

# Experimental study of forced convective boiling for non-azeotropic refrigerant mixtures R-22/R-124 in horizontal smooth tube

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## ABSTRACT

In this article, the experiment is carried out with four stages of test section and new experimental data of non-azeotropic refrigerant mixtures R-22/R-124 are presented. Existing correlations are analysed and new ones are proposed for pure refrigerants. The nucleate boiling correlation of the refrigerant mixtures is modified so as to obtain the evaporation heat transfer coefficient of the refrigerant mixtures in a smooth tube. The result shows that the majority of the predictions fall within  $\pm 30\%$  of the experiments.

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## 1. Introduction

Refrigerant mixtures are the major substitute refrigerants of CFCs and HCFCs. Thome [1] and Celata et al. [2] had pointed out that the heat transfer coefficient of refrigerant mixtures is lower than that of pure refrigerants. Heat transfer enhancement method is required to enhance the heat transfer coefficient of the refrigerant mixtures. Therefore, before using the refrigerant mixtures, it is necessary to conduct a further detailed investigation of their basic characteristics.

Several investigators [3–5] have performed studies on the evaporation of quasi-azeotropic mixtures (R410A) and zeotropic mixtures (R407C) in smooth and microfin tubes. The heat transfer phenomenon occurring during the evaporation of refrigerant mixtures in a tube is more complicated than that of pure refrigerants. For example, the experimental results of Kedzierski and Didion [6] indicate that a bubble is more difficult to be formed in refrigerant mixtures. Kedzierski et al. [7] analyzed that the reasons behind the lower heat transfer coefficient of the refrigerant mixtures are as follows:

- The effect of the nonlinear thermal physical properties of the refrigerant mixtures.
- A mass transfer resistance caused by a radial concentration gradient.
- A mass transfer resistance caused by a circumferential concentration gradient.

Considering R-22/R-114 refrigerant mixtures as an example, the contribution of the above three effects toward its low heat transfer coefficient can be split as approximately 14%, 55%, and 31%, respectively.

Since the heat transfer coefficient of non-azeotropic refrigerant mixtures is lower than that of pure refrigerants. The heat exchangers with larger heat transfer areas are required when using non-azeotropic refrigerant mixtures. Therefore, during the design of a heat exchanger, a highly accurate heat transfer formula is required to design the perfect heat exchanger. This article aims at investigating the heat transfer phenomenon occurring during the evaporation of refrigerant mixtures within a smooth tube. A new experimental data of non-azeotropic refrigerant mixtures R-22/R-124 are presented. The experimental data was used to derive empirical correlations for the evaporation heat transfer coefficients of pure and refrigerant mixtures in a smooth tube.

## 2. Experimental system

Fig. 1 illustrates the experimental system. It can be observed that the entire experimental system can be divided into four sections, i.e., a condensing system, refrigerant circulation system, pre-heater system, and hot water system.

### (1) Condensing system.

The condensing system consists of a thermostat that provides chilled water, a condenser, and a subcooler. After the refrigerant vapor exits the test section, it enters the condenser, and the chilled water provided by the thermostat

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**Nomenclature**

|                 |                                                                          |                      |                                                                    |
|-----------------|--------------------------------------------------------------------------|----------------------|--------------------------------------------------------------------|
| $A$             | heat transfer area, $\text{m}^2$                                         | $\Delta T_{id}$      | ideal temperature difference, K                                    |
| $AE$            | average error                                                            | $\Delta T_o$         | reference temperature difference, K                                |
| $C_p$           | specific heat, $\text{J kg}^{-1} \text{K}^{-1}$                          | $U$                  | overall heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$ |
| $d$             | inside diameter of the tube, m                                           | $X_f$                | flow quality                                                       |
| $G$             | mass flux, $\text{kg m}^{-2} \text{s}^{-1}$                              | $X_{tt}$             | Martinelli parameter                                               |
| $h$             | heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$               | $x$                  | mass concentration of volatile component                           |
| $i_{fg}$        | latent heat, $\text{J kg}^{-1}$                                          |                      |                                                                    |
| $k$             | thermal conductivity, $\text{W m}^{-1} \text{K}^{-1}$                    |                      |                                                                    |
| LMTD            | logarithm mean temperature difference                                    |                      |                                                                    |
| $M$             | molecular weight, $\text{g mol}^{-1}$                                    | <i>Greek symbols</i> |                                                                    |
| $\dot{m}$       | mass flow rate, $\text{kg s}^{-1}$                                       | $\delta_w$           | wall thickness, m                                                  |
| $\dot{m}_r$     | refrigerant mass flow rate, $\text{kg s}^{-1}$                           | $\mu$                | viscosity, $\text{kg m}^{-1} \text{s}^{-1}$                        |
| $N_p$           | numbers of experimental data                                             | $\rho$               | density, $\text{kg m}^{-3}$                                        |
| $PF$            | property factor                                                          |                      |                                                                    |
| $Pr$            | Prandtl number, $\mu C_p/k$                                              | <i>Subscripts</i>    |                                                                    |
| $P_c$           | critical pressure of refrigerant mixtures, $xP_{c1} + (1-x)P_{c2}$ , kPa | $b$                  | bubble point                                                       |
| $P_{c1}$        | critical pressure of volatile component, kPa                             | $cal$                | calculation                                                        |
| $P_{c2}$        | critical pressure of heavy component, kPa                                | $d$                  | dew point                                                          |
| $p_r$           | reduced pressure, $P/P_c$                                                | $exp$                | experiment                                                         |
| $\dot{Q}$       | average heat transfer rate, W                                            | $i$                  | tube side                                                          |
| $q''$           | heat flux, $\text{W m}^{-2}$                                             | $id$                 | ideal                                                              |
| $Re$            | Reynolds number, $Gd_i/\mu$                                              | $in$                 | inlet                                                              |
| $R_w$           | wall resistance, $\text{K W}^{-1}$                                       | $l$                  | liquid                                                             |
| $SD$            | standard deviation                                                       | $lo$                 | liquid only                                                        |
| $T$             | temperature, K                                                           | $o$                  | shell side                                                         |
| $T_b$           | bubble point temperature, K                                              | $p$                  | pure refrigerant                                                   |
| $T_d$           | dew point temperature, K                                                 | $p1$                 | pure refrigerant 1                                                 |
| $\Delta T$      | temperature difference, K                                                | $p2$                 | pure refrigerant 2                                                 |
| $\Delta T_{db}$ | temperature difference between dew point and bubble point, K             | $tp$                 | two phase                                                          |
|                 |                                                                          | $v$                  | vapor                                                              |

condenses the refrigerant vapor into a saturated liquid. Before the refrigerant enters the refrigerant flow meter, it passes through the subcooler so as to ensure that the refrigerant entering the flow meter is in a liquid state and to facilitate the refrigerant flow rate measurement. In addition, the flow rate and temperature of the chilled water entering the condenser can be adjusted so as to maintain a stable system pressure.

## (2) Refrigerant circulation system.

The main components of the refrigerant circulation system include a refrigerant pump, needle valve, dryer and filter, refrigerant flow meter, and test section. The refrigerant pump is a gear pump with an input voltage of 220 V and an output power of 373 W. The refrigerant pump is connected to an inverter that can be used to adjust the rotational speed of the pump and the refrigerant flow rate. The test section consists of a double tube with hot water flowing on the shell side and the refrigerant on the tube side. Thus, a counter flow is produced between the shell and tube sides. Pressure gauges are installed at both the entrance and exit ends of the test section to measure the pressure in the system. The exterior of the test section is coated with an insulation material of 50 mm thickness to prevent heat losses. The experiment is carried out with four stages of test section. The test tube was divided into four sections with the length of each section as 1.2 m. The advantage of using four stages experiment is that the adjustment of primary inlet quality can measure simultaneously four sets of heat transfer coefficients under different qualities. Therefore, the time needed for the experiment can be greatly reduced. The entrance and exit ends of each sub-section are installed with

T-type thermocouples so as to measure the temperature of the water and refrigerant. The overall length of the test tube is 4.8 m, the outer diameter is 9.52 mm, and inner diameter is 7.92 mm. The shell tube in the test section is a copper tube that has an outer diameter of 15.875 mm and an inner diameter of 13.875 mm. Furthermore, a sight glass tube is used to be connected the test tube at each small section. The flow pattern of the refrigerant mixtures can be observed through the sight glass.

## (3) Pre-heater system.

Pre-heater is a double tube heat exchanger with its shell side filled with hot water, and the flow rate and temperature of the hot water are adjusted to control the flow quality of the refrigerant exiting the pre-heater.

## (4) Hot water system.

Hot water system includes a hot water tank that has a heating capability of 5 kW and a hot water pump of 373 W. The hot water pump is connected to an inverter so as to adjust the flow rate of the hot water at the test section.

The refrigerant mixture used in experimental stand is R-22/R-124. As shown in Fig. 2, since the phase diagram of R-22/R-124 refrigerant mixtures and the heat and mass transfer effects of the non-azeotropic refrigerant mixtures are more apparent, the R-22/R-124 refrigerant mixtures has been selected for the current experiment rather than R-22 and R-124 that are also types of HCFC refrigerants.

Table 1 presents the test conditions of the experiment. The mass concentration of the R-22/R-124 mixtures varied between  $x = 0.0$  (pure R-124),  $x = 0.2$ ,  $x = 0.5$ ,  $x = 0.85$ , and  $x = 1.0$  (pure R-22); the mass flux ranged from 100–400  $\text{kg/m}^2 \text{s}$ ; flow quality ranged

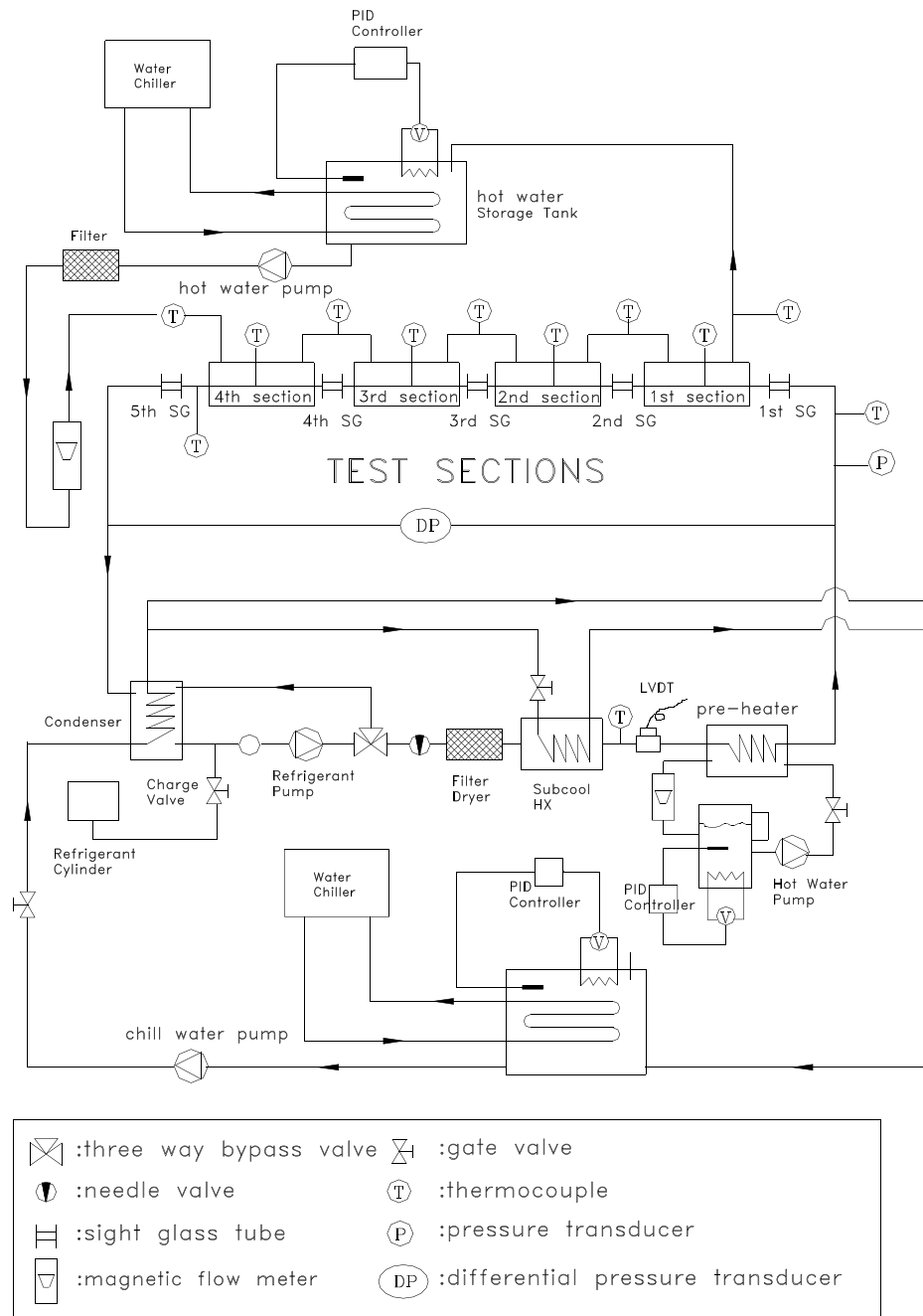


Fig. 1. A schematic diagram of the test apparatus.

between 0.1 and 0.9; and the heat flux varied between 5, 10, and 20 kW/m<sup>2</sup>. Since the saturation temperature of the refrigerant mixtures is not only related to the pressure but also related to the mass concentration of the refrigerants. The pressure is represented by a reduced pressure which is defined as the ratio between saturated pressure to critical pressure, wherein critical pressure  $P_c = xP_{c1} + (1 - x)P_{c2}$ . During the experimental process, we have adjusted the flow rate and temperature of chill water of condenser so as to change the saturated pressure of the system and to let reduced pressure maintain at constant value of 0.15.

Since the T-type thermocouple used in this system has been calibrated by a HP2804A-type quartz thermometer, the error in the measurement accuracy can be as small as  $\pm 0.1$  °C; further, the error in the measurement accuracy of the refrigerant and hot water flow meter after calibration can reach  $\pm 0.001$  l/s and that of the pressure

gauge can reach  $\pm 0.02\%$ . All the measurement signals, for example, temperature and voltage, pass through a personal computer HP3852 and get recorded therein. Table 2 presents the experimental uncertainty calculated by the method suggested by Moffat [8]. It can be observed that the uncertainty of the heat transfer coefficient within the tube is 8%.

Table 3 provides the measurement values of the heat loss at the test section. The measurement of heat loss was performed before loading the refrigerant into the system. After the tube side was evacuated, the shell side of the test section was filled with hot water, and the temperature difference between the entrance and the exit of the hot water was measured so as to calculate the heat lost to the environment on the hot water side of the test section. During the experiment, the variation in the hot water flow rate at the shell side was 1–5 kg/min, and it can be observed from Table 3 that the heat loss of

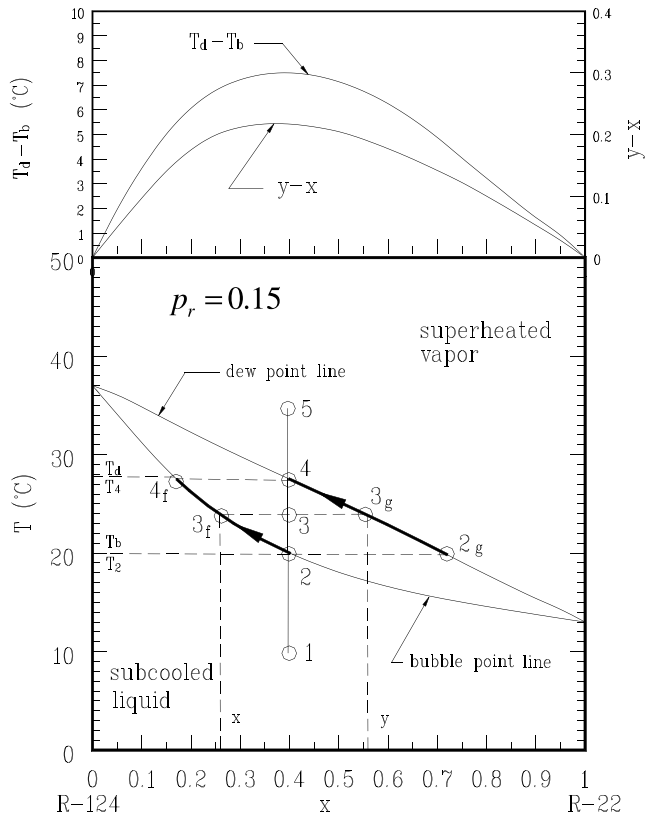


Fig. 2. The phase diagram of refrigerant mixtures R-22/R-124 at  $P_r = 0.15$ .

**Table 1**  
The test conditions of the experiment

| Parameters                              | Range                    |
|-----------------------------------------|--------------------------|
| Reduced pressure ( $P_r$ )              | 0.15                     |
| R-22 mass concentration ( $x$ )         | 0.0, 0.2, 0.5, 0.85, 1.0 |
| Quality ( $X_f$ )                       | 0.1 ~ 0.9                |
| Mass flux ( $G$ , kg/m <sup>2</sup> s)  | 100, 200, 300, 400       |
| Heat flux ( $q''$ , kW/m <sup>2</sup> ) | 5, 10, 20                |

**Table 2**  
The uncertainties of the experiment

| Primary uncertainties    |            | Derived uncertainties % |   |
|--------------------------|------------|-------------------------|---|
| $T$                      | 0.1 °C     | $\Delta T$              | 2 |
| $d_i$                    | 0.01 mm    | $A_i$                   | 1 |
| $l$                      | 2 mm       | $\dot{Q}$               | 4 |
| $\dot{m}_r$              | 0.001 kg/s | $U$                     | 5 |
| $\dot{m}_{\text{water}}$ | 0.001 kg/s | $h_{tp}$                | 8 |

**Table 3**  
The heat loss at the test section

|                          |          |          |
|--------------------------|----------|----------|
| Hot water mass flow rate | 1 kg/min | 5 kg/min |
| Heat loss                | 21.4 W   | 8.2 W    |

**Table 4**  
The energy error between refrigerant side and hot water side in the test section

| $G$ (kg/m <sup>2</sup> s) | $\dot{Q}_{\text{ref}}$ (W) | $\dot{Q}_{\text{water}}$ (W) | $\dot{Q}_{\text{ave}} = \frac{\dot{Q}_{\text{ref}} + \dot{Q}_{\text{water}}}{2}$ (W) | $\frac{ \dot{Q}_{\text{ave}} - \dot{Q}_{\text{water}} }{\dot{Q}_{\text{ave}}}$ % |
|---------------------------|----------------------------|------------------------------|--------------------------------------------------------------------------------------|----------------------------------------------------------------------------------|
| 100                       | 1136.8                     | 1282.9                       | 1209.9                                                                               | 7                                                                                |
| 200                       | 2239.7                     | 2375.4                       | 2307.6                                                                               | 3                                                                                |
| 300                       | 3372.7                     | 3465.9                       | 3419.3                                                                               | 2                                                                                |

this system was very small, i.e., even when the hot water flow rate was 1 kg/min, the heat loss was merely 21.4 W.

Table 4 presents the error in the energy measurement between the refrigerant side and the hot water side at the test section. During the test, the subcooled liquid refrigerant enters the test section, and then the hot water is used to heat the refrigerant into a superheated vapor. By calculating the enthalpy difference between the entrance and exit of the refrigerant, the heat transfer at the refrigerant side can be calculated. It can be observed from Table 4 that at  $G = 100$  kg/m<sup>2</sup> s, the heat capacity error is 7%, and when  $G = 300$  kg/m<sup>2</sup> s, the heat capacity error is 2%. Therefore, the temperature and flow rate measurements performed in this system at both the refrigerant side and hot water side are highly accurate.

### 3. Data reduction

If the temperature of the subcooled liquid refrigerant entering the pre-heater is  $T_i$ , refrigerant flow rate is  $\dot{m}_r$ , the heat transfer provided by the hot water of the pre-heater is  $\dot{Q}_{\text{pre}}$  wherein  $\dot{Q}_{\text{pre}}$  includes the sensible heat required to heat the refrigerant from the subcooled liquid to the saturated state and the latent heat required to heat the refrigerant in the saturated state to attain an inlet quality  $X_{f,\text{in}}$ , the quality at the entrance of the test section can be calculated as follows:

$$X_{f,\text{in}} = \frac{1}{i_{fg}} \left[ \frac{\dot{Q}_{\text{pre}}}{\dot{m}_r} - C_{p,l}(T_{b,\text{in}} - T_i) \right] \quad (1)$$

If the amount of heat transferred at the first section of the test tube is  $\dot{Q}_1$ , then the quality at the exit of the first section is

$$X_{f,1} = X_{f,\text{in}} + \frac{\dot{Q}_1}{\dot{m}_r i_{fg}} \quad (2)$$

However, the flow quality in the first section is defined as

$$X_{f,1\text{ave}} = \frac{X_{f,\text{in}} + X_{f,1}}{2} \quad (3)$$

Similarly, we can obtain the flow quality at the exit and the average flow quality in the other sections.

The relationship between the overall heat resistance and the individual heat resistances in the test section can be expressed as

$$\frac{1}{U_o A_o} = \frac{1}{h_c A_o} + R_f + R_w + \frac{1}{h_i A_i} \quad (4)$$

wherein  $R_w = \delta_w / (k_w A_w)$  is the thermal resistance at the tube wall,

$$U_o = \frac{\dot{Q}_{\text{water}}}{\text{LMTD} \times A_o} \quad (5)$$

$$\dot{Q}_{\text{water}} = (\dot{m} C_p \Delta T)_{\text{water}} \quad (6)$$

$$\text{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln \left( \frac{\Delta T_1}{\Delta T_2} \right)} \quad (7)$$

$$\Delta T_1 = T_{\text{water,in}} - T_{b,\text{out}} \quad (8)$$

$$\Delta T_2 = T_{\text{water,out}} - T_{b,\text{in}} \quad (9)$$

$T_{b,\text{in}}$  and  $T_{b,\text{out}}$  are the bubble point temperatures of the refrigerant at the entrance and the exit of the test section, respectively, calculated by using the REFPROP [9] program. The heat transfer coefficient of hot water  $h_o$  at the shell side is 1966 W m<sup>-2</sup> K<sup>-1</sup>, which can be obtained by means of Wilson plot techniques [10]. Eq. (4) can be used to calculate the experimental evaporation heat transfer coefficient  $h_{tp}$  for the refrigerant mixtures.

### 4. Result and discussion

The horizontal axis of Fig. 3 represents the flow quality ( $X_f$ ) of the refrigerant mixtures R-22/R-124, and the vertical axis, the

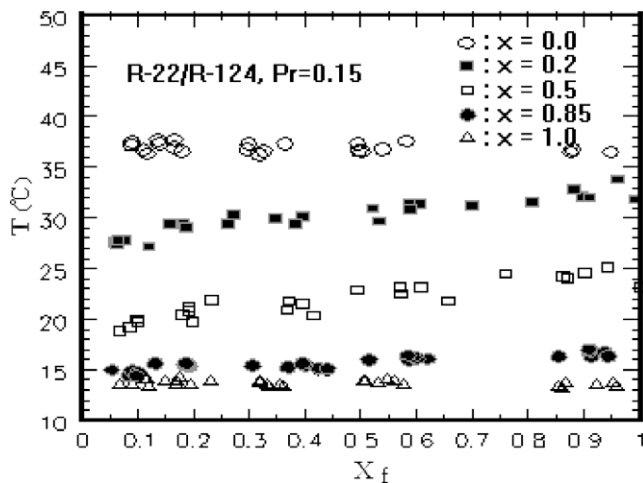


Fig. 3. Measured saturation temperature versus inlet quality for R22/R124.

measured saturation temperature of the refrigerant. It can be observed from the figure that the evaporation temperatures of the pure refrigerants R124 ( $x = 0.0$ ) and R22 ( $x = 1.0$ ) decrease slightly as the flow quality increases. The reason behind this is that as the test section is 4.8 m long, the pressure drop caused by the flow friction can lead to a reduction in the saturation pressure, which in turn, leads to a slight decrease in the saturation temperature of the pure refrigerants. However, the evaporation temperature of the refrigerant mixtures exhibits an obvious rise as the quality increases. Because when the flow quality is high, the liquid refrigerant mixture is mostly composed of the R-124 refrigerant that has a high saturation temperature. Hence, the measured saturation temperature of the refrigerant mixtures increases with the flow quality. Although the pressure drop in the refrigerant mixtures is an issue too, the effect of the temperature rise caused by the concentration variation is higher than the effect of the temperature drop caused by the frictional pressure drop. The saturation temperature of the refrigerant mixtures exhibits a rise instead of a drop. It can be observed from the phase diagram in Fig. 2 that since the temperature difference ( $T_d - T_b$ ) between the dew point and the bubble point at the mass concentration of  $x = 0.5$  is larger than that at  $x = 0.2$  and  $x = 0.85$ . The temperature rise in the refrigerant mixtures, as observed from Fig. 3, at the mass concentration of  $x = 0.5$  is approximately 6 °C, which is larger than the temperature rise at other mass concentrations.

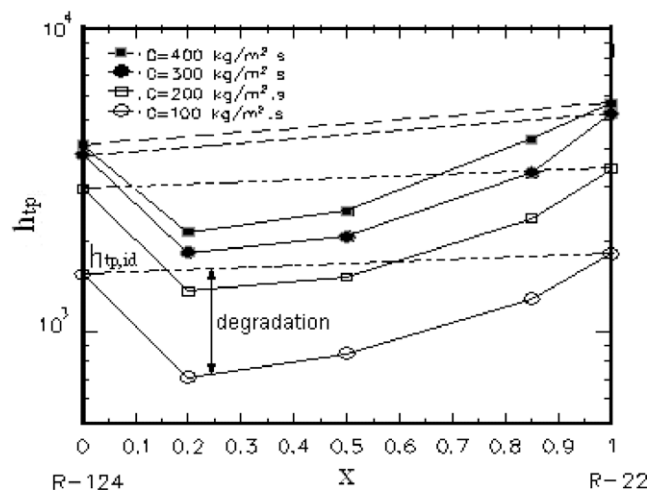


Fig. 4. The heat transfer coefficient versus concentration for R-22/R-124.

It can be observed from Fig. 4 that the heat transfer coefficient of the refrigerant mixtures exhibits a serious degradation phenomenon that can be explained by Ross et al. [11]. They have pointed out that such a heat transfer coefficient degradation phenomenon observed in refrigerant mixtures is because it is more difficult for bubbles to be formed in the refrigerant mixtures than in the pure refrigerants. Fig. 4 illustrates a comparison of the effect of the mass flux on the heat transfer coefficient. It can be observed that the larger the mass flux, the higher the heat transfer coefficient. It can be seen from our experimental data that when mass flux  $G = 100, 200, 300, 400 \text{ kg m}^{-2} \text{ s}^{-1}$ , the relative heat transfer coefficient ratios  $h_{tp}/h_{tp,uid}$  are 0.39, 0.40, 0.45, 0.49, respectively. Therefore, the trend that degradation get decreased along with the increase of mass flux. Because at a high mass flux, the flow rate is high, turbulence is large, the mass concentration of the refrigerant mixtures within the tube becomes highly homogeneous, and the mass transfer resistance in turn reduces. Therefore, at a high mass flux, the degradation of the heat transfer coefficient of the refrigerant mixtures is smaller than that at a low mass flux.

Figs. 5–7 illustrate the plots of  $h_{tp}/h_{io}$  versus  $1/X_{tt}$  for the pure refrigerants R-22 ( $x = 1.0$ ) and R-124 ( $x = 0.0$ ) and the refrigerant mixtures R-22/R-124 ( $x = 0.5$ ), wherein  $h_{tp}$  is the evaporation heat transfer coefficient,  $h_{io}$  is the heat transfer coefficient of forced

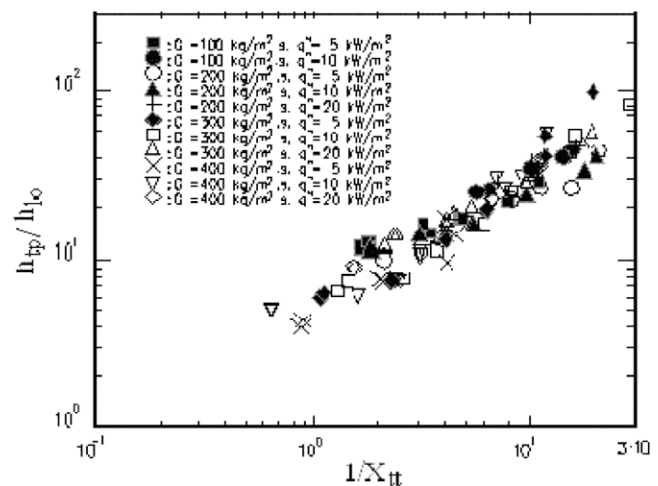


Fig. 5. The relationship between heat transfer coefficient and  $1/X_{tt}$  for R-22 ( $x = 1.0$ ).

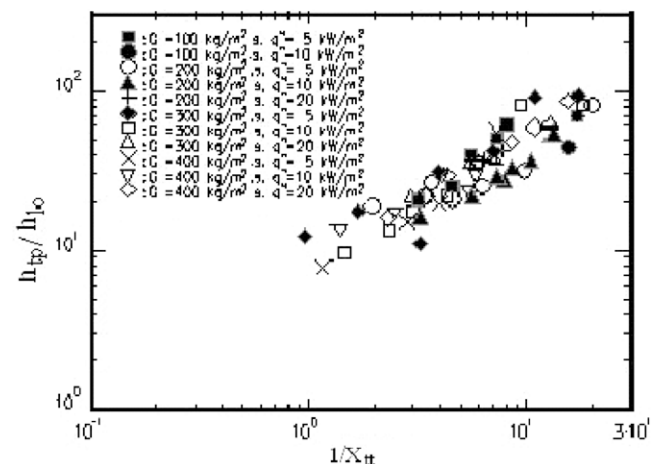


Fig. 6. The relationship between the heat transfer coefficient and  $1/X_{tt}$  for R-124 ( $x = 0.0$ ).

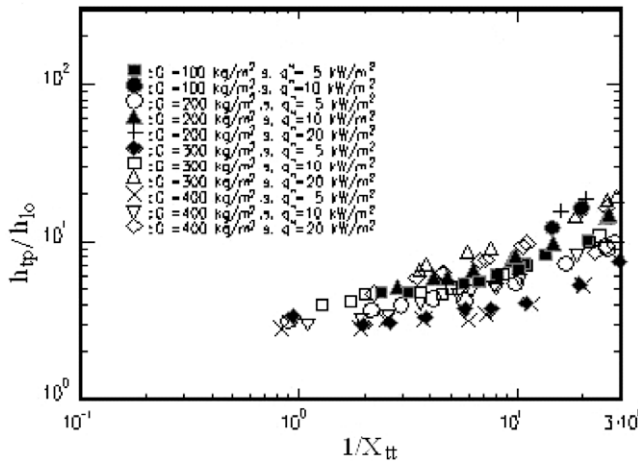


Fig. 7. The relationship between the heat transfer coefficient and  $1/X_{tt}$  for R-22/R-124 ( $x = 0.5$ ).

convection in the tube for the liquid refrigerant, and  $X_{tt}$  is the Martinelli parameter, and they are defined as [20]

$$h_{lo} = 0.023 \frac{k_l}{d_i} Re_{lo}^{0.8} Pr_l^{0.4} \quad (10)$$

$$Re_{lo} = \frac{G(1 - X_f)d_i}{\mu_l} \quad (11)$$

$$X_{tt} = \left( \frac{1 - X_f}{X_f} \right)^{0.9} \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \left( \frac{\mu_l}{\mu_v} \right)^{0.1} \quad (12)$$

It can be observed from Figs. 5–7 that the slope of the evaporation heat transfer coefficient of the refrigerant mixtures is smaller than that of the pure refrigerants, which implies that due to a mass transfer resistance factor, even under an annular flow state, the enhancement effect of the heat transfer coefficient of the refrigerant mixtures is lower than that of the pure refrigerants. It can also be observed from Figs. 5 and 6 that the heat flux  $q''$  does not have an obvious effect on the evaporation heat transfer coefficient of the pure refrigerants in the smooth tube. In fact, the effect of heat flux  $q''$  is very small, for example, in a research performed by Jung et al. [19], it was found that of all the effects observed on the evaporation heat transfer coefficient, only approximately 5% is contributed by  $q''$ . In many boiling conditions the convective heat transfer contribution is dominant. In order to simplify the heat transfer formula of the pure refrigerants in a smooth tube so as to facilitate engineering applications, the Wadekar [17] correlation was modified. We have neglected the heat flux effect and employed a regression analysis method to obtain the correlation between the evaporation heat transfer coefficient and  $1/X_{tt}$  for the pure refrigerants in the smooth tube, as shown by Eq. (13) wherein  $M$  is the molecular weight of the refrigerant. The  $R$  square value of Eq. (13) is 0.875.

$$\ln \left( \frac{h_{tp,p}}{h_{lo}} \right) = -1.71 + 0.77 \ln \left( \frac{1}{X_{tt}} M \right) \quad (13)$$

Using the sight glass on the experimental system to view the flow pattern, it can be found that when the mass flux  $G = 100 \text{ kg/m}^2 \text{ s}$ , no matter how good the flow quality of the refrigerant is, its flow pattern is always stratified. In order to avoid complications due to the heat transfer correlation, Eq. (13) is applicable only when  $G > 100 \text{ kg/m}^2 \text{ s}$  and flow quality ranged between 0.1 and 0.9.

Fig. 8 illustrates the comparisons between the experimental values obtained for the evaporation of R-22 and R-124 pure refrigerants in the smooth tube and the prediction values of the correlations proposed by Kandlikar [12], Mikielwicz et al. [13], Shah [14],

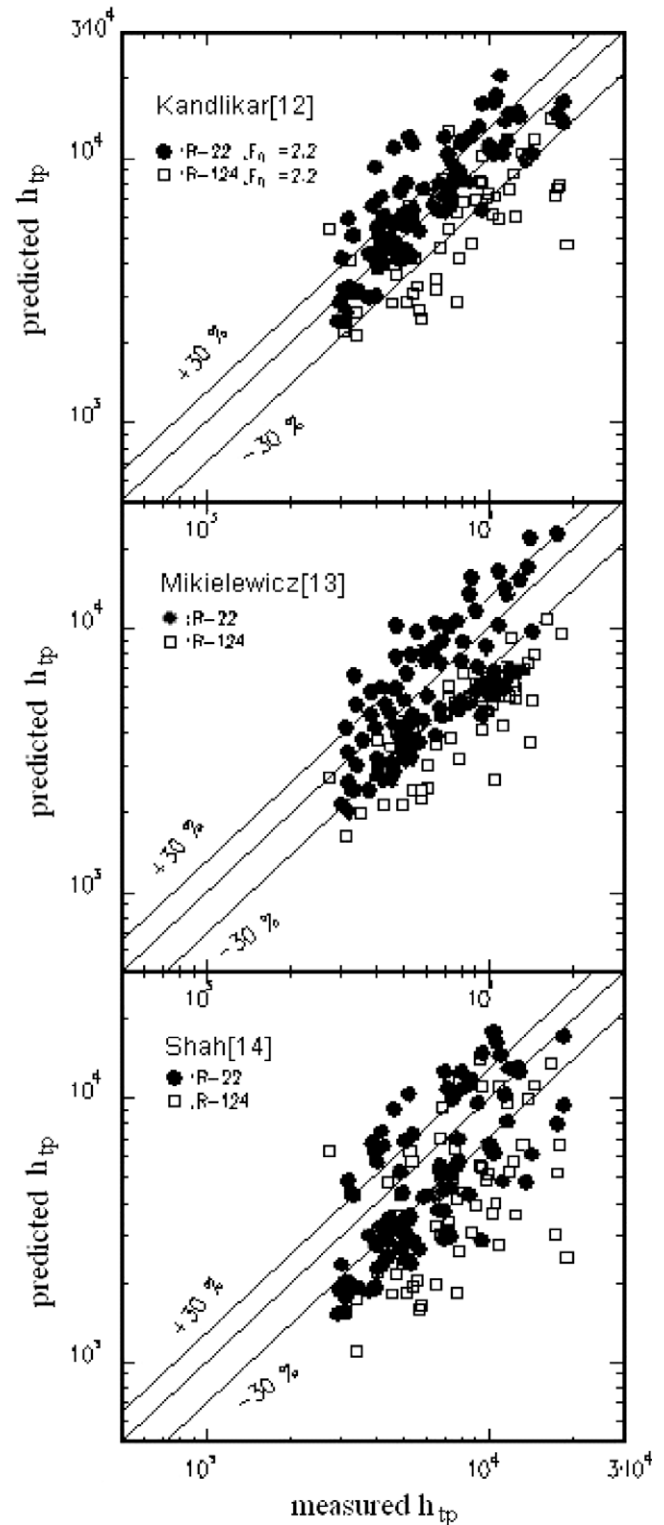


Fig. 8. A comparison between the prediction value of published correlations and the experimental value of pure refrigerants R-22 and R-124.

etc. Among the results, the prediction result of Kandlikar [12] is more appropriate; however, since Kandlikar [12] did not provide the  $F_{fl}$  value of R-124, here we have assumed that the  $F_{fl}$  value of R-124 is the same as that of R-22, that is, 2.2. Table 5 lists the comparisons between the correlation prediction values and the prediction values of Eq. (13). The result indicates that Eq. (13) provides the best prediction result. Thus, the prediction result of Kandlikar

**Table 5**

Comparison of standard deviation between the prediction heat transfer coefficient of pure refrigerant and the experimental value of R-22, R-124

| Correlations              | Standard deviation (SD) % |
|---------------------------|---------------------------|
| Gungor and Winterton [15] | 74                        |
| Steiner and Taborek [16]  | 53                        |
| Shah [14]                 | 48                        |
| Wadekar [17]              | 41                        |
| Mikielewicz et al. [13]   | 40                        |
| Kandlikar [12]            | 38                        |
| Eq. (13)                  | 23                        |

Note. SD =  $\sqrt{\frac{\sum[(h_{cal} - h_{exp})/h_{exp}]^2}{N_p}}$

[12] is the second best, and the prediction standard deviations are 23% and 38%.

From Eq. (13), we can obtain, respectively, the heat transfer coefficient  $h_{tp,p1}$  of the higher volatile component, i.e., R-22 and the heat transfer coefficient  $h_{tp,p2}$  of the lesser volatile component, i.e., R-124. The ideal heat transfer coefficient of the refrigerant mixtures, as denoted by the dashed line in Fig. 4, can be obtained.

$$h_{tp,id} = \left[ \frac{x}{h_{tp,p1}} + \frac{(1-x)}{h_{tp,p2}} \right]^{-1} \quad (14)$$

From our previous research [11] on the pool boiling of refrigerant mixtures, the correlation equations of the heat transfer coefficient of the pool boiling of refrigerant mixtures can be expressed as shown in Eqs. (15)–(21).

$$\frac{h_{tp}}{h_{tp,id}} = \frac{1}{1 + \frac{\Delta T_{db}}{\Delta T_{id}} (PF) fn(q'', p_r)} \quad (15)$$

$$\text{wherein } \Delta T_{db} = T_d - T_b \quad (16)$$

$$\Delta T_{id} = \frac{q''}{h_{tp,id}} \quad (17)$$

$$PF = \frac{\rho_v}{\rho_l} \left( \frac{k_l \mu_l M}{\rho_l C_{p,l} T_b} \right)^{0.5} \quad (18)$$

$$fn(q'', p_r) = \left( 1.5 \times 10^{-5} \frac{q''}{p_r} + \frac{25}{\Delta T_0} \right) \times 10^6 \quad (19)$$

$$\Delta T_0 = \Delta T_{db} \quad \text{for } \Delta T_{db} > 5 \quad (20)$$

$$\Delta T_0 = 5 \quad \text{for } \Delta T_{db} \leq 5 \quad (21)$$

We found that when the numerator of  $\Delta T_0$  in Eq. (19) is changed from 25 to 75; as shown in Eq. (22) which the square value is 0.858. Then Eqs. (15)–(18) and Eqs. (20)–(22) can be used to predict the heat transfer coefficient when the refrigerant mixtures is evaporated in the smooth tube.

$$fn(q'', p_r) = \left( 1.5 \times 10^{-5} \frac{q''}{p_r} + \frac{75}{\Delta T_0} \right) \times 10^6 \quad (22)$$

Fig. 9 illustrates the comparisons between the experimental values of the heat transfer coefficient and the prediction value of the correlations for the R-22/R-124 refrigerant mixtures as proposed by Murata [18] and Jung et al. [19]. The correlation prediction values of Murata [18] are lower than the experimental values, but the correlation prediction values of Jung et al. [19] are higher than the experimental values. Table 6 presents a comparison of the error between the prediction and experimental values of the refrigerant mixtures. Among them, the prediction result of Eq. (15) is the best with a standard deviation of 26%.

Fig. 10 is a comparison between the experimental value of the heat transfer coefficient and the prediction value of Eq. (15), and approximately 92% of the prediction values fall within  $\pm 30\%$ . Fig. 11 is a comparison between the prediction value of Eq. (15) and the experimental value of Jung et al. [19]; Jung et al. [19] used

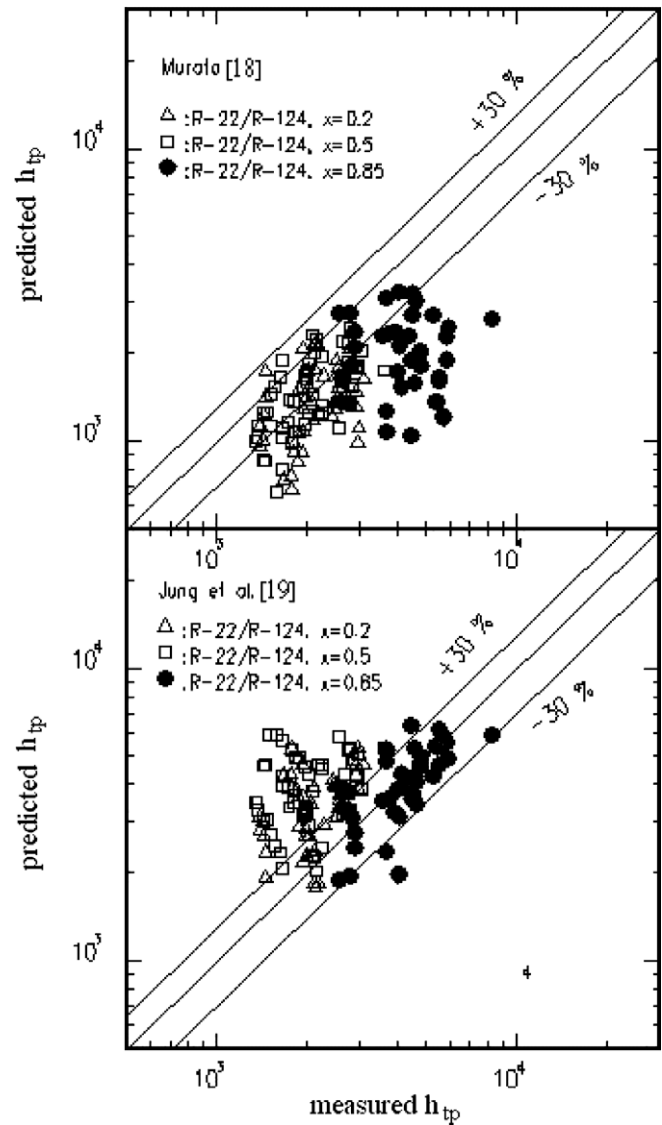


**Fig. 9.** A comparison between the prediction value of published correlations and the experimental value of R-22/R-124 refrigerant mixtures.

**Table 6**

Comparison of standard deviation between the prediction heat transfer coefficient of refrigerant mixtures and the experimental value of R-22/R-124

| Correlations     | Standard deviation (SD) % |
|------------------|---------------------------|
| Jung et al. [19] | 116                       |
| Murata [18]      | 41                        |
| Eq. (15)         | 26                        |

stainless steel test tube with the inner diameter of the tube  $d_i = 9.1$  mm, reduced pressure  $p_r = 0.08$ , and the refrigerant mixtures R-22/R-114. The prediction result of Fig. 11 indicates that 85% of the prediction values fall within  $\pm 30\%$ , and this result again proves the reliability of Eq. (15).

## 5. Conclusion

This article aims at investigating the heat transfer phenomenon occurring during the evaporation of non-azeotropic refrigerant mixtures in a smooth tube. The experiment is carried out with four stages of test section and new experimental data of non-azeotropic

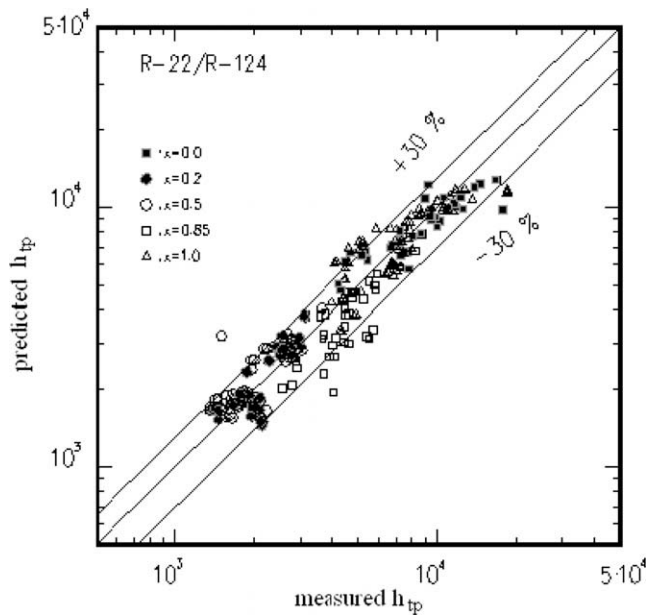


Fig. 10. A comparison between the prediction value of Eq. (15) and the experimental value of R-22/R-124 refrigerant mixture.

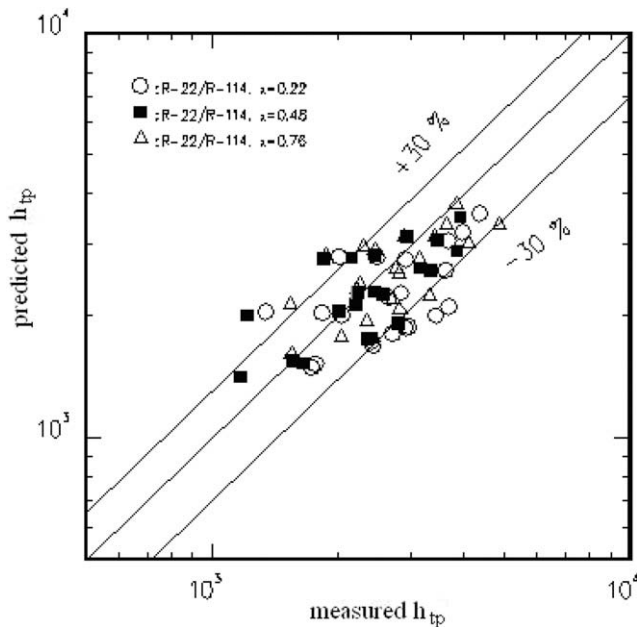


Fig. 11. A comparison between the prediction value of Eq. (15) and the experimental value of R-22/R-114 found by Jung et al. [19].

refrigerant mixtures R-22/R-124 are presented. We proposed a general correlation for predicting the evaporation of a pure refrigerant in a smooth tube. Eq. (13) shows a better prediction result as compared to the existing correlations. The majority of the prediction values fall within  $\pm 30\%$  of the pure refrigerant data. It can be found from a microscopic view point that the growth process of a bubble is an evaporation phenomenon. Therefore, the evaporation heat transfer coefficient of the refrigerant mixtures in a smooth tube can be predicted by modified the nucleate boiling correlations. The prediction results of Eqs. (15)–(22) show that the majority of the prediction values fall within  $\pm 30\%$ . Therefore, Eqs. (10)–(22) can be used for predicting the evaporation heat transfer coefficients of pure refrigerants and refrigerant mixtures.

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