Forced Convective Boiling in Vertical Tube for Binary Refrigerant Mixtures of R11 and R113

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An experimental study was carried out on convective boiling heat transfer for mixtures of R11 and R113 flowing in a uniformly heated vertical tube by measuring the wall and bulk temperatures, and the results were compared with an existing correlation. A reduction of the average heat transfer coefficient for mixtures was verified for flow boiling. It was observed that two kinds of boiling behavior existed depending on mass flux. It was also found that the Chen's correlation was particularly successful for the case of high mass rate flow in which convective boiling prevailed. However in the case of low mass rate flow where nucleate boiling was dominant, the Chen's correlation was found to be inappropriate. Mass transfer resistance in the liquid film played a vital role for determining the heat transfer coefficient of refrigerant mixtures. It has been also found that the equilibrium assumption was hardly applicable to the convective boiling phenomena.

Key Words: Convective Boiling, Refrigerant Mixtures, Mass Transfer Resistance

Nomenclature

C_P : Specific heat (J/kgK)
D : Molecular diffusion coefficient (m^2/s)
d : Inside diameter of heating tube (m)
F : Two phase multiplier defined by Chen
G : Mass flux (kg/m^2s)
h_{fg} : Latent heat of vaporization (J/kg)
h_m : Mass transfer coefficient (m/s)
k : Thermal conductivity (W/mK)
P : Pressure (kPa)
Pr : Prandtl number
P_s : Saturation pressure (kPa)
ΔP_s : Different in saturation pressure corresponding to $\Delta T_s = T_w - T_s$
q : Heat flux (W/m^2)
Re : Reynolds number
S : Nucleate boiling suppression factor defined by Chen
Sc : Schmidt number
T : Temperature (K)
x_m : Local liquid mass fraction of the less volatile component
y_m : Local liquid vapor fraction of the less volatile component
z : Distance from the inlet of test section (m)

Greek Letters

α_B : Nucleate boiling heat transfer coefficient (W/m^2K)
α_c : Total heat transfer coefficient (W/m^2K)
α_L, α_{LO} : Forced convection heat transfer coefficient (W/m^2K)
α_m : Measured heat transfer coefficient (W/m^2K)
α_T : Thermal diffusivity (m^2/s)
μ : Viscosity $(pa \cdot s)$
ρ : Density (kg/m^3)
σ : Surface tension (N/m)

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Subscripts

Binary : Binary liquid
l : Liquid component
m : Mixture
s : Saturated
TP : Two-phase
v : Vapor
w : Wall

1. Introduction

The study of flow boiling of multi-component mixtures is of practical significance to design the evaporators and the condensers used in chemical and petrochemical processes. Especially the knowledge for the flow boiling of azeotropic and non- azeotropic binary mixtures is demanded at present because of their possibility of alternative refrigerants (Watanabe, 1988). Furthermore, the non-azeotropic binary mixture was found to be an appropriate refrigerant to achieve the higher COP for heat pumps (NEDO Report, 1988).

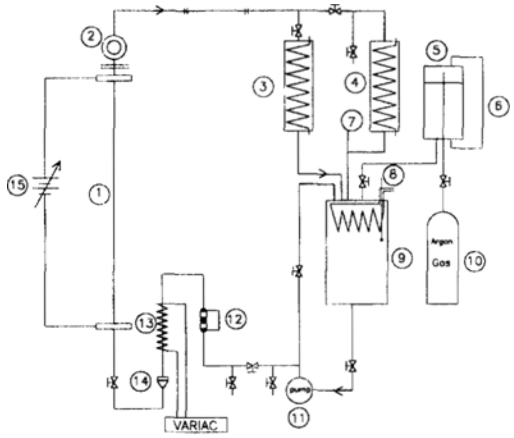
It is well known that the pool boiling heat transfer coefficient of binary mixture is lower than the one based on linear interpolation between the values of two pure components. This reduction of the heat transfer coefficient of binary mixture in pool boiling has been verified experimentally by many workers (Thome and Shock, 1984). However much less work has been carried out on the convective boiling of mixtures (Toral et al., Ross et al., Fujita et al., Jung et al). Full suppression of nucleate boiling was found to be easier to achieve with mixtures than the pure liquids[Ross et al. 1987]. Also it has been found that the physical property variation associated with mixture was responsible for the degradation of the heat transfer coefficent under the suppression of nucleate boiling[Jung et al, 1989]. However they have done the experiments only in the nucleate and convective boiling regimes where the bulk temperature of liquid might be regarded as the saturation temperature corresponding to the inlet pressure of the test section. In this study, we tried to measure the bulk liquid temperature to investigate the boiling inception as well as the dry out phenomena where an abrupt change of the bulk liquid temperature occurred. Also we tried to investigate the nonequilibrium effect in convective boiling regime by this method. In order to examine the specific physical processes governing the heat transfer of the mixtures, the data were compared with an existing correlation.

Heat transfer deterioration was observed in the boiling of the mixtures. It was also found that the Chen's correlation (Bennett and Chen, 1980) was particularly successful, for the prediction of the flow boiling heat transfer coefficient of mixtures in the case of high mass flux flow in which convective boiling prevailed. However in the case of low mass flux flow where nucleate boiling was dominant the Chen's correlation was found to be inappropriate. The equilibrium assumption turned out to be valid only for a particular case of the moderate heat flux (40 kW/m^2) with low mass flux ($250 kg/m^2s$).

2. Experimental Apparatus

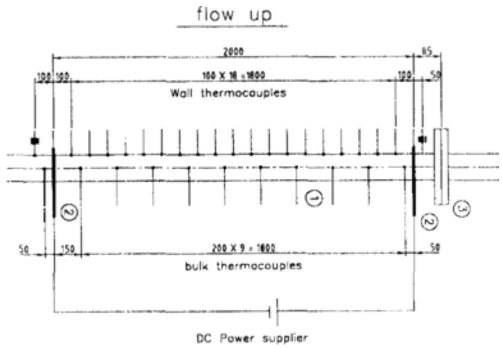
The experimental apparatus which is essentially a forced circulation loop, consists of condensers (③,④), storage tank (⑨) and surge tank (⑤), a circulation pump (⑪), and associated valves and piping. The schematic of the experimental apparatus is shown in Fig. 1.A friction pump (Wesco type) with mechanical seal delivered subcooled liquid refrigerant to the test section. The test section consisted of a stainless steel tube of 2.4 m long with a 10.2 mm ID and a 1 mm wall thickness. The heated length of a test section was 2 m. Heat was generated in the section (①) by applying a DC voltage difference along the tube. The inlet section was long enough (400 mm) to secure the fully developed flow at the entrance of the heated section.

The vapor generated in the test section was condensed in the water-cooled condensers placed above the storage tank. The pump then drew on the liquid in the storage tank to complete a cycle. The surge tank liquid level controlled by argon gas and the water flow rate to the condensers were used to adjust the desired pressure level at the inlet of the test section. The refrigerant flow rate



① Test section
② Sight glass
③ 1st condenser
④ 2nd condenser
⑤ Surge tank
⑥ Level meter
⑦ Pressure gauge
⑧ Thermometer
⑨ Storage tank
⑩ Argon-gas tank
⑪ System pump
⑫ Turbine flow meter
⑬ Preheater
⑭ Mixing chamber
⑮ DC power supplier

Fig. 1 Schematic diagram of experimental apparatus.



① Test section
② Electrode
③ Flange
■ Pressure transducer
● Thermocouple

Fig. 2 Schematics diagram of test section.

modified by the by-pass line was determined by means of a calibrated turbine flow meter(⑫) in the subcooled liquid line.

The outside wall temperatures were measured at 19 axial stations with equal interval of 100 mm in the heated section as shown in Fig. 2.The junction made of 0.30mm diameter K-type thermocouple was isolated electrically from the tube by a very thin layer of mica(less than 0.1 mm). To maintain good thermal contact, the thermocouple was clamped to the tube by teflon coated copper strip. The bulk fluid temperatures were also measured at ten stations in the heated section and one in the inlet section as shown in Fig. 2.The sheathed thermocouples of 1.6mm diameter with bare junction were inserted into the center of test section through the 3.175 mm diameter stainless tube welded at test section. The liquid temperature was measured correctly with these instream thermocouples (Bennedict, 1966). The temperature of liquid at inlet section was controlled by the preheater. Pressures were also measured by the calibrated pressure transducer at the single phase inlet and two phase outlet positions outside the heated section. Sight glass was installed at the two phase outlet to observe the flow pattern. The test section was entirely insulated by the glass-fiber with approximately 30 mm radial thickness to reduce the heat flow to the surroundings.

The acquisition of the data obtained from the thermocouples and the pressure transducers was performed by DAS-16F(OMEGA) connected to a PC. The current to the heated section was measured by DC Ammeter (HIOKI, Model 3265) and the DC voltage difference by precision Multimeter(Analogic, Model DP 100). If the time rate change of the wall temperature was less than 0.1 ℃,the system was assumed to be reached at steady state. Once the steady state has been established, data aquisition was started. The data acquisition process was as followed. For each 3 second after the data acquisition started, the data collected from each thermocouple(30channel) and pressure transducer(2 channel) at 0.2 second interval were averaged separately to make a data set(or 32 data points). This process was continued during 30 seconds to obtain 10 data sets for the 32 data points. Experimental data at each measured station were determined by averaging the ten data values, which gave one final data set for the 32 data points. Such procedure was repeated for every experimental condition, which provided 150 data sets (or 4, 800 data points), covering various composition of R11, various mass and heat fluxes and two different ambient pressures of 3 and 5 bar.

3. Data Evaluation and Data Validity

Heat flux applied to the heated section was calculated based on the inside surface area of the tube. Inside wall temperatures (T_{wi}) were calculated from the measured outside temperature by employing the one dimensional, radial steady state conduction equation with uniform heat generation. The temperature drops inside the tube wall were 0.3℃ to 2.7℃, depending on the heat flux applied. The heat transfer coefficient was calculated from the relation,

$$\alpha = q/(T_w - T_b) \tag{1}$$

where T_b was the bulk temperature measured at the center of the test section. Note that the previous researchers used T_b as T_s, the saturation temperature by assuming the equilibrium condition.

The accuracy of the wall superheat measured by K-type thermocouples was about ± 0.7℃ and of the bulk liquid temperature was about ± 0.1℃. With these errors in the wall superheat of $\pm$8% and $\pm$4% accuracy in the heat flux ($\pm$3% for measuring current and $\pm$1% for voltage), the accuracies of the local heat transfer coefficients were determined to be $\pm$13%. The uncertainties in the system pressure and flow rate measurements were about $\pm$2% and $\pm$2.5% respectively. However these quantities did not affect the heat transfer measurements. The quality was calculated from the energy balance equation by assuming of the phase and thermal equilibrium conditions.

In order to verify the temperature measurements, forced convection test was made for each test fluid. Because of nonuniformity in the pipe material due to the welding of thermocouple guide tubes on the pipe wall, the differences between wall and bulk temperature were not constant for the forced convection tests. Therefore the temperature data measured were calibrated so that both temperatures increased linearly along the tube for every pure and mixture tests. The maximum deviation obtained by the least square technique was less than ± 1.3℃.

The measured values of the average heat transfer coefficients along the test section were within the range of $\pm$7.5% for pure liquids and $\pm$10.5% for mixtures to the well known correlation[Incropera and Dewitt, 1985] such as

$$\alpha_L = 0.023 \frac{k_l}{d} R_{el}^{0.8} P_{rl}^{0.4} \tag{2}$$

For convective flow, the calibrated data of wall and bulk temperature and the corresponding heat transfer coefficient along the test section for R11 and a mixture (X=0.5) were shown in Fig. 3.The heat transfer coefficients calculated from Eq. (2) were 1180 kW/m²K for R11 and 1084 kW/m²K for the mixture of X=0.5.On the other hand the measured values were 1192 kW/m²K and 932 kW/m²K respectively.

The phase equilibrium diagram for the R11/R113 mixture at various pressures, which was obtained by using the Peng-Robinson equation of state(1976) which yielded good estimation of the binary refrigerant mixtures[Nakaiwa et al. 1988] was shown in Fig. 4.The mole fraction of a more volatile component, R11 was used to express the mixture composition. The other physical properties of mixtures were obtained from the appropriate mixing rule(Reid et al., 1977).

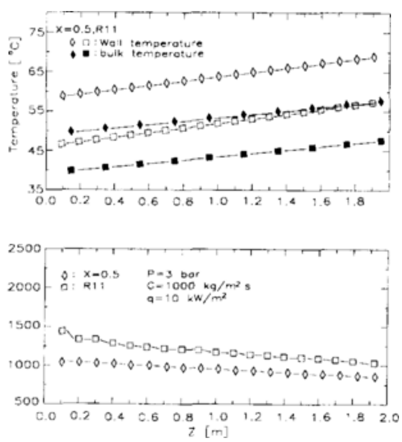


Fig. 3 Axial variations of wall temperature, bulk temperature(above) and heat transfer coefficient(below) for a single phase foced convection.

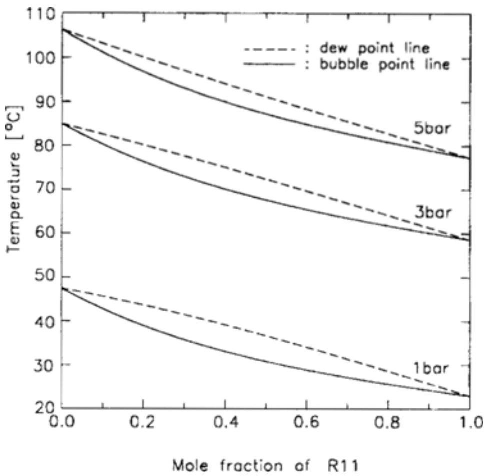


Fig. 4 Temperature equilibrium phase diagram of R11/R113 mixtures at different pressure.

The range of physical variables for the reported test results was as followed; a heat flux of 10-80 kW/m²,a mass flux of 250-1,000 kg/m²s, and a composition of R11, such as 0.25, 0.50 and 0.75. Inlet temperatures were usually less than the saturation temperature by 10-15℃.

5. Experimental Results and Discussions

5.1 General Aspects

Figures 5, 6, and 7 show the typical behavior which characterize the flow boiling phenomena obtained in this experiment. The onset of nucleate boiling(ONB), which accompanies the abrupt wall temperature drop can be clearly seen in Fig. 5.For low mass flux the temperature drop experienced at this point is as much as 15℃.However this temperature drop is not significant for the mass flux greater than 500 kg/m²s. As expected the heat transfer coefficient begins to increase at this point approaching a constant level in the saturated boiling region. At the level of heat flux of 80 kW/m²,the saturated boiling region is achieved very rapidly, as shown in Fig. 7.

The bulk temperature slightly decreases along the channel because of pressure drop for pure liquid. However the bulk temperature is found to be almost constant along the tube for the mixtures of X=0.5 as shown in Fig. 6.It should be noted

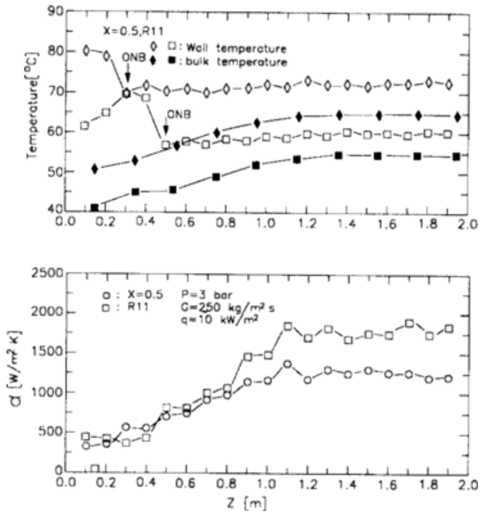


Fig. 5 Axial variations of wall temperature, bulk temperature(above) and heat transfer coefficient(below) for the case showing ONB.

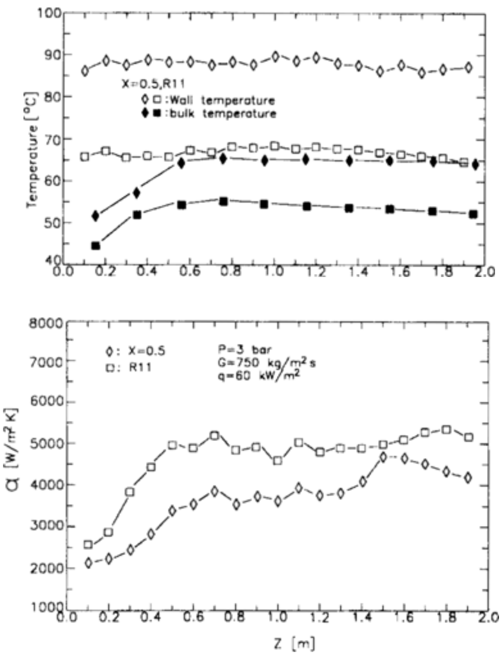


Fig. 6 Axial variations of wall temperature, bulk temperature(above) and heat transter coefficient(below) for the case of nucleate boiling.

that the bulk temperature measured was always slightly less than the equilibrium temperature corresponding to the pressure at the inlet. For the case of heat fluxes, of 40~80 kW/m² with high mass flux of 1000 kg/m²s, the temperature differ-

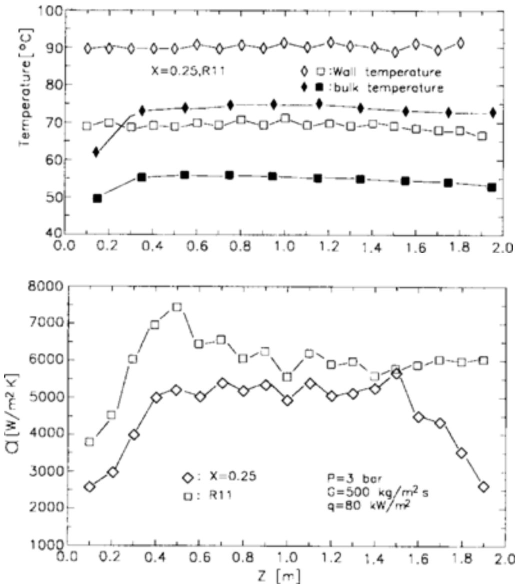


Fig. 7 Axial variations of wall temperature, bulk temperature(above) and heat transfer coefficient(below) for the case showing ONB (R11) and the Dryout(X=0.25).

ence between the measured and the calculated one was about 3℃, which means that the thermodynamic equilibrium could not be achieved for such case. Thermodynamic equilibrium occurred only for the moderate heat flux with low mass flow rate. Even superheated state was observed for the high heat flux with low mass flow rate. Thus, the bulk temperature obtained with the equilibrium assumption may yield erroneous results in the measurement of the covective boiling heat transfer coefficients.

With moderate mass flux (500 kg/m^2s) and high heat flux, the local heat transfer coefficient experiences rapid increase at ONB for R11, as shown in Fig. 7.This is due to the fact that slight drop of the wall temperature at this transition (ONB) results in the large increase in the heat transfer coefficient. In this figure, the dryout point where the wall temperature increases abruptly and the heat transfer coefficient decreases correspondingly is clearly seen for X=0.5.

Figure 8 shows that the local heat transfer coefficient increases with heat flux, while the coefficients are insensitive on the quality. This behavior is observed for the low mass flux case

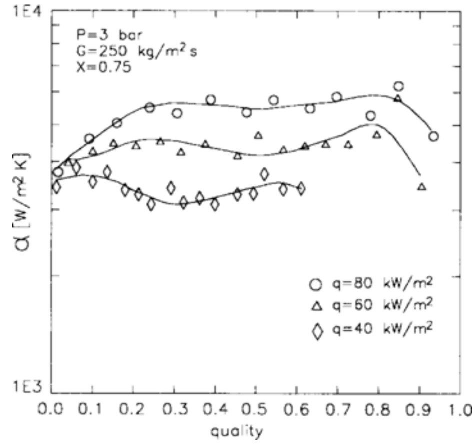


Fig. 8 The effect of quality and heat flux on heat transfer coefficient for the low mass flux flow at 3 bar.

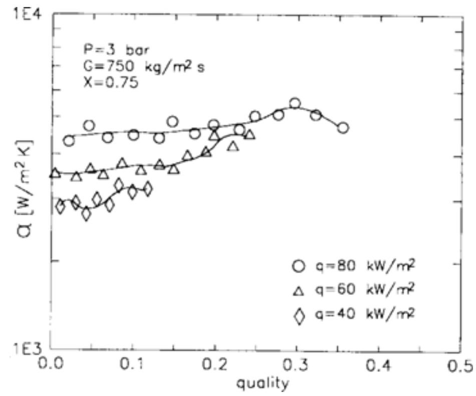


Fig. 9 The effect of quality and flux on heat transfer coefficient for the low mass flux flow at 3 bar.

where nucleate boiling prevails. However for the case of high mass flux, the curves representing each heat flux case merge into a single curve as the quality increases as shown in Fig. 9.This phenomena occur in the region where nucleate boiling is fully suppressed and the two-phase forced convection becomes dominant. Such distinct behavior which demarcates the boiling behaviour between the low mass flux and high one was also observed at the ambient pressure of 5 bar. However the heat transfer coefficients obtained at 5 bar is greater than the ones obtained at 3 bar at low quality. This difference in the heat transfer coefficients disappears at high quality.

Figure 10 shows the effect of mixture composi-

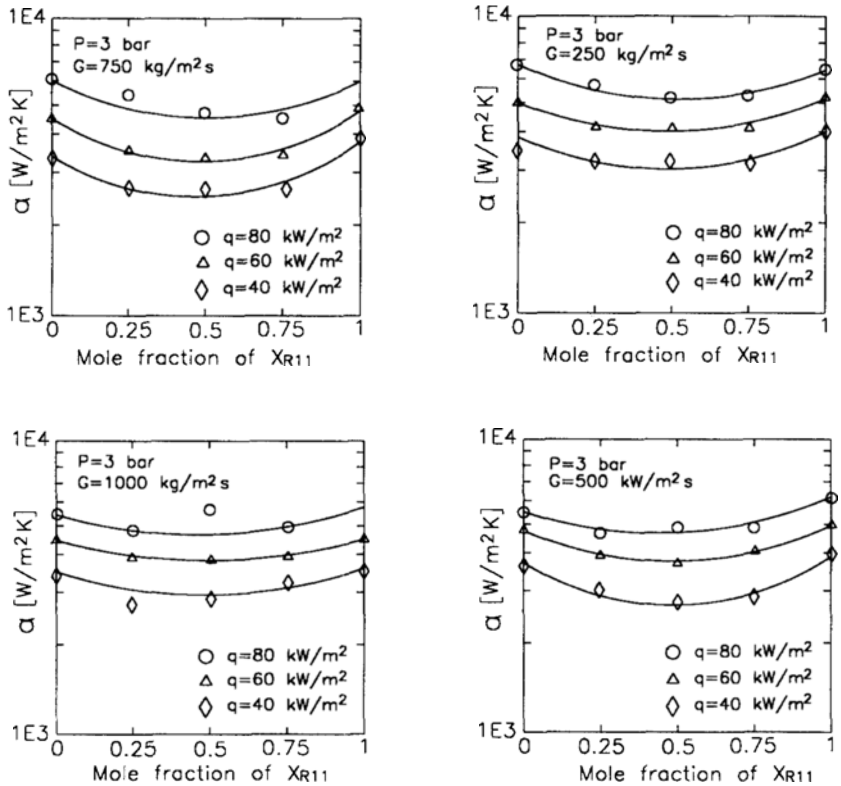


Fig. 10 Effect of muxture composition of flow boiling heat transfer coefficient with various mass fluxes as at 3 bar.

tion on the heat transfer coefficients fitted by the least square method. As observed in pool boiling, the heat transfer coefficients of mixture are always lower than the ones based on linear interpolation between the value of two pure components. Note that the heat transfer coefficient used in this figure is the average value of the local ones available.

As seen in Fig. 11, the measured heat transfer coefficients dependence on the Lockhart-Martinelli parameter (Whalley, 1987) are shown in a nondimensional form where X_{tt} denotes the parameter which is given as

$$X_{tt}=\left(\frac{1-x_m}{x_m}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \qquad (3)$$

At the high mass flow rate where convective boiling prevails, the heat transfer coefficients are shown to be less scattered even at low qualities.

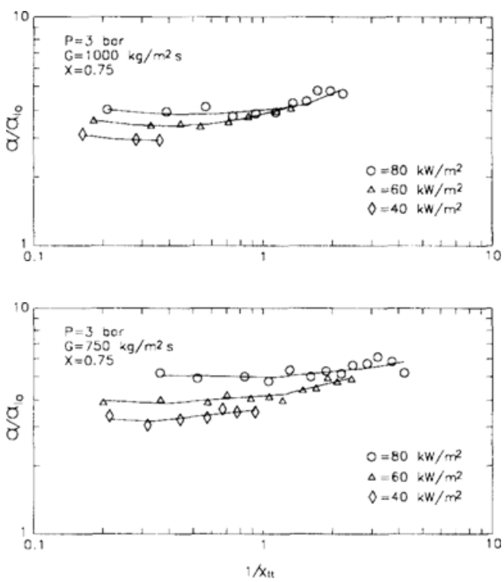


Fig. 11 Flow boiling heat transfer results for mixture of X=0.75 using dimensionless parameters.

5.2 Measured heat transfer coefficients compared with the Chen's correlation.

Comparing the measured data of the heat transfer coefficients to the Chen's correlation which includes various possible physical effects involved in the heat transfer mechanism is employed in this study. For binary mixture, this correlation may be written as

$$\alpha_c=\alpha_{LO}\left[\frac{\left(\frac{dp}{dz}\right)_{TP}}{\left(\frac{dp}{dz}\right)_l}\right]^{0.444} f(\mathrm{Pr}_l)\left[\frac{\Delta T}{\Delta T_s}\right]_{mac} + \alpha_B S_{Binary}(Re_{TP}) \tag{4}$$

where

$$\alpha_{LO}=0.023\frac{k_l}{d}\left[\frac{(1-x_m)Gd}{\mu_l}\right]^{0.8}\mathrm{Pr}_l^{0.4} \tag{5}$$

$$\alpha_B=0.00122\frac{k_l^{0.79}Cp_l^{0.45}\rho_l^{0.49}}{\sigma^{0.5}\mu_l^{0.29}h_{fg}^{0.24}\rho_v^{0.24}}(\Delta Ts)^{0.24}(\Delta Ps)^{0.75} \tag{6}$$

$$S_{Binary}(Re_{TP})=\frac{S}{1-\frac{Cp_l(y_m-x_m)}{h_{fg}}\frac{\partial T}{\partial x_m}\left[\frac{\alpha_T}{D}\right]^{\frac{1}{2}}} \tag{7}$$

$$f(Pr_l)=Pr_l^{0.296} \tag{8}$$

$$\left[\frac{\Delta T}{\Delta Ts}\right]_{mac}=1-\frac{(1-y_m)q}{\rho_l h_{fg} h_m \Delta Ts}\frac{\partial T}{\partial x_m} \tag{9}$$

$$S=\frac{1}{1+2.53\times10^{-6}Re_{TP}^{1.17}} \tag{10}$$

$$h_m=0.023(Re_{TP})^{0.8}(Sc)^{0.4}\frac{D}{d} \tag{11}$$

and

$$Re_{TP}=Re\left(f(Pr_l)\left[\frac{\left(\frac{dp}{dz}\right)_{TP}}{\left(\frac{dp}{dz}\right)_l}\right]^{0.444}\right)^{1.25} \tag{12}$$

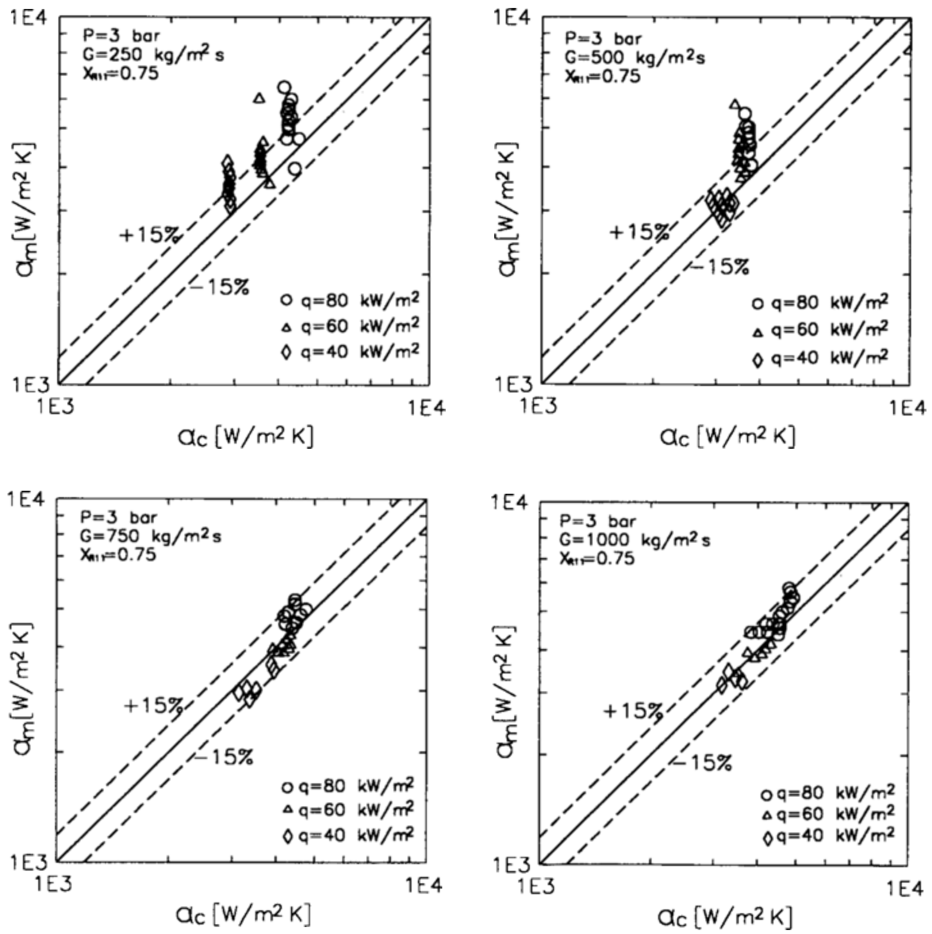


Fig. 12 Comparison between measured and predicted heat transfer coefficients by Chen's Correlation with various mass fluxes for mixture of X=0.75 at 3 bar.

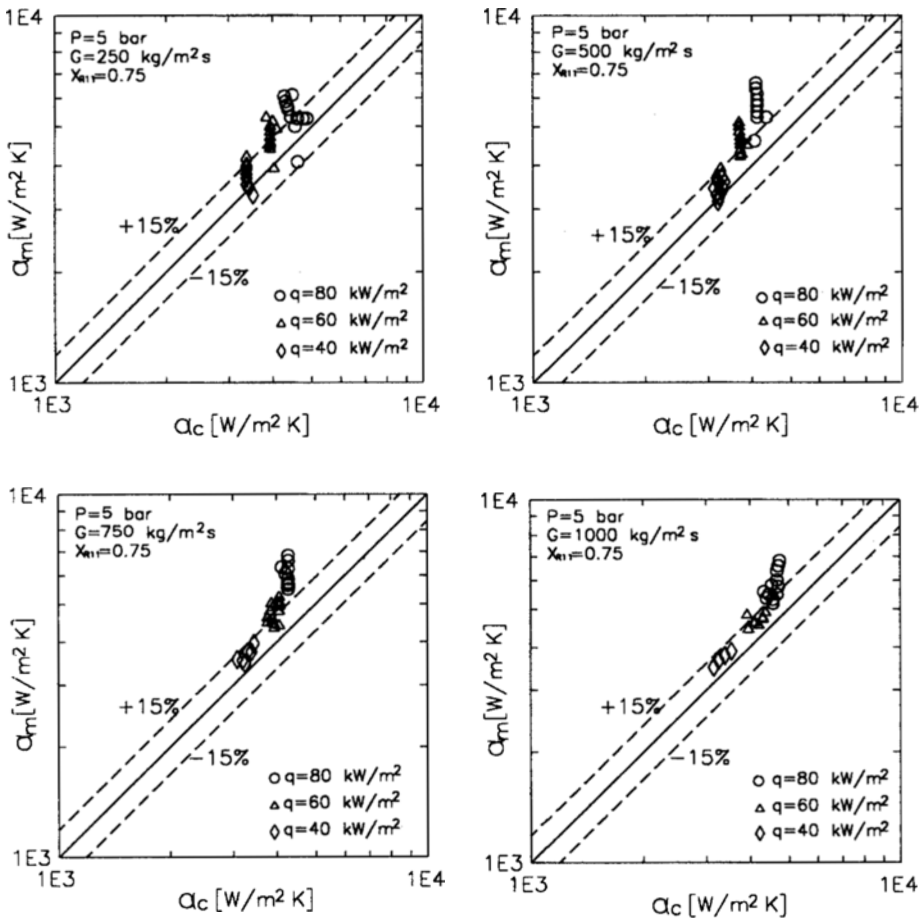


Fig. 13 Comparison between measured and predicted heat transfer coefficients by Chen's correlation with various mass fluxes for mixture of X=0.75 at 5 bar.

For pure liquid case, the following type of Chen's correlation is used

$$\alpha_C = \alpha_{LO} F + \alpha_B S \tag{13}$$

with

$$F = Pr_l^{0.296}\left[\frac{\left(\frac{dp}{dz}\right)_{TP}}{\left(\frac{dp}{dz}\right)_l}\right]^{0.444} \tag{14}$$

and

$$Re_{TP} = Re_l F^{1.25} \tag{15}$$

Friedel correlation (Whalley) for the two phase multiplier which is needed to estimate F has been used.

In order to examine the specific physical process governing the heat transfer of binary mixtures, the heat transfer data obtained at given mass flux were compared with the Chen's correlation. As can be clearly seen in Fig. 12, the Chen's correlation gives good prediction for the high mass rate flow (>750 kg/m²s) where two-phase forced convection is dominant over nucleate boiling. In this case, the calculated heat transfer coefficients are in well agreement with the measured data within $\pm 15\%$. Taking account the uncertainty in the measurement of the heat transfer coefficients, the observed data may be used within the error bound of $\pm 28\%$. However for the case of low mass rate flow (≤ 500 kg/m²s) where nucleate boiling prevails, the measured data are higher than the calculated one data by the Chen's corre-

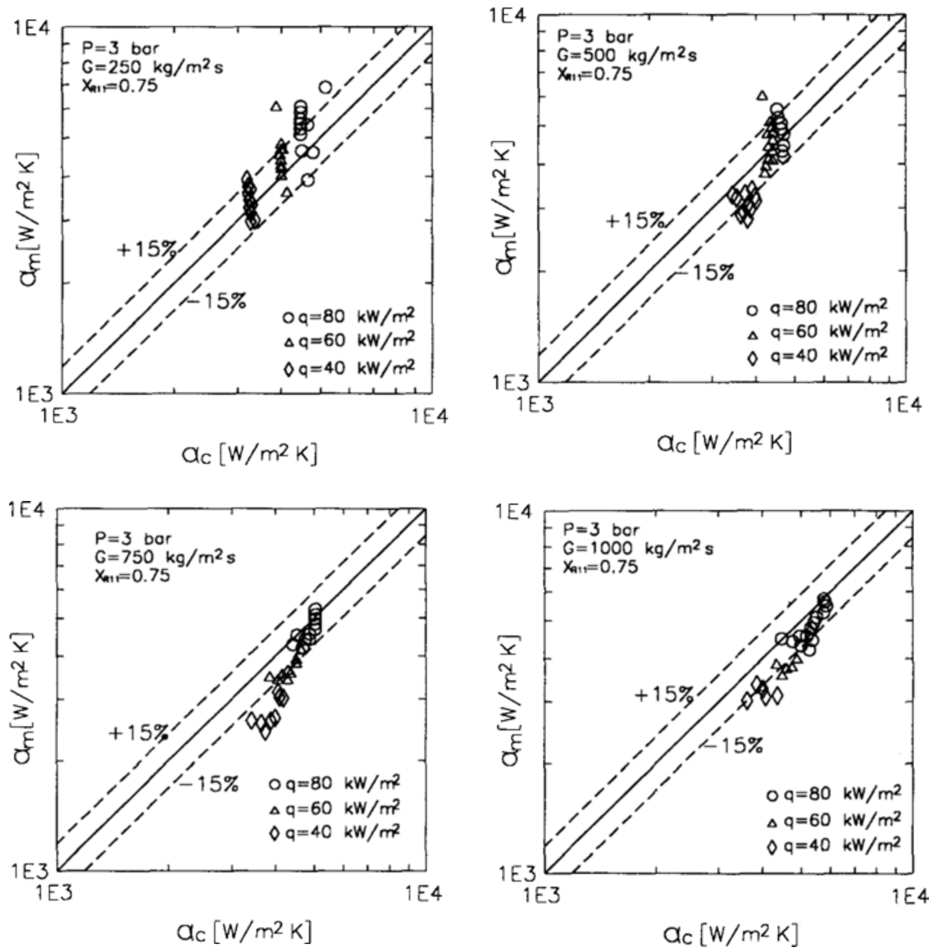


Fig. 14 Comparison between measured and predieted heat transfer coefficients by Chen's correlation without mass transfer resistance term with various mass fluxes for mixture of X=0.75 at 3 bar.

lation for any case. Same behavior was observed at 3 bar. It was found that the Chen's correlation failed to predict rather high heat transfer coefficient at ONB and Dryout point mentioned above. Also the Chen's correlation did not cover the case of the increase in the heat transfer coefficient due to the increase of the ambient pressure as can be seen in Fig. 13.

It was also tried to figure out how the mass transfer resistance term in the Chen's correlation influenced the heat transfer coefficient. Figure 14 shows the comparison between the experimental data and the calculated one which deleted the mass transfer resistance term for the mixture of X =0.75.Surprisingly, all data set shifted to the right when one compared this with the data shown in Fig. 12.This shift was observed in the other mixtures with different composition of R11, which turned out to be overprediction of the heat transfer coefficient. This observation confirms that the mass transfer resistance term $[\Delta T/\Delta T_S]_{mac}$ is very important factor to determine the heat transfer coefficient of binary mixtures. On the other hand, the binary suppression factor does not play any decisive role in determining the heat transfer coefficient of mixtures.

6. Summary

This experimental research was focused on determining experimental heat transfer coefficients for both pure and mixture refrigerants of

R11 and R113.The results are summarized as follows. It is observed that there exist marked difference in boiling behavior depending on mass flux. For the low mass flux flow where nucleate boiling phenomena is dominant, the ONB which accompanies abrupt wall temperature drop occurs. It is also found that Chen's correlation is inappropriate to predict the heat transfer coefficient for this case. On the other hand, the Chen's correlation is quite successful to predict the heat transfer coefficient for the high mass flux flow in which two phase convection prevails. The temperature drop at ONB is small for the high mass flux flow. It is also found that the mass transfer resistance term in the Chen's correlation for mixture is very important factor to determining the heat transfer coefficient. While the suppression factor for mixture boiling does not play any decisive role for the calculation of the coefficient. The average heat transfer coefficient of mixtures is found to be lower than those linear interpolations between the values of the two pure components, as in the pool boiling cases. Hardly the equilibrium case is observed. Thus equilibrium assumption may lead errors in the measurement of the convective boiling heat transfer coefficients.

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