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# EXPERIMENTAL STUDY ON R-134A CONDENSATION HEAT TRANSFER CHARACTERISTICS IN PLATE AND SHELL HEAT EXCHANGERS

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## ABSTRACT

In this study, condensation heat transfer experiments were conducted with two types of plate and shell heat exchangers (P&SHE) using R-134A. An experimental refrigerant loop has been established to measure the condensation heat transfer coefficient of R-134A in a vertical P&SHE. Two vertical counter flow channels were formed in the P&SHE by stacking three plates of geometry with a corrugated trapezoid shape of a chevron angle of 45 degree. The R-134A flows down in on channel exchanging heat with the cooling water flowing up in the other channel. The effect of the refrigerant mass flux, average heat flux, saturation temperature and vapor quality of R-134A on the measured data were explored in detail. The present data showed that two types of P&SHE have similar trends. The condensation heat transfer coefficients increase with the vapor quality for both types of P&SHE. A rise in the refrigerant mass flux causes an increase in the *hr*. Also, a rise in the average heat flux causes an increase in the *hr*. Finally, at a higher saturation temperature the *hr* is found to be lower. Correlation is also provided for the measured heat transfer coefficients in terms of the Nusselt number.

## NOMENCLATURE

A: heat transfer area of the plate
b: channel spacing
$c_p$: specific heat
$d_h$: hydraulic diameter
$i_{fg}$: enthalpy of vaporization
m: mass flow rate
Re: Reynolds number
x: vapor quality

## Subscript

fg: difference between liquid and vapor phase
i, o: at inlet and exit
lat, sens: latent and sensible heats
p: pre-heater
r: refrigerant
w: water

## INTRODUCTION

The plate and shell heat exchangers (P&SHE) are different from the conventional plate heat exchanger. The plates that have oblique line grooves are circular, and stacked together in contrary arrangements, which are enclosed a cylindrical shell. Operating temperature rises up to 350□, and pressures can be achieved to 10 MPa. Although P&SHE is apparently different from the conventional rectangular plate exchanger, the underlying flow passage structure through the exchanger is the same as in the conventional plate heat exchanger. The P&SHE is being introduced to the refrigeration and air conditioning systems as evaporators or condensers for their high efficiency and compactness.

Due to the serious destruction of the ozone layer in the atmosphere by the CFC refrigerants, various new refrigerants such as R-134A, R-143A, and R-125 were recently developed and introduced into the refrigeration and air conditioning systems. The heat transfer data for these new refrigerants are very scarce. In order to set up the database for using the new refrigerants in the design of the plate pattern for the P&SHE, condensation of R-134A flow in a P&SHE is experimentally investigated in this study.

In this study, the characteristics of the condensation heat transfer for refrigerant R-134A flowing in the plate and shell heat exchangers were explored experimentally

## EXPERIMENTAL APPARATUS AND METHOD

The heat transfer plates and experimental system established to study the condensation of R-134A are schematically shown in Fig. 1 and 2. The experimental system was constituted with test section, refrigerant loop, water loop and a data acquisition unit. Refrigerant R-134A is circulated in the refrigerant loop. In order to obtain different test conditions of R-134A including vapor quality, system pressure and imposed heat flux in the test, we need to control the temperatures and flow rates of the working fluids in the water loop and current in the pre-heater.

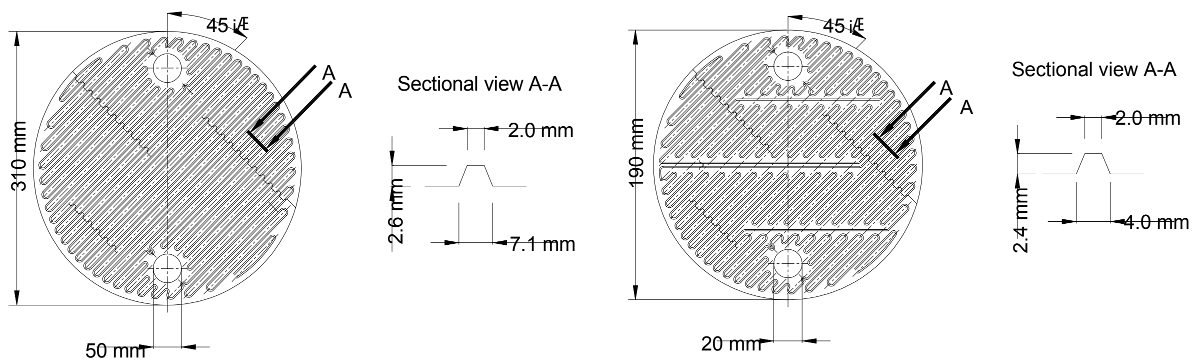



Figure 1: Schematic diagram of plate and shell heat exchangers.

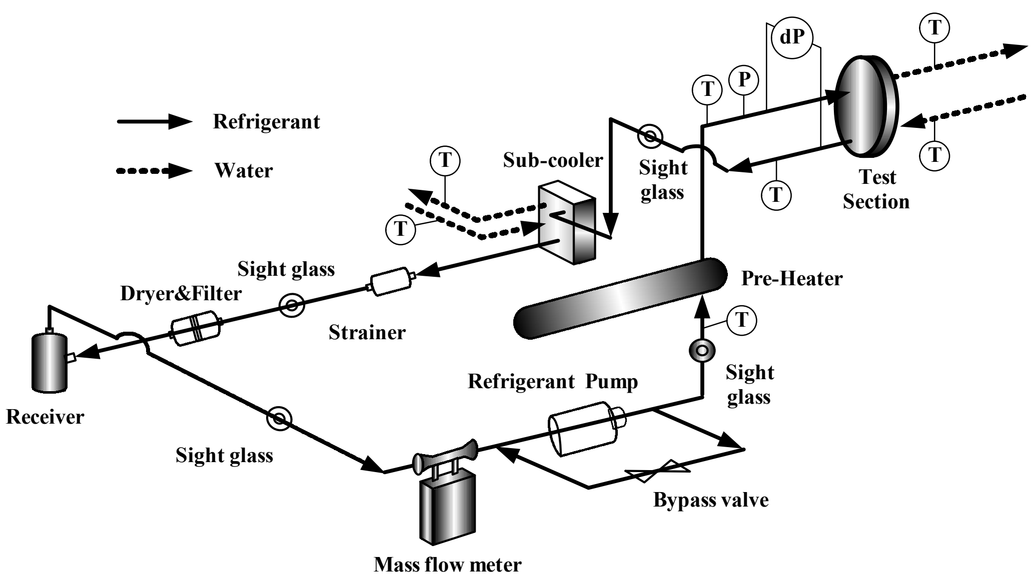



Figure 2: Schematic diagram of the experimental system.

## Test section

The plate and shell heat exchangers used in this study were formed by three commercialized SUS-304 plates. The plate surfaces were pressed to become grooved with a corrugated trapezoid shape and 45 deg of chevron angle. The corrugated grooves on the right and left outer plates have an oblique shape but those in the middle plate have a contrary oblique shape on both sides. Due to the contrary oblique shapes between two neighbor plates the flow streams near the two plates cross each other in each channel. This cross flow creates a significantly unsteady and random flow. In fact, the flow is highly turbulent even at low Reynolds number.

## Refrigerant loop

The refrigerant loop contains a refrigerant pump, a pre-heater, a test section (P&SHE), a sub-cooler, a strainer, a refrigerant mass flow meter, a dryer/filter, and four sight glasses. The refrigerant pump is a magnetic pump (TUTHILL-DDS 1.2) driven by a DC motor that is, in turn, controlled by a variable DC output motor controller. The variation of the liquid R-134A flow rate was controlled by a rotational DC motor through the change of the DC current. The refrigerant flow rate was measured by a mass flow meter (Oval, D040S-SS-322) installed between the receiver and refrigerant pump with an accuracy of ± 0.2%. The pre-heater is used to evaporate the refrigerant to a specified vapor quality at the test section inlet by transferring heat from the electrical heater to R-134A. Note that the amount of heat transfer from the electrical heater to the refrigerant in the pre-heater is calculated from the power meter (YOKOGAWA WT110). The dryer/filter intends to filter the solid particles possibly present in the loop. Meanwhile, a sub-cooler was used to condense the refrigerant vapor from the test section by a cold water to avoid cavitations at the pump inlet. The pressure of the refrigerant loop can be controlled by varying the temperature and flow rate of the water loop in the sub-cooler and current in the pre-heater. After condensed, the liquid refrigerant flows back to the receiver.

## Water loop for test section

The water loop in the condensation experimental test system designed for circulation cold water through the test section contain a 200 liter constant temperature water bath with a 5 kW heater and an air cooled refrigerant unit of 1 RT cooling capacity intending to accurately control the water temperature. A 0.5 hp water pump controlled by inverter is used to drive the cold water to test section with a specified water flow rate. The accuracy of measuring the water flow rate by the flow meter (Oval, D040S-SS-200) is ± 0.2%.

## Water loop for sub-cooler

The water loop designed for the two-phase flow the R-134A vapor contains 200 liter constant temperature bath with a water both with a 5 kW heater and an air cooled refrigeration unit of 3 RT cooling capacity intending to accurately control the water temperature. Then, a 0.5 hp water pump is also used to drive the cool water at a specified water flow rate to the sub-cooler.

## Data acquisition

The data acquisition unit includes a 20 channel Fluke NetDAQ 2645A recorder combined with a personal computer. The recorder was used to record the temperature and voltage data. The pressure and differential pressure transducer need a power supply as a driver to output an electric voltage of 0-10 V. The NetDAQ 2645A recorder allows the measured data to transmit to personal computer and then to be analyzed by the computer immediately.

# EXPERIMENTAL PROCEDURES

In each test pressure of the refrigerant loop can be controlled by varying the temperature and flow rate of the water loop in the sub-cooler and current in the pre-heater. The vapor quality of R-134A at the test section inlet can be kept at the desired value by adjusting the voltage of the voltage transformer for the pre-heater. Finally, the heat transfer rate between the counter flow channels in the test section can be varied by changing the temperature and flow rate in the water loop for the test section. Any change of the system variables will lead to fluctuations in the temperature and pressure of the flow. It takes about 60-120 min to reach a statistically steady state at which variations of the time-average inlet and outlet temperatures are less than 0.1 and the variations of the pressure and heat flux are within 1% and 5%, respectively. Then the data acquisition unit is initiated to scan all the data channels for 30 times in 5 min. The mean values of the data for each channel are obtained to calculate the heat transfer coefficient.

Before examining the condensation heat transfer characteristics, the preliminary experiments for single phase water convection in the plate and shell heat exchangers were performed. The modified Wilson's method [1] was adopted to calculate the relation between single phase heat transfer coefficient and flow rate from these data. This single phase heat transfer coefficients can then be used to analyze the data acquired from the two-phase heat transfer experiments.

## DATA REDUCTION

An analysis in needed in the present measurement to deduce the heat transfer rate from the water flow to the refrigerant flow in the test section. From the definition of the hydraulic diameter, Shah and Wanniarachchi [2] suggested to use two times of the channel spacing as the hydraulic diameter for plate heat exchangers when the channel width is much larger than the channel spacing. So we follow this suggestion.

$$d_h \approx 2b \tag{1}$$

The procedures to calculate heat transfer coefficient of the refrigerant flow are described in the following. The total heat transfer rate between the counter flows in the test section is calculated from the hot water side

$$Q_w = m_{w,c} c_{p,w} (T_{w,c,o} - T_{w,c,i}) \tag{2}$$

Then, the refrigerant vapor quality entering the test section is evaluated from the energy balance for the pre-heater. While the heat transfer to the refrigerant in the pre-heater is the summation of the sensible heat transfer (for the temperature rise of the refrigerant to the saturated value) and latent heat transfer (for the evaporation of the refrigerant).

$$Q_p = Q_{sens} + Q_{lat} \tag{3}$$

where

$$Q_{sens} = m_r c_{p,r} (T_{r,sat} - T_{r,p,i}) \tag{4}$$

$$Q_{lat} = m_r i_{fg} x_{p,o} \tag{5}$$

The above equations can be combined to evaluate the refrigerant quality at the exit of pre-heater that is considered to be the same as the vapor quality of the refrigerant entering the test section. Specifically,

$$x_i = x_{p,o} = \frac{1}{i_{fg}} \left( \frac{Q_p}{m_r} - c_{p,r} (T_{r,sat} - T_{r,p,i}) \right) \tag{6}$$

The change in the refrigerant vapor quality in the test section is then deduced from the heat transfer to the refrigerant in the test section,

$$\Delta x = \frac{Q_w}{m_r \cdot i_{fg}} \tag{7}$$

The average quality in the test section is given as

$$x_{ave} = x_m = x_i - \frac{\Delta x}{2} \tag{8}$$

The overall heat transfer coefficient $U$ between the two counter channel flows can be expressed as

$$U = \frac{Q_w}{A \cdot LMTD} \tag{9}$$

The log mean temperature difference (LMTD) is again determined from the inlet and exit temperatures in the two channels,

$$LMTD = \frac{(\Delta T_1 - \Delta T_2)}{\ln(\Delta T_1 / \Delta T_2)} \tag{10}$$

where

$$\Delta T_1 = T_{r,o} - T_{w,c,i} \tag{11}$$

$$\Delta T_2 = T_{r,i} - T_{w,c,o} \tag{12}$$

with $T_{r,i}$ and $T_{r,o}$ being the saturation temperature R-134A corresponding respectively to the inlet and outlet pressures in the refrigerant flow in the P&SHE. Finally, the condensation heat transfer coefficient in the flow of R-134A is evaluated from the equation

$$\left(\frac{1}{h_r}\right) = \left(\frac{1}{U}\right) - \left(\frac{1}{h_{w,c}}\right) - R_{wall}A \tag{13}$$

where the Modified Wilson method was applied to calculate $h_{w,c}$.

## RESULTS AND DISCUSSION

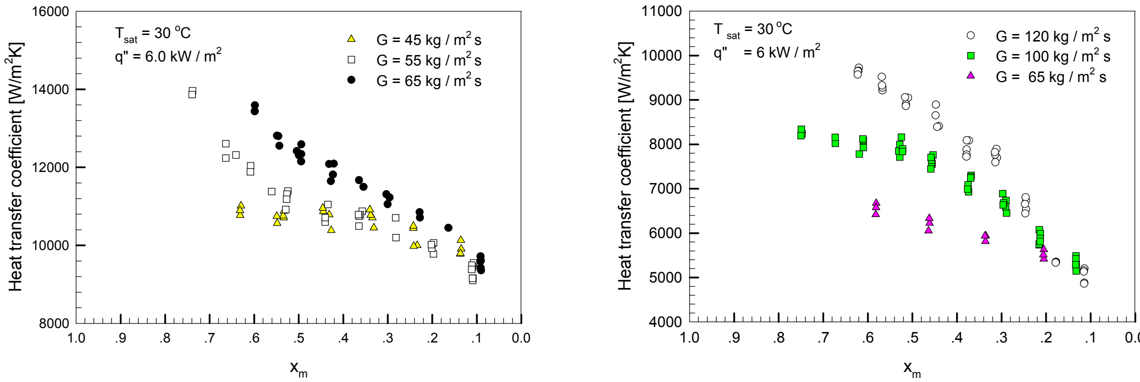
In the present investigation of the R-134A condensation in the P&SHE the R-134A mass flux was varied from 45 to 120 kg/m$^2$s, average imposed heat flux from 6.0 to 8.0 kW/m$^2$ and saturation temperature from 30 to 40℃. The measured heat transfer coefficient are to be presented in terms of their variations with their average vapor quality in the test section, since the P&SHE is small and has only three plates, the vapor quality change in the test section is small, $\Delta x < 0.06$.

Figure 3 shows the effect of the refrigerant mass flux on the measured R-134A condensation heat transfer coefficient at saturation temperature of 30℃ and an average imposed heat flux of 6.0 kW/m$^2$ for the mass flux ranging from 45 to 65 kg/m$^2$s in type A, 65 to 120 kg/m$^2$s in type B and the mean vapor quality varying from 0.1 to 0.8. The mean vapor quality $x_m$ is the average vapor quality in the P&SHE estimated from $x_i$ and $\Delta x$. The results show that in the vapor quality region for $x_m > 0.25$ the heat transfer coefficients increase linearly with the mass flux. This increase is rather significant. For instance at 100 kg/m$^2$s the condensation heat transfer coefficient at the quality $x_m$ of 0.75 is about 53.5% larger than that at 0.13. This obviously results from the simple fact that at a higher $x_m$ the liquid film on the surface was thinner and the condensation rate is thus higher. But at a low quality with $x_m > 0.21$ the heat transfer coefficient is only slightly affected by the mass flux. Note that at a low quality the vapor flow is slow. For the flow the velocity of liquid induced from the shear force associated with the low vapor flow is limited. Thus, the differences in the condensation rates for different mass fluxes are limited. [3]

The effect of the average imposed heat flux on the condensation heat transfer is shown Fig. 4 by presenting the heat transfer data for two heat fluxes of 6.0 kW/m$^2$ and 8.0 kW/m$^2$ at G = 55 kg/m$^2$s in type A, G = 100 kg/m$^2$s in the B and $T_{sat}$ = 30℃. It is well known that the condensation rate would be proportional to the heat flux. The results indicate that at a given vapor quality the heat transfer coefficient is higher for a higher heat flux except at a higher vapor quality with $x_m > 0.65$ in type A and $x_m > 0.52$ in type B. Note that the R-134A quality-averaged condensation heat transfer coefficients at 8.0 kW/m$^2$ are about 10% larger than 6.0 kW/m$^2$. However, compared with the mass flux effects shown in Fig. 5, the heat flux has a smaller effect on the condensation heat transfer coefficient in the higher vapor quality regime. Again at a certain $x_m$, a rise in condensation heat transfer coefficients with the mean vapor quality weakened for each heat flux. Note that these changes in condensation heat transfer coefficients appear for $x_m$ around 0.45 for each case. This results from the fact that horizontal corrugation in the P&SHE suppresses vapor velocity in the higher vapor quality regime.

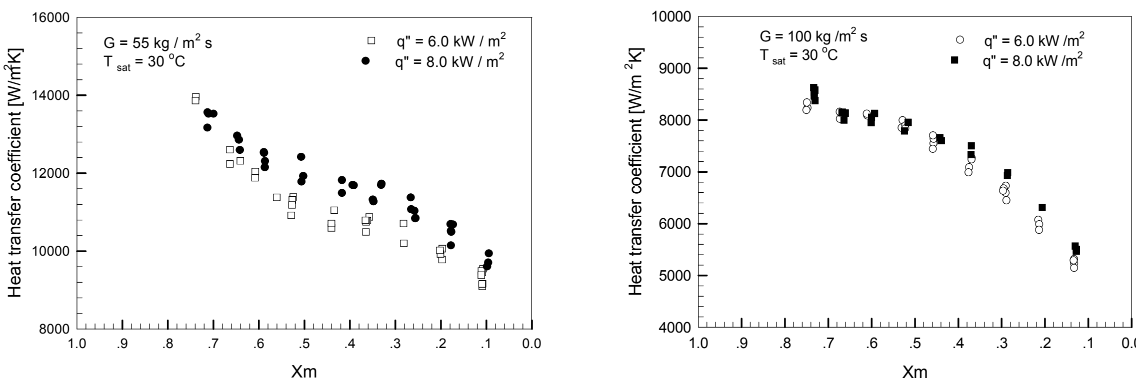
The effects of the refrigerant saturation temperature (pressure) on the condensation heat transfer coefficient is illustrated in Fig. 5 by presenting the data for two typical cases at G = 55 kg/m$^2$s in type A, G = 100 kg/m$^2$s in type B and $q_w''$ = 6.0 kW/m$^2$ at different mean vapor qualities for $T_{sat}$ ranging from 30 to 40℃. The results suggest that at a given saturation temperature the condensation heat transfer coefficient rises significantly with the mean vapor quality. While at a fixed $x_m$, the condensation heat transfer coefficient is poorer at a higher $T_{sat}$ in the total quality region. Specifically, the mean heat transfer coefficient at 30℃ is about 30 to 40% bigger than that at 40℃. Compared with the heat flux, the saturation temperature has an effect on the condensation heat transfer coefficient in the total vapor quality regime. Note that at a high vapor quality the vapor moves at a higher speed and the interfacial condensation is higher. On the other hand, at a higher saturation temperature the thermal conductivity of the liquid R-134A is lower. Especially, the conductivity of liquid film reduces about 6.13% for the R-134A saturation temperature raised from 30 to 40℃. The associated thermal resistance of the liquid film is larger, causing a poorer heat transfer rate.

It is necessary to compare the present data for the R-134A condensation heat transfer coefficient in the P&SHE to those in plate heat exchanger reported in the literature. Due to the limited availability of the data for plate heat exchanger with the same ranges of the parameters covered in the present study.



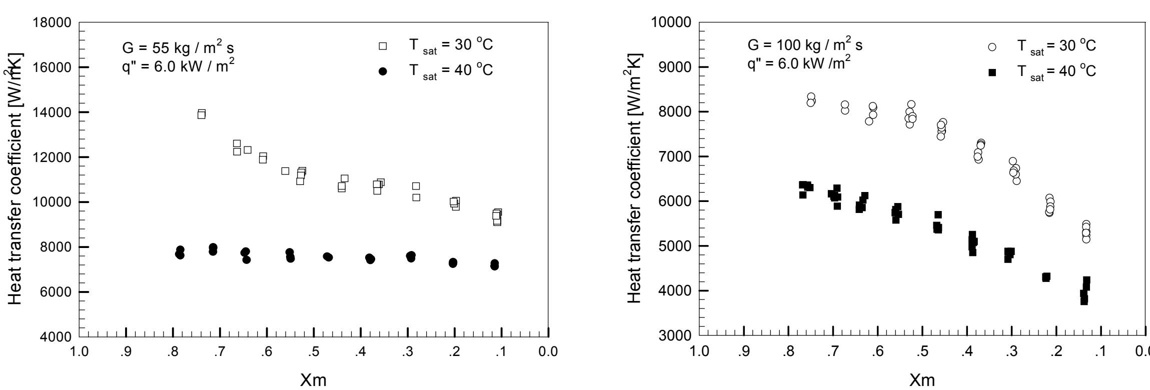

Type (A) Type (B)

Figure 3: Variations of condensation heat transfer coefficient with mean vapor quality for various mass fluxes at $T_{sat}$= 30℃ and $q_w$"=6.0 kW/m$^2$.




Type (A) Type (B)

Figure 4: Variations of condensation heat transfer coefficient with mean vapor quality for two heat fluxes at G= 55 kg/m$^2$s and $T_{sat}$ = 30 ℃, Type A and G = 100 kg/m$^2$s and $T_{sat}$ = 30 ℃, Type B




Type (A) Type (B)

Figure 5: Variations of condensation heat transfer coefficient with mean vapor quality for saturation temperature at G = 55 kg/m$^2$s and $q_w$" = 6.0 kW/m$^2$, Type A and G = 100 kg/m$^2$s and $q_w$" = 6.0 kW/m$^2$, Type B.

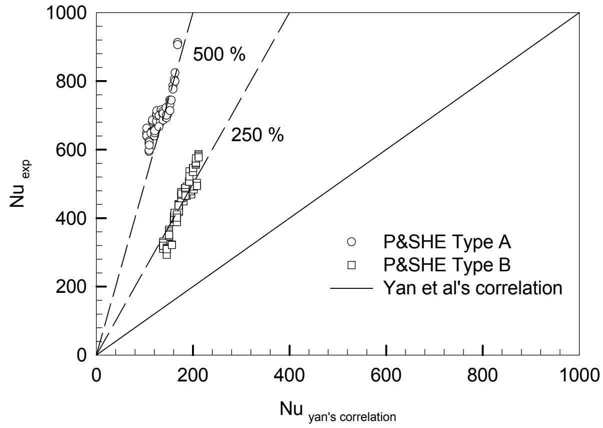
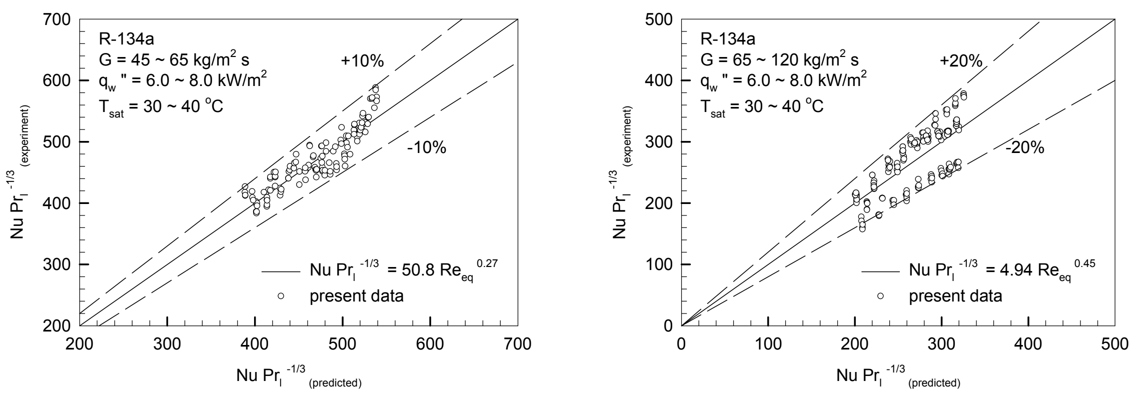



Figure 6: Comparison of the present heat transfer data with those for plate heat exchanger from Yan et al(1999).




Type (A) Type (B)

Figure 7: Comparison of the proposed correlation for Nusselt number with the present data, Type A and Type B.

The comparison is only possible for a few cases. This is illustrated in Fig. 6, in which our data are compared with correlation of Yan et al. Note that the data from Yan et al. [4] are average condensation heat transfer coefficient measured in a plate heat exchanger with the vapor quality from 0.08 to 0.86. Yan et al. proposed condensation heat transfer correlation equation such as

$$Nu = h_r \frac{D_h}{k_l} = 4.118\,\mathrm{Re}_{eq}^{0.4} \tag{14}$$

$\mathrm{Re}_{eq}$ is the equivalent Reynolds number. $\mathrm{Re}_{eq}$ is defined as

$$\mathrm{Re}_{eq} = \frac{G_{eq} D_h}{\mu_l} \tag{15}$$

in which

$$G_{eq} = G\left[1 - x_m + x_m\left(\frac{\rho_l}{\rho_\upsilon}\right)^{1/2}\right] \tag{16}$$

Here $G_{eq}$ was proposed by Akers et al. [5] and is an equivalent mass flux which is a function of the R-134A mass flux, mean quality and densities at the saturated condition. The comparison clearly shows that the R-134A condensation heat transfer coefficient for P&SHE is about 5 times (Type A) and 2.5 times (Type B) in average higher than that for the plate heat exchanger. To facilitate the use of the plate and shell heat exchanger as a condenser, correlating equations for the dimensionless condensation heat transfer coefficient on the present data are provided. They are modified Yan et al's correlation

Type A :

$$Nu = h_r \frac{D_h}{k_l} = 4.94\,\mathrm{Re}_{eq}^{0.45} \quad 3000<\mathrm{Re}_{eq}<11000 \tag{17}$$

Type B :

$$Nu = h_r \frac{D_h}{k_l} = 450.8\,\mathrm{Re}_{eq}^{0.27} \quad 1800<\mathrm{Re}_{eq}<6500 \tag{18}$$

Figure 7 show the comparison of the proposed condensation heat transfer correlation to the present data, indicating that most of the experimental values are within ±10%(Type A) and ±20%(Type B).

## CONCLUSIONS

An experimental investigation has been conducted in the present study to measure the condensation heat transfer coefficient of R-134A in a plate and shell heat exchanger. The effects of the mass flux of R-134A, average imposed heat flux, saturation temperature and vapor quality of R-134A on the measured data were examined in detail. The results show that the condensation heat transfer coefficients normally increase with the refrigerant mass flux. Contrary to the mass flux effects, the heat flux does not have significant effects on the heat transfer at high quality, but at a high wall heat flux it shows some influences at low vapor quality. The increase in the saturation temperature results in a lower heat transfer coefficient in the total vapor quality regime. Correlations were also proposed for the measured heat transfer coefficients in terms of Nusselt number.

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