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An experimental investigation of condensation heat transfer coefficient using R-410A in horizontal circular tubes.

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Abstract

Condensation heat transfer coefficient has been evaluated experimentally on the tube side of three different circular tubes with inner diameter of 6.61, 7.5 and 9.2mm, respectively. Two-phase fluid flow conditions include mass fluxes from 200 to 320kg/m^{2}s, qualities between 0.1 to 0.9, and heat flux range from 5 to 20kW/m^2 at a fixed saturation temperature of 48^{o}C. Results showed that the average heat transfer coefficient increased with the increase of vapor quality, mass flux and heat flux, but decreased with inner diameter. The experiment results are compared with the existing heat transfer coefficient correlations, and a new correlation is developed with good prediction.

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Keywords: heat transfer; coefficient; R410A; condensation; correlation

1. Introduction

R-410A is a near-azeotropic refrigerant and widely used in refrigerator, air conditioning and heat pump systems. The heat transfer efficiency of evaporator and condenser plays an important role in increasing of the COP of system and the minimum of its size in air conditioning and heat pump system. Recently, air conditioning system use R-410A have been developed and sold on the market. There are some studies on the horizontal flow condensing of hydrocarbon and mixtures refrigerant.

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Kim et al. [1] performed the condensation heat transfer of R22 and R410A in 9.52mm O.D. horizontal tubes. The data were measured for the mass flow rate of 40-80kg/h, constant heat flux of 11kW/m^2 and the saturation temperature at 45^{0}C. They concluded that the average condensation coefficient of R22 and R410A for microfin tubes were 1.7-3.19 and 1.7-2.94 times larger those in smooth tubes, respectively. Wijaya and Sparz [2] evaluated experimental condensation heat transfer of R410A in a horizontal smooth tube and compared the results with R22 data under the same experimental condition. They proposed that the condensation heat transfer coefficients and pressure drop for R410A were 2-6% higher and 25-45% lower, respectively, than those with R22. Dobson and Chato [3] performed an experimental study on the condensation of R12, R134a, R410A and R32/125 (60/40%) in horizontal smooth tubes with various ID of 3.14, 4.57, and 7.04 mm. They presented two correlations to predict the heat transfer coefficient in the stratification and annular flow regime, respectively. Jung et al. [4] investigated the flow condensation experiments using R134a, R407C, and R410A inside horizontal smooth and microfin tubes of 9.52 mm O.D. at the saturation temperature of 40 °C with mass fluxes of 10, 20, and 30 kg/m^2 s and a heat flux of 7.7– 7.9 kW/m^2. They concluded that R134a showed lower heat transfer coefficients with the smooth tube while R407C and R410A showed lower heat transfer of R134a, R125, R236a, R32 and R410A coefficients with the microfin tube. Cavallini el al [5] reported the experimental heat transfer coefficient during condensation of R134a, R125, R236a, R32 and R410A inside a smooth tube. The results were evaluated at the following conditions: mass velocities varying from 100 to 750kg/m2s, saturation temperature ranging between 30-50^{0}C and vapor quality range from 0.15 to 0.85. They suggested the Kosky and Staub model [6], Jaster and Kosky equation [7] and Friedel model [8] to predict the Nusselt number of annular and stratified flows and frictional pressure drop, respectively.

This study investigated the heat transfer during condensation of R410 in horizontal tubes of 6.61, 7.45 and 9.2mm I.D. The effects of mass flux, heat flux and inner diameter tube on heat transfer coefficient were reported. The experiment results are compared with existing heat transfer coefficient correlations and the new correlation is developed. The results from the current study will make an important contribution to understanding of heat transfer mechanisms during condensation in circular tubes. The proposed model may be used by engineers for analyzing and manufacturing the condenser in air-conditioning and heat pump systems.

2. Experimental apparatus and test conditions

The experimental apparatus is schematically shown in the Fig.1. The refrigerant loop is designed to measure the heat transfer characteristic of refrigerant. It is mainly composed of a receiver, a magnetic gear pump, a sub-cooler, a Coriolis flow meter, a pre-heater, a test section and a condenser. The refrigerant flow rate can be controlled by a magnetic gear pump and the vapor quality of refrigerant was controlled by a pre-heater before it entered the test section. The test section is obtained by one 1.45m long tube and the effective heating length is 1.2m with four separated 0.3m long PVC jacket, as illustrated in Fig.2. For condensing at the test section, a certain heat flux was applied heat from cooling water in the PVC jacket. The heat transfer rate in the test section can be varied by adjusting the temperature and flow rate of the cooling water.

Thermocouples and pressure transducers were installed at the inlet and outlet of the test section. To measure the wall temperature of tube, twelve thermocouples were installed on the outer surface of the tube at the four locations along the length of the tube, mount at the top position, the side position and bottom position of the tube in the square section. The test sections were insulated with foam to minimize heat transfer between test section and the environment. The energy balance between inside and outside of the test tube was checked using water at the proper heat flux, and the results showed that was less than 3% of the total heat transfer

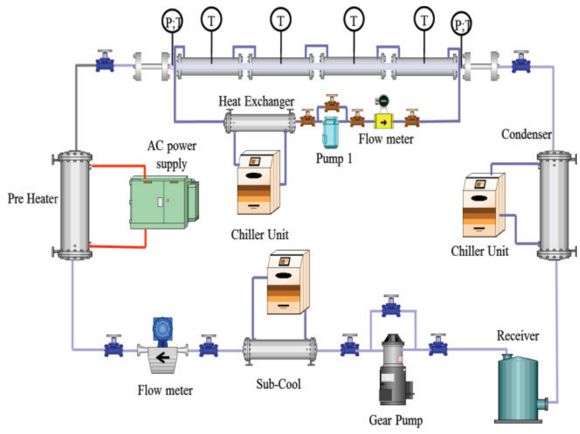

Fig. 1 Schematic diagram of experimental apparatus

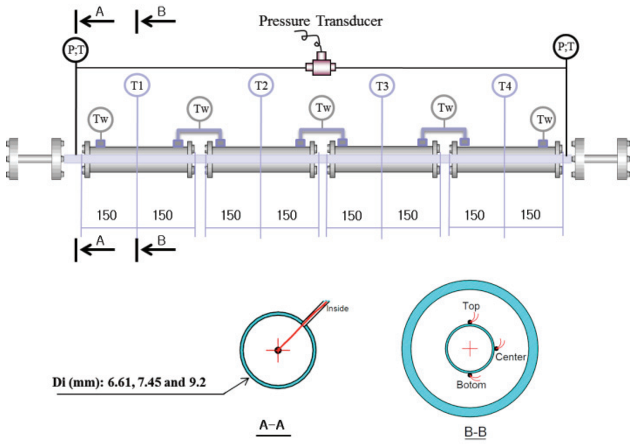

Fig. 2 Schematic diagram of test ections

The water temperature was measured by eight thermocouples installed in the center water flow of inlet and outlet for each subsection, respectively. The thermocouples are set up at the inlet and outlet of each subsection and the average measurement values are used to represent the water temperatures at those points.

The physical properties of R-410A using in the study are all calculated by REFPROP version 8.0. All signals of temperatures, pressure drop and mass flow rate were collected using a data acquisition system in real time with a PC running the data reduction program. The test conditions showed in the Table 1.

3. Data reduction

The local flow condensation heat transfer coefficient as obtained as:

$$h = \frac{Q}{A\left(T_{sast} - T_{w,i}\right)} \tag{1}$$

Where h is the heat transfer coefficient for two-phase refrigerant, $Wm^{-2}k^{-1}$. Q is the heat flow rate exchanged in the test tube, W. A is the exchange surface area, m^2. T_{sat} and T_{wall} are the saturation and inside tube wall temperature of the refrigerant, respectively. The heat transfer removed from the water cooling Q was calculated as follows:

$$Q = m_{water} c_p \Delta T \tag{2}$$

Where C_p is the special heat of the cooling water, m_{water} is the water cooling mass flow rate and ΔT is temperature difference between the inlet and outlet of the test section.

The vapor quality x at the inlet of the test section was defined from Eq. (3)

$$x_{in} = \frac{1}{i_{fg}}\left[\frac{Q_p}{m_r} - C_{p,r}\left(T_{sat} - T_{p,in}\right)\right] \tag{3}$$

Here, Q_p is the heat supplied to the pre-heater and $T_{p,in}$ is the refrigerant temperature entered the pre-heater.

4. Result and discussion

4.1. Condensation heat transfer coefficient

Experimental tests were carried out during condensation of R410Awith the mass velocity ranged from 200 to 320kgm^{-2}s^{-1} and the saturation temperature of 48°C. The experimental values of the heat transfer coefficient are plotted in Fig. 3. The data show that the condensation heat transfer coefficient increase with increasing mass velocity and vapor quality. The effect of mass flux on the heat transfer coefficient suggests that the contribution of forced convective heat transfer is dominant.

The effect of heat flux on the heat transfer coefficient at heat fluxes 9 and 12kWm^{-2} inside 6.61 and 9.2mm ID are presented in Fig.4, respectively. The results show that no effect of heat flux was observed in this study. The similar phenomenon could be found in many experimental tests used tube-in-tube heat exchanger. Zhang et al. [9] reported the effect of heat fluxes on the heat transfer coefficient are not strong and not clearly at the low mass flux (300kgm^{-2}s^{-1}). Shin and Kim [10] reported that there is no the relationship between the heat flux and condensation heat transfer.

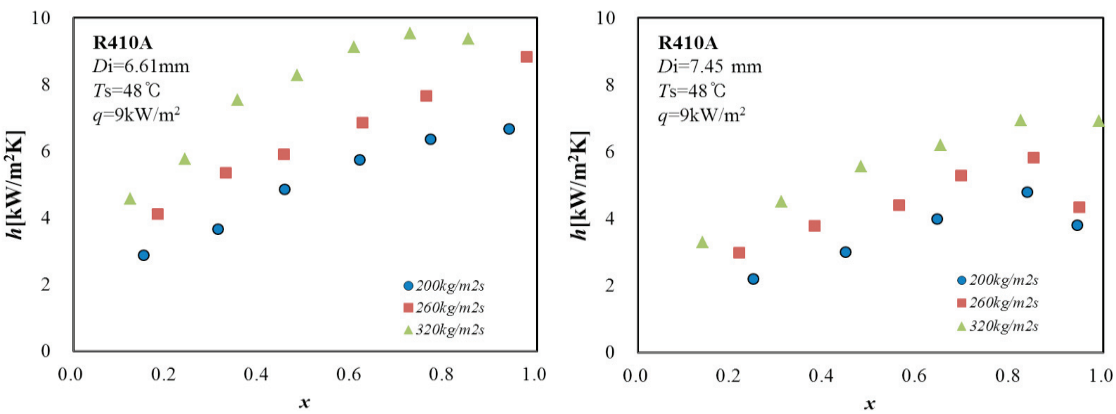

Fig. 3. The effect of mass flux on the heat transfer coefficient

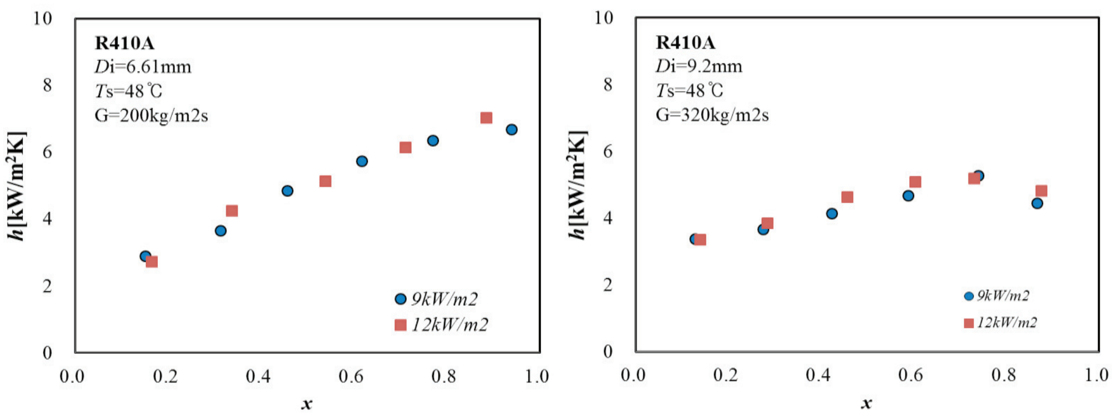

Fig. 4. The effect of heat flux on the heat transfer coefficient

Table 1. Test conditions

Refrigerant	R-410A
Inner diameter (mm)	6.61, 7.45, 9.2
Condensing temperature (°C)	48
Heat flux (kW/m2s)	9, 12
Mass flux (kg/m2s)	200, 260, 320
Quality range	0.1-0.9

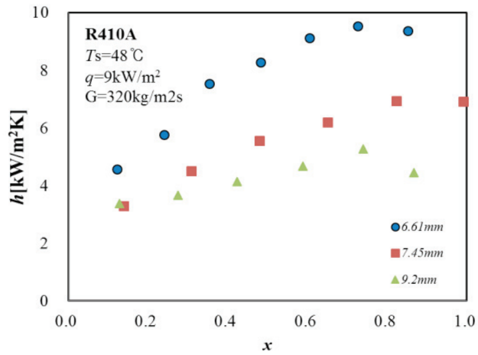

Fig. 5 The effect of inner diameter on the heat transfer coefficient

The effect of inner diameter on the condensation heat transfer coefficient was shown in Fig.5. It is seen that the heat transfer coefficient increased when the inner diameter decreased. The tube diameter effected on the flow regime transitions. As the tube diameter decreased, the transition from wavy flow to wavy-annular flow and the transition from wavy-annular flow to annular flow shifted to lower qualities [3].

4.2. Developing correlation

The experimental heat transfer coefficients in our study were compared with several widely used practical researches developed on experimental data with ID larger than 3mm, specially the Shah [11], Cavallini et al. [5], Thome et al. [12] correlations. Among them, the correlation developed by Shah shows the best prediction with the experimental data.

Based on the numerical prediction of the heat transfer for problem defined in Fig. 2, it was attempted to develop a new correlation for heat transfer coefficient. This new correlation is a form of modified Shah [11] correlation

$$h = 0.023\,\mathrm{Re}_{lo}^{0.8}\,\mathrm{Pr}_l^{0.4}\,\frac{k_l}{d_i}\left[(1-x)^{0.8} + \frac{3.8}{\mathrm{Pr}^{0.52}}\left(\frac{x}{1-x}\right)^{0.201}\right]$$

Where $\mathrm{Re}_{lo} = \dfrac{GD}{\mu_l}$; $\mathrm{Pr}_l = \dfrac{\mu_l c_{pl}}{k_l}$; $\mathrm{Pr} = \dfrac{P_{sat}}{P_{critical}}$

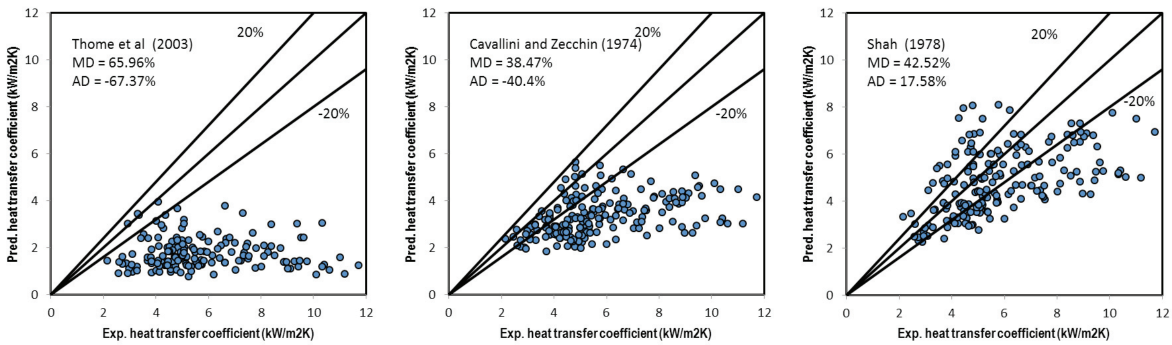

Fig. 6 Comparison experimental data with existing correlations

5. Conclusion

Condensation heat transfer coefficients of R410A measured inside 6.61, 7.45 and 9.2mm I.D. were reported in this paper. The experimental heat transfer coefficient has been obtained by directly measuring the wall temperature. The results were summarized as follows:

1) The heat transfer coefficient of R401A during condensation tended to increase with increasing vapor quality, mass flux, but tended to increase with decreasing inner diameter. The effect of heat flux on condensation heat transfer coefficient is negligible.
2) A heat transfer coefficient correlation was made by modifying Shah [11] correlation based upon the experimental data. The correlation showed the good agreement with the mean and the average deviation of 14.6% and -3.1%, respectively.

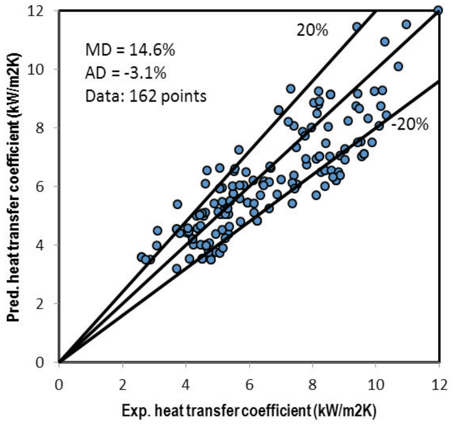

Fig. 7. Comparison of experimental heat transfer coefficient with new correlation

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Biography
Jong-Taek Oh is currently a Professor at the Department of Refrigeration and Air Conditioning Engineering, Chonnam National University at Yeosu, South Korea.
Research fields: boiling and condensation heat transfer experiment and simulation of a natural and alternative refrigerants in small tubes and conventional tubes, heat pump system, refrigeration storage and transportation..