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IN-TUBE HEAT TRANSFER CHARACTERISTICS OF REFRIGERANT MIXTURES OF HFC-32/134a AND HFC-32/125/134a

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ABSTRACT

The evaporative and condensational heat transfer and pressure drop characteristics for the refrigerant mixtures of HFC-32/HFC-134a and HFC-32/HFC-125/HFC-134a inside a horizontal tube have been studied experimentally. The experiments were performed for an inner grooved tube utilized in the actual heat exchangers of air-conditioning machines for residential use. The compositions of HFC-32 and HFC-125 in the binary and ternary refrigerant mixtures were varied to examine the effect of the mixture composition on heat transfer characteristics. The evaporation heat transfer coefficients for HFC-32/HFC-134a (30/70wt.%) and HFC-32/HFC-125/HFC-134a (23/25/52wt.%) became lower compared with that for a pure refrigerant HCFC-22 at the same mass flux. Similarly, the condensation heat transfer results for both the refrigerant mixtures were found to be lower than that for HCFC-22.

NOMENCLATURE

A	=	surface area, m^2
C	=	composition of refrigerant mixture, wt.%
ΔP	=	pressure drop across tube, kPa/m
ΔT	=	temperature difference, K
d	=	tube diameter, mm
f	=	fin height, mm
G	=	refrigerant mass flux, kg/m^2s
h	=	heat transfer coefficient, kW/m^2K
K	=	overall heat transfer coefficient, kW/m^2K
N	=	number of fins inside tube
Q	=	heat quantity, kW
Rm	=	thermal resistance at inner tube, m^2K/kW
t	=	tube wall thickness, mm

Greeks

θ	=	twisted angle of fins, degree

Subscripts

c	=	condensation
e	=	evaporation
i	=	inside
o	=	outside
r	=	refrigerant
w	=	water

1 INTRODUCTION

The international agreement of the Montreal Protocol made at the Copenhagen Conference in November 1992 has dictated a new schedule that HCFC-22 was to be totally phased out by the year 2030. Concerning the HCFC-22 phasing-out regulation, the zero ozone depleting refrigerants such as azeotropic mixture of HFC-32/HFC-125, a non-azeotropic mixture of HFC-32/HFC-134a and a ternary mixture of HFC-32/HFC-125/HFC-134a are proposed to be the potential direct replacement for HCFC-22 in air-conditioning machines currently running on HCFC-22.

Evaporation and condensation heat transfer characteristics inside heat transfer tubes of heat exchangers due to the use of the alternative refrigerants has been a subject of fundamental importance to evaluate the heat exchanger performance in air-conditioning machines. From the view point of the practical importance to design heat exchangers installed in air-conditioning machines, the authors [1, 2, 3] have undertaken the experiments to investigate the evaporation and condensation heat transfer characteristics inside a horizontal tube. The authors [1] reported the heat transfer and pressure drop data for pure HFC-134a inside the horizontal smooth and inner grooved tubes. Furthermore, the authors [3] presented that the heat transfer coefficients for the non-azeotropic refrigerant mixture of HFC-32/HFC-134a inside a smooth tube fell below those for HCFC-22. Although the study on the evaporation and condensation heat transfer for the binary refrigerant mixture of HFC-32/134a has been investigated by the authors, there is a lack of information about the evaporation and condensation heat transfer for the ternary refrigerant mixture of HFC-32/125/134a.

The purpose of the present study is to examine experimentally the evaporation and condensation heat transfer characteristics of the refrigerant mixtures of HFC-32/134a and HFC-32/125/134a inside an inner grooved tube, and to compare the heat transfer coefficients for the refrigerant mixtures with those for HCFC-22. The experiments were conducted under the conditions corresponding to the practical conditions of the air-cooled heat exchangers used for 2.5kW air-conditioning machines. In order to obtain the more practical data for designing the heat exchangers, the inner grooved tube with an outer diameter of 7.0mm was employed as the heat transfer tube in the present study.

2 EXPERIMENTS

2.1 Experimental apparatus

A schematic diagram of the experimental apparatus used to determine both evaporation and condensation heat transfer coefficients in a single horizontal tube for the alternative refrigerants is shown in Figure 1. The system is mainly made up of a refrigerant loop and a water loop. A special feature incorporated in the apparatus enabled both the evaporation and condensation experiments. Refrigerant flow directions in both the experiments are indicated by arrows in the diagram.

The main components of the refrigerant loop in the apparatus consisted of a test section, a pump for circulating the refrigerant into the refrigerant loop, a flowmeter, a heating section and two heat exchangers. In place of a compressor, the circulating pump was used to examine accurately the evaporation and condensation heat transfer characteristics of refrigerants without the lubricant. To control the quality of the refrigerant at the inlet of the test section, the heating section was located just before the test section. The pressure drop of the refrigerant across the test section was measured with a differential pressure transducer connected to the pressure taps at the inlet and outlet of the test section. The main heat exchanger was installed in the refrigerant loop to regulate the pressure of the refrigerant. In addition, the auxiliary heat exchanger was used to completely liquefy the refrigerant entering the circulating pump.

The water loop was constructed to supply water to the annulus side of the test section for the purpose of heating or cooling the refrigerant flowing in the tube. To measure the inlet and outlet temperatures of the water and the refrigerant in the test section, Pt resistance sensors were inserted at both the inlet and outlet of the refrigerant and water sides of the test section.

Details of the test section are illustrated in Figure 2. The test section consisted of an inner copper tube and an outer stainless steel tube. In the present study, the inner grooved tube, whose dimensions are shown in Figure 3, was employed as the heat transfer tube in order to obtain the heat

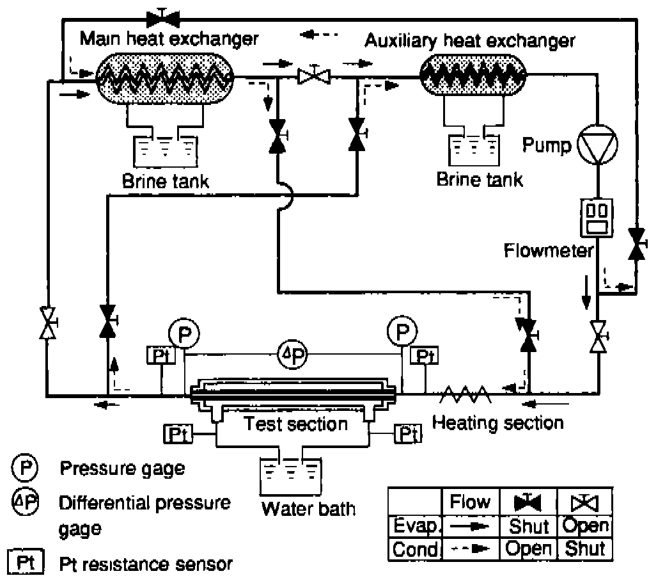

Figure 1 Schematic diagram of experimental apparatus

transfer data available for designing the actual heat exchangers. The test section and the refrigerant and water loops were well insulated thermally by glass wool.

2.2 Experimental Procedure

Prior to the initiation of each data run, noncondensables were removed from the refrigerant loop by a vacuum pump. For evaporation heat transfer experiments, the quality at the inlet of the test section and the superheat at the outlet were maintained at the desired values listed in Table 1. Similarly, the superheat at the inlet of the test section and the subcooling at the outlet during the condensation heat transfer experiments were set at the desired values. In both the experiments, the refrigerant mass flux ranged from 90 to 520 kg/m^{2}s. The values of mass flux selected in the present study corresponds to the amount of refrigerant mass flux among each tube in one pass of the heat exchanger for a 2.5 kW air-conditioning unit.

In the present experiments, a binary blend of HFC-32/134a and a ternary blend of HFC-32/125/134a were prepared as the candidates for HCFC-22. The data runs for HCFC-22 were carried out to compare with the results for the refrigerant blends at similar conditions. During two different experiments on evaporation and condensation, all of the measured values of the Pt resistance sensor, flowmeter and pressure transducer were taken by a data logger when a steady state condition was reached. For each data run, the overall composition of the refrigerant mixture though the test section was checked with gas chromatography.

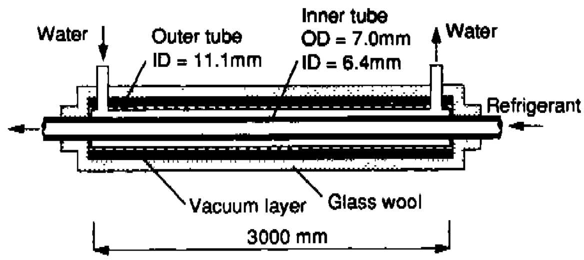

Figure 2 Details of test section

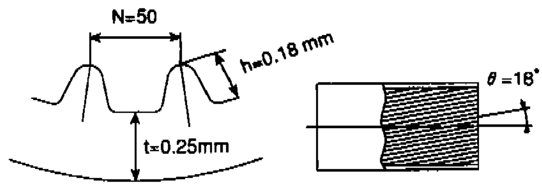

Figure 3 Geometrical parameters of test tube

Table 1 Experimental conditions

	Evaporation	Condensation
Saturation temperature	5 ℃	50 ℃
Inlet condition of test section	Quality = 0.2	SH = 45 ℃
Outlet condition of test section	SH = 5 ℃	SC = 5 ℃
Refrigarant mass flux	90 ~ 520 kg / m^{2}s	

3 DATA REDUCTION

The starting point of the data-reduction procedure is the definition of the average heat transfer coefficients over the length of the test tube on the refrigerant side:

$$h_r = (A_o \,/\, A_i) \cdot (1 / K - R_m - 1 / h_w)^{-1} \tag{1}$$

in which A_o and A_i are the surface areas of the outside and inside of the test tube, respectively. The quantity R_m denotes the thermal resistance at the inner tube and h_w is the water side heat transfer coefficient. The heat transfer coefficient, h_w in the annulus was determined using the well-known Dittus-Boelter correlation.

In equation (1), the quantity K presents the overall heat transfer coefficient and is defined here as follows:

$$K = Q_w \,/\, (A_i \cdot \Delta T) \tag{2}$$

in which Q_w is the heat quantity transferred in the test tube and ΔT is the log mean temperature difference. The heat quantity Q_w was given on the water side of the annulus. The temperature difference, ΔT, was determined from the inlet and exit temperatures of the water through the annulus and from the inlet and outlet saturation temperatures of the refrigerant flowing in the test tube. To check heat balance on the water and refrigerant sides of the test section, comparisons between both the heat balances were made. The difference between both the heat balances were found to be within 5% for all runs.

For the binary and ternary refrigerant mixtures, the saturation temperatures at both the inlet and outlet of the test section were defined as follows:

(1) For the evaporation experiment, the refrigerant temperature at the inlet of the test section was determined from the temperature of the two-phase region corresponded to the saturation pressure and the quality, and the refrigerant temperature at the outlet of the test section was determined from the dew-point temperature corresponding to the saturation pressure.

(2) For the condensation experiment, the refrigerant temperature at the inlet of the test section was determined from the dew-point temperature corresponding to the saturation pressure, and the refrigerant temperature at the outlet of the test section was determined from the bubble temperature corresponding to the saturation pressure.

4 RESULTS AND DISCUSSION

4.1 Heat transfer characteristics of refrigerant mixtures

The evaporation heat transfer coefficient results for a binary refrigerant mixture of HFC-32/134a(30/70wt.%) and a ternary refrigerant mixture of HFC-32/125/134a (23/25/52wt.%) are plotted in Figure 4 as a function of refrigerant mass flux. The data for HCFC-22 are given in the figure in order to compare with the results for the mixtures. The figure shows that the increases of the heat transfer coefficient with an increase in the value of mass flux for both the refrigerant mixtures is found to be significantly greater than that for HCFC-22. Compared the results for the binary refrigerant mixture with those for HCFC-22, the former is about 36% lower than the latter at G=150kg/m^{2}s, while the results for the both refrigerants show almost the same values at G=520kg/m^{2}s. Although the heat transfer results for the ternary refrigerant mixture show qualitatively the same as those for the binary refrigerant mixture, the values for the ternary mixture lie about 20% below those for the binary mixture.

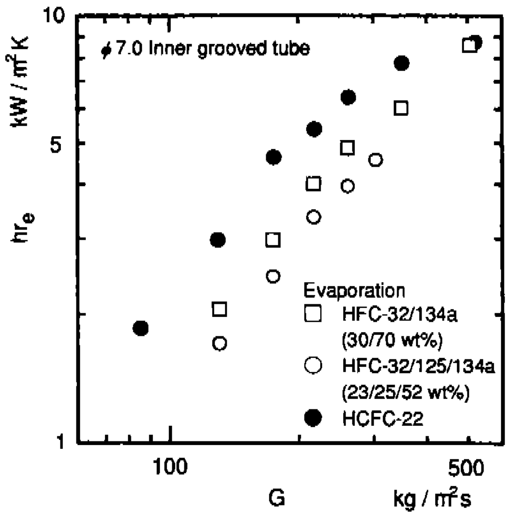

Figure 4 Variations of heat transfer coefficients with refrigerant mass flux during evaporation

The condensation heat transfer results are presented in Figure 5. The curve showing the results for HCFC-22 in the figure displays a remarkable trend that the heat transfer coefficients increase as the mass flux increases for G≦170kg/m^{2}s and then the results reach almost to the constant values for G >170kg/m^{2}s. On the contrary, both the results for the binary and ternary refrigerant mixtures increase monotonously with the mass flux. For instance, the result for the binary mixture at G=150kg/m^{2}s is about 45% lower than that for HCFC-22, while both the results for HCFC-22 and HFC-32/134a show almost the same values at G=520kg/m^{2}s. The fact that the heat transfer coefficient increases for the larger refrigerant mass flux has also appeared in the results for the binary mixture of HFC-32/134a (30/70wt.%) reported by Uchida et al [4]. The comparison between the results for the ternary and binary refrigerant mixtures demonstrates that the former is 10% smaller than those for the latter.

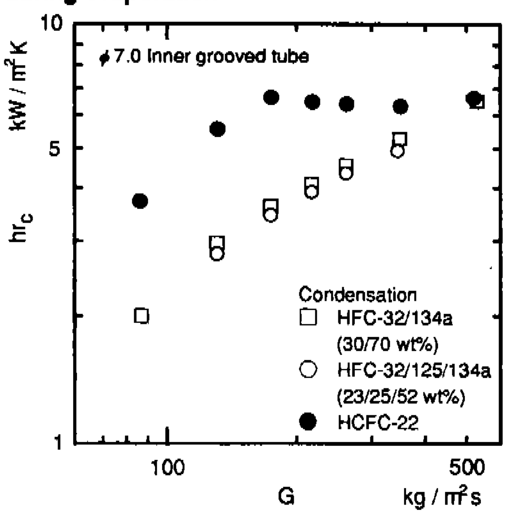

Figure 5 Variations of heat transfer coefficients with refrigerant mass flux during condensation

4.2 Pressure drop characteristics of refrigerant mixtures

The evaporation and condensation pressure drop results are presented in Figure 6. The figure shows that the relationships between pressure drop and mass flux for both the refrigerant mixtures are in qualitatively agreement with that for HCFC-22. For the binary refrigerant mixture, the evaporation and the condensation pressure drop results are respectively 33% and 10% larger than those for HCFC-22. For the ternary mixture, the evaporation pressure drop results are 25% larger than those for HCFC-22, while the results for condensation show the same values as the results of HCFC-22. The quantitative difference in the pressure drop results among HCFC-22, the binary and ternary refrigerant mixtures is mainly ascribed to the difference in vapor density of the refrigerants.

4.3 Effect of composition on heat transfer characteristics

The evaporation and condensation experiments were performed using HFC-32/134a and HFC-32/125/134a with various compositions, in order to verify the effect of composition on the heat transfer and pressure drop characteristics of the refrigerant mixtures.

Representative heat transfer coefficient data for the binary mixture are plotted in Figure 7 during evaporation and condensation as a function of the composition of HFC-32. All of the results are given for G=215 kg/m²s. In the figure, the dashed and chained lines correspond, respectively, to the experimental results for HCFC-22 and to the values interpolated linearly between the values of HFC-32 and HFC-134a. An overall perspective on the heat transfer results reveals a characteristic trend that both the evaporation and condensation heat transfer coefficients decrease with increasing the composition of HFC-32 for C<25~35wt.%, and then both of the results increase with the composition of HFC-32. The minimum values for the evaporation and condensation heat transfer coefficients are encountered at C=25wt.% and 30wt.%, respectively. The finding that the degradation of the heat transfer coefficients from the values interpolated between the values of HFC-32 and HFC-134a may be attributed to the non-linearity of the thermal properties and the circumferential and longitudinal mass transfer resistances proposed by Kedzierski et al [5].

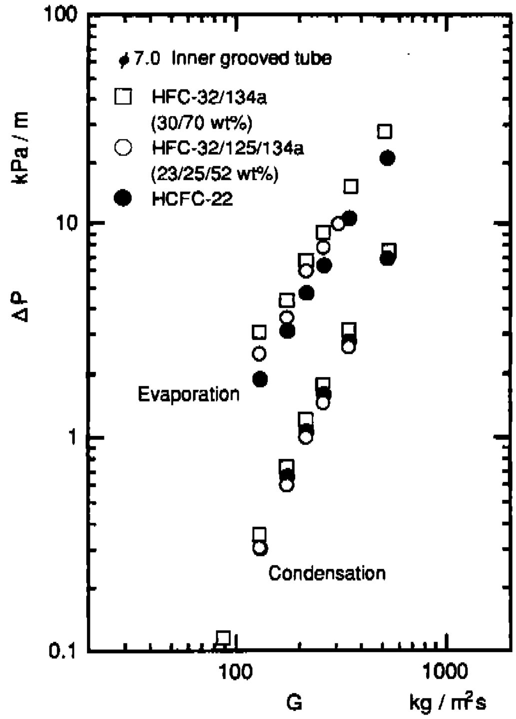

Figure 6 Variations of pressure drops with refrigerant mass flux during evaporation and condensation

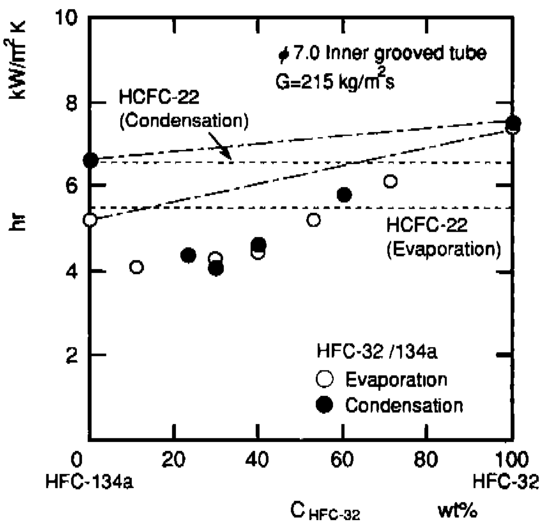

Figure 7 Effects of HFC-32 composition on heat transfer coefficients of HFC-32/134a during evaporation and condensation

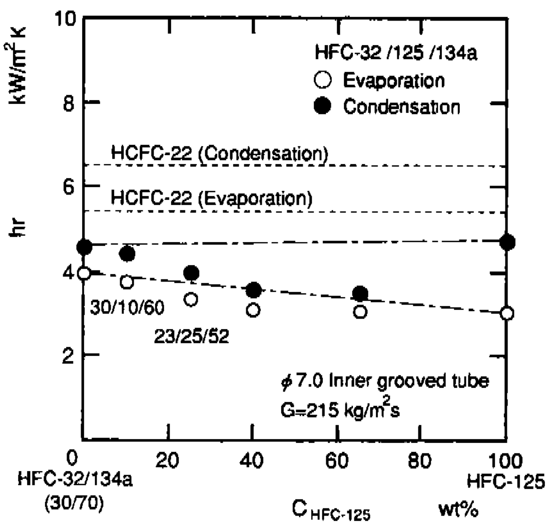

Figure 8 Effects of HFC-125 composition on heat transfer coefficients of HFC-32/125/134a during evaporation and condensation

The results of the evaporation and condensation heat transfer coefficients for the ternary mixture with a relative content 30wt.% HFC-32/70wt.% HFC-134a are shown in Figure 8. The evaporation heat transfer coefficients decrease monotonously with an increase in the composition of HFC-125, while the condensation heat transfer coefficients decrease for C<60wt.% and then increase for C≧ 60wt.%. However both the heat transfer coefficients are found to be lower than the values interpolated linearly between the results for HFC32/134a (30/70wt.%) and HFC-125.

4.4 Effect of composition on pressure drop characteristics

Figures 9 and 10 respectively display the results of the pressure drop for the binary and ternary refrigerant mixtures as a function of the composition. The Figure 9 shows that the values of the pressure drop decrease gradually with an composition of HFC-32 and that those are larger than the values interpolated between the values of HFC-32 and HFC-134a. On the other hand, the inspection of the Figure 10 reveals that the values of the pressure drop for the ternary mixture are just smaller than the interpolated values corresponded to the chained line in the figure. For the results of the pressure drop during condensation, it is clear that the pressure drop shows almost the same trends as those for evaporation.

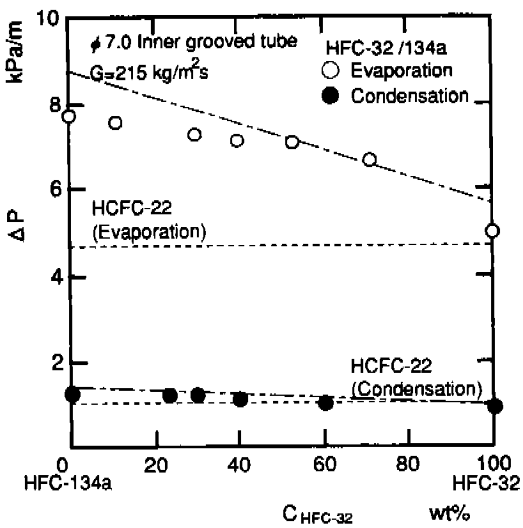

Figure 9 Effects of HFC-32 composition on pressure drop of HFC-32/134a during evaporation and condensation

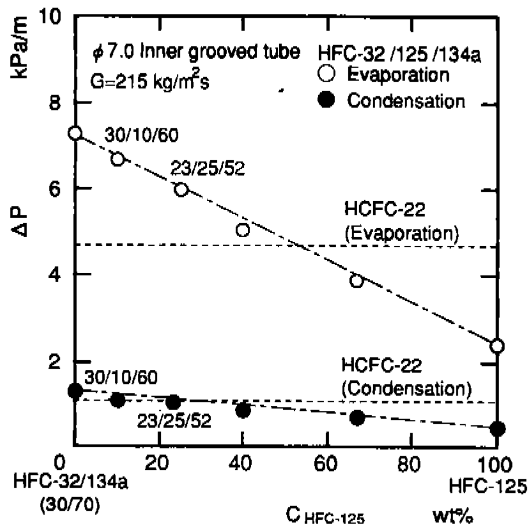

Figure 10 Effects of HFC-125 composition on pressure drop of HFC-32/125/134a during evaporation and condensation

CONCLUSIONS

The present study clarified the evaporation/condensation heat transfer and pressure drop characteristics for the binary and ternary refrigerant mixtures of HFC-32/134a and HFC-32/125/134a inside a horizontal grooved tube. Comparison between the evaporation/ condensation heat transfer coefficients for HFC-32/134a(30/70wt.%) and HFC-32/125/134a(23/25/52wt.%) showed that the results for the ternary mixture fell below those for the binary mixture. Furthermore, an interesting result for the binary and ternary refrigerant mixtures with various compositions indicated that the heat transfer coefficients were lower than the values interpolated linearly between the results for pure refrigerants.

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