equation (5).

$$q_L = \sum_{i=1}^n \dot{n}_i h_{i-fg} \quad (5)$$

Equation (5) can be rewritten using the total condensation rate, \dot{n}_T and average latent heat, h_{m-fg} as in equation (6).

$$q_L = \dot{n}_T h_{m-fg} \quad (6)$$

$$\dot{n}_T = \sum_{i=1}^n \dot{n}_i \quad (7)$$

$$h_{m-fg} = \sum_{i=1}^n z_i h_{i-fg} \quad (8)$$

It is also assumed that the total heat flux due to the sensible heat from the bulk vapor to the interface, q_v is composed of the heat transferred through the diffusion vapor film

and sensible heat transferred due to the temperature difference between the bulk vapor and interface and it can be summarized as in equation (9).

$$q_v = h_g (T_{sat} - T_i) \frac{a}{1 - e^{-a}} \quad (9)$$

$$a = \frac{\dot{n}_T C p_m}{h_g} \quad (10)$$

Since q_c is relatively small as compared to q_L , it can be neglected and thus the heat flux absorbed by the coolant, q_w , can be expressed as the sum of equations (6) and (9) due to the energy balance.

$$q_w = h_g (T_{sat} - T_i) \frac{a}{1 - e^{-a}} + \dot{n}_T h_{m-fg} \quad (11)$$

In order to apply equation (11), the properties at the interface are to be known. First of all, the vapor composition of the more volatile component at the interface between the condensate and the vapor diffusion film, y_i , is determined from the vapor-liquid phase equilibrium data when the liquid composition of the more volatile component at the interface is given as x_i . For this, the interface temperature T_i becomes

the boiling point corresponding to the liquid composition x_i . With this information the properties at the interface can be determined and consequently q_w also can be known. Then the equilibrium point can be determined using equation (11) by iteration and through this process T_i can be determined. Finally, the condensation HTC is calculated as follows:

$$h = \frac{q_w}{(T_i - T_{wall})} \quad (12)$$

3.2.2 Comparison between the theoretical and experimental data

Fig. 7 and Fig. 8 show the experimental data and theoretical data calculated by the film model of Colburn and Drew for HFC32/HFC134a and HFC134a/HCFC123 mixtures. One can see a good agreement between them.

For HFC32/HFC134a mixture, the predicted data were higher than the measured data by 7.0~13.8%, 6.0~11.1%, 3.1~6.6% for Cases 2 to 4 respectively while for HFC134a/HCFC123 mixture, the predicted ones were higher than the measured ones by 2.0~3.9%, 9.7~11.0%, 5.7~8.8%, 2.8~10.7%, 3.7~7.2% for Cases 2 to 6 respectively. Altogether, the predicted data agree with the experimental ones within 15%

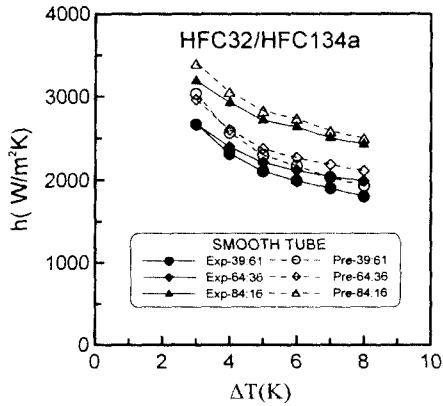


Fig. 7 Comparison of measured HTCs with calculated ones by Colburn and Drew's film model for HFC32/HFC134a.

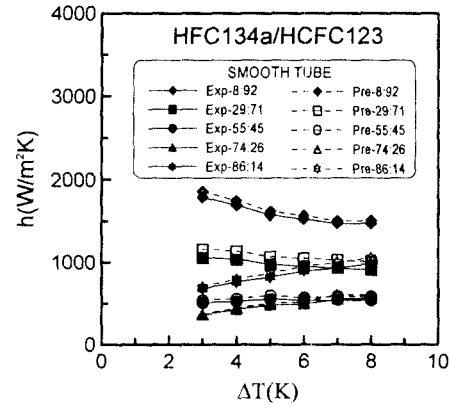


Fig. 8 Comparison of measured HTCs with calculated ones by Colburn and Drew's film model for HFC134a/HCFC123.

deviation. Also the decrease in HTC's for HFC134a/HCF123 mixture with the wall sub-cooling when the HFC134a composition is higher than 55% is shown well with the prediction based upon the film model of Colburn and Drew.

This unique phenomenon seems to occur for mixtures with large GTDs for which the heat transfer is more influenced by the diffusion vapor film. For highly nonazeotropic mixtures, the thermal resistance decreases significantly with the increase in wall subcooling, which results in an increase in heat transfer as shown in Fig. 8.

4. Conclusions

In this study, external condensation HTC's of HFC32/HFC134a and HFC134a/HCF123 mixtures were measured at various composition on a smooth tube and following conclusions were drawn.

(1) For HFC32/HFC134a mixture, the HTC's were 11.0~24.3% lower than the ideal values while for HFC134a/HCF123 mixture, the HTC's were 18.2~85.0% lower than the ideal values. Unlike the pool boiling HTC's, however, the largest reduction in HTC's as compared to the ideal values did not occur at the composition where the GTD is the largest.

(2) HFC134a/HCF123 mixture has the larger thermal resistance than HFC32/HFC134a mixture due to its large GTDs which exert higher mass transfer resistance.

(3) For pure fluids, the HTC's decrease with wall subcooling since the thermal resistance of the condensate film increases. For highly non-azeotropic mixture such as HFC134a/HCF123, however, the HTC's increase with wall subcooling at some compositions since the thermal resistance due to the diffusion vapor film decreases considerably with an increase in wall subcooling.

(4) Colburn and Drew's film model predicted

the data well within a deviation of 15% for all mixture data.

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