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Experimental Study on Condensation Heat Transfer Characteristics of R410A in Short Vertical Tubes

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An experimental study on condensation heat transfer of R410A in short vertical tubes (8.02 mm ID and 10.7mm ID) was presented. Experiments were performed in eight short copper tubes length varied from 300mm to 600mm at mass fluxes range of 58–246 kg $m^{-2}s^{-1}$ and saturation temperature of 38℃. Effects of mass flux, tube length on condensation heat transfer coefficient were investigated. The distribution of temperature, thickness of condensate film and local condensation heat transfer coefficient along the tube were also analyzed.. It is indicated that the entrance effect played an important role in condensation heat transfer of vertical tube, and the influence of entrance effect on average condensation heat transfer coefficients will weaken with the length of tube in the experimental condensation. The experimental results were compared with four well known correlations available in literatures, and the Chen correlation shows good agreement with the experimental data but with ±40% deviation. A new modified condensation heat transfer correlation with 12.7% mean deviation was developed to predict the condensation heat transfer coefficients in short vertical tube based on the experimental data.

Keywords: Condensation, Heat transfer characteristic, R410A, Short vertical tube, Entrance effect

Introduction

Improvement of condensation heat transfer rate for compact heat exchanger has been investigated for decades. The usage of micro-fin tubes has increased the heat transfer performance inside tubes and is one of the most efficient and common heat transfer enhancement techniques. Miyara et al.[1] observed the flow pattern of R123 at the exit of a herringbone tube and visually verified that the heat transfer was enhanced by the liquid removal and collection effect of the herringbone micro-fins. Numerous experimental studies have been done to measure heat transfer rate in small or mini tubes. Yan and Lin[2] compared the condensation rates of R134A in a pipe (2.0 mm ID) with some other published data, and they concluded that the averaged condensation heat transfer coefficient over the entire quality range tested for the small pipe was about 10% higher than that for the large pipe with d_i= 8.0mm. However, the way to improve the performance of condensation heat transfer, in principle, can be achieved by thinning the film thickness[3] or inducing the liquid film turbulence[4]. But few of them tried to improve the heat transfer performance by entrance effect. It is well known that condensate film in entrance region is thinner relatively with high heat transfer performance. The condensate increases gradually along the tube, even in the

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Nomenclature

c_p	specific heat capacity at constant pressure, J kg^{-1}K^{-1}	λ	thermal conductivity, W m^{-1}K^{-1}
d	diameter of tube, m	μ	dynamic viscosity, m^{2}s^{-1}
G	mass flux of R410A, kg m^{-2}s^{-1}	ρ	density, kg m^{-3}
g	gravitational acceleration, m s^{-2}	τ	shear stress, N m^{-2}
h	heat transfer coefficient, W m^{-2}K^{-1}	***Subscripts***	
L	tube length, m	*exp*	experimental
m	mass flow rate, kg s^{-1}	*i*	inner
p	pressure, Mpa	*in*	inlet
Pr	Prandtl number	*l*	liquid
Q	heat transfer rate, W	*o*	outer
Re	Reynolds number	*out*	outlet
r	latent heat of condensation, J kg^{-1}	*pre*	predicted
T	temperature, ℃	*re*	refrigeration
x	vapor quality	*sat*	saturation
z	coordinate along the tube direction, m	*to*	total
Greek letters		*v*	vapor
δ	film thickness, m	*w*	cooling water

downstream of the tube, and the wall is fully covered by condensate. As a result, condensation heat transfer rate was much lower than that in entrance region. Hence, reducing the length of tube can considerably improve the heat transfer performance because of the entrance effect, and it is also a reduction of the size and material of equipment. So it is worthwhile to study the characteristic of condensation heat transfer in short vertical tube.

There are few existing literatures investigated the performance of refrigerant condensation in vertical tube, especially in short tubes[5]. Oh and Revankar[6] performed a theoretical analysis on complete condensation in a vertical tube passive condenser, and the effect of mass transfer at the interface for the interfacial shear was considered. Dalkilic et al.[7-9] experimentally investigated average heat transfer coefficient and pressure drop of R134A condensation with vertical downward flow in a smooth tube (L=500mm). They discussed the effects of heat flux, mass flux and condensation temperature on the condensation heat transfer coefficient and emphasized that interfacial shear effect was significant for the laminar condensation heat transfer under the given conditions. Olivier et al.[10] presented a study on flow regimes, pressure drops, and heat transfer coefficients of refrigerant condensation in smooth and micro-fin tubes.

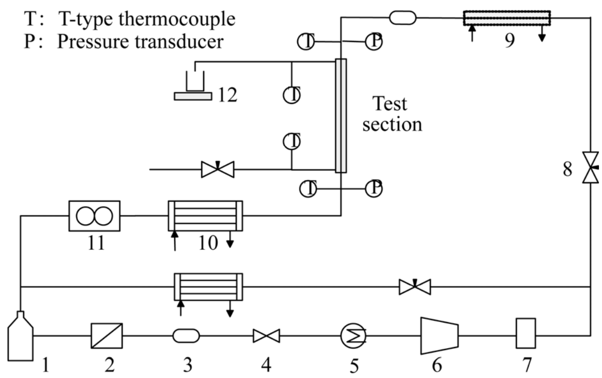
R410A, as a nearly azeotropic refrigerant with zero ODP (ozone depleting potential), and a kind of substitution of traditional refrigerants, particularly of R22, was used in this experiment. Cavallini et al.[11] experimentally measured the condensation heat transfer coefficients and pressure drops inside a smooth tube when operating with R134a, R125, R236ea, R32 and R410A. Oh and Son[12] experimentally investigated the condensation heat transfer coefficients of R22, R134a and R410A in a single circular micro-tube, and concluded that the condensation heat transfer coefficient of R410A was higher than that of R22 and R134a at the given mass flux.

The purpose of this study was to obtain experimental data of R410A in short vertical tube and search the suitable existing correlation to predict the condensation heat transfer characteristic of R410A in short vertical tube.

Experimental apparatus

The schematic diagram of the test system was showed in Fig.1. The experimental system was comprised of a refrigerant loop and a water cooling loop. The refrigerant loop was mainly consisted of an expansion valve, evaporator, compressor, test section, two condensers, and two condensation lines: the test line and the bypass line. Each of them had a control valve to control the refrigerant mass flux in the test line. Refrigerant vapor through the bypass line would be fully condensed into liquid by controlling the cooling water flow rate of the condenser, and then flowed into accumulator. The pre-condenser was used to control the inlet vapor temperature and quality of the test section. Cylindrical sight glass was positioned at the outlet of the pre-condenser and the fluid flow states could be observed. A band-type heater was wrapped around the side of the sight glass to compensate heat losses due to heat dissipation.

Test section was a tube-in-tube counter flow heat exchanger, with cooling water in the annulus and refrigerant vapor inside. The inner tube was made of copper with inner diameter of 8.02 or 10.7mm and length of 300, 400, 500 and 600mm. The outer wall temperature of inner tube was measured at four stations evenly arranged along the tube. At each station, two thermocouples with an accuracy of 0.1℃ were placed with 180 degree angle. The inlet and outlet temperatures of refrigerant were measured with two T-type thermocouples with an accuracy of 0.1℃. Two pressure transducers with an accuracy of ±0.25% were installed at the inlet and outlet of test section respectively with capillary piping to measure the pressure of vapor respectively. Post-condenser was used to fully condense the vapor flowed out from the test section into liquid and make sure that the Coriolis mass flow meter with an accuracy of ±0.2% can accurately measure the mass fluxes of refrigerant. Then test line combines with bypass line into accumulator.



1. Accumulator; 2. Solenoid valve; 3. Sight glass; 4. Expansion valve; 5. Evaporator; 6. Compressor; 7. Oil separator; 8. Control valve; 9. Pre-condenser; 10. Post-condenser; 11. Coriolis mass flow meter; 12. Electronic balance

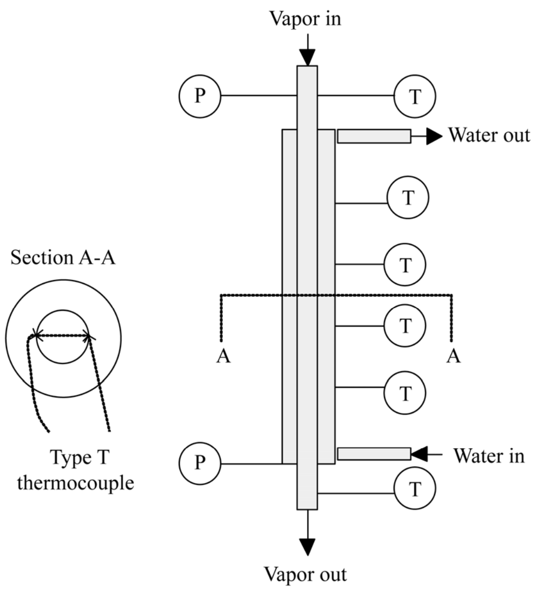


Fig. 1 Schematic diagram of the test system

Water cooling loop was used to cool the vapor flowing through the test section. In order to control the water flow rate, a valve was positioned at the inlet of the test section. The mass flow rate of cooling water was measured by electronic balance. The inlet and outlet temperature of cooling water were also measured using two T-type thermocouples with an accuracy of 0.1℃.

Data reduction

The experimental data collections were considered as relatively steady state during the condensation heat transfer experiments. The steady state condition of experiment was reached when the refrigerant inlet pressure (2.3Mpa), temperature (38℃), mass flow rate, water mass flow rate and temperature kept in stationary state. Each run took about 2 minutes at least. The inlet vapor of test section was kept saturated by controlling pre-condenser.

The average condensation heat transfer coefficient in the test section was obtained as:

$$h=\mathrm{Q}/(\pi d_i L\Delta T_{re,\ln}) \tag{1}$$

$$\Delta T_{re,Ln}=\frac{(T_{re,in}-T_{wall})-(T_{re,out}-T_{wall})}{ln\,(T_{re,in}-T_{wall})\big/(T_{re,out}-T_{wall})} \tag{2}$$

$$T_{wall}=T_{o,wall}+\mathrm{Q}\ln(\mathrm{d_o}/\mathrm{d_i})/2\pi\lambda L \tag{3}$$

Where $\Delta T_{re,ln}$ is the logarithmic mean temperature difference between the refrigerate vapor and inner wall of the tube. T_{wall} is arithmetic mean of the inner wall temperature calculated by the data of outer wall temperature.

The heat transfer rate of the exchanger, Q, was obtained with the mass flow rates and the temperature difference between inlet and outlet at the cooling water side:

$$\mathrm{Q}=\mathrm{m}_w c_{p,w}\left(T_{w,out}-T_{w,in}\right) \tag{4}$$

The outlet quality of R410A from the test section (the influence of film subcooling been taken into account) was calculated as:

$$x_{out}=1-Q/m_{re}r' \tag{5}$$

$$r'=r(1+3/8\,\mathrm{Ja})\ ^{[7]} \tag{6}$$

$$Ja=c_{p,re}(\mathrm{T_{sat}}-\mathrm{T_{wall}})/\mathrm{r} \tag{7}$$

Table 1 Ranges and fluctuations of experimental parameters

Parameter	Range	Fluctuations
$T_{re,sat}$	38℃	±0.5℃
G	58-246kg $m^{-2}s^{-1}$	±1.25%
m_w	57.06-58.61g/s	±1.5%
$T_{w,in}$	26.5℃	±0.25℃
x_{out}	0.84-0.90	±5%

Uncertainties

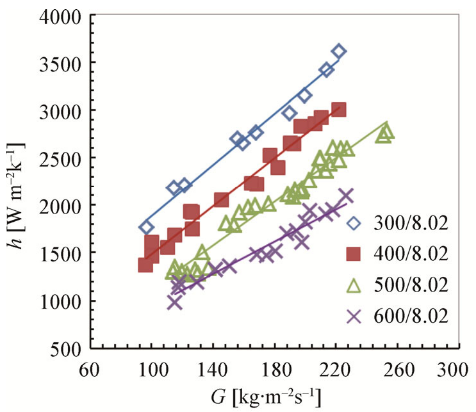
The ranges and fluctuations of experimental parameters were summarized in Table 1. The experimental uncertainties mainly came from the temperature on the outer wall of the inner tube. It led to an uncertainty of condensation temperature difference between saturate vapor and inner wall. Uncertainty induced by vapor quality was negligible because inlet vapor was almost kept as saturate according to control the vapor inlet temperature and pressure in the experimental condensation. The relative uncertainties of condensation heat transfer coefficients were calculated to be less than 15%, depending on the experimental conditions.

Results and discussion

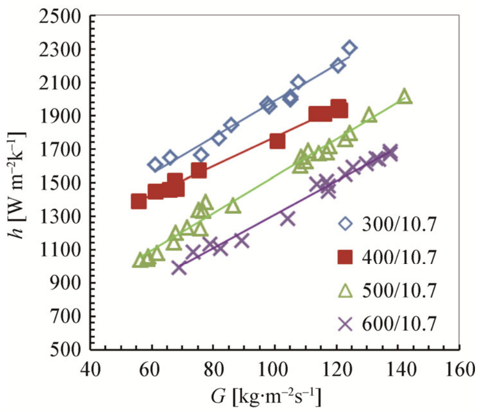
For each experimental condition, the heat transfer coefficients during condensation of R410A in vertical tube for different mass fluxes were obtained. The state of refrigerant at the inlet of test section was saturated at 38℃.

Effect of length on heat transfer coefficients

Experimental average condensation heat transfer coefficients for R410A were plotted over mass fluxes from 58 to 246 kg $m^{-2}s^{-1}$ with different length of vertical tube, shown in Fig.2. The effect of vapor quality on condensation heat transfer was neglected because of relatively little change in this experimental condition.



(a) L = 300, 400, 500, 600 mm; d_i = 8.02 mm



(a) L = 300, 400, 500, 600 mm; d_i = 10.7 mm

Fig. 2 Average condensation heat transfer coefficients vs. mass flux in different length tube

Fig.2 showed that condensation heat transfer coefficients almost increased linearly with mass fluxes for each tube. For the same diameter, the shorter the tube length, the larger the heat transfer coefficients. For d_i=8.02mm, the condensation heat transfer coefficients of 300mm increased by an average of about 19.26% than that of 400 mm, and 400mm increased by about 26.52% than 500 mm, 500mm increased by about 24.12% than 600mm. It can be explained by that the gravitational force and shear stress lead to an increase in the liquid velocity, and the entrance effect lead to a thinner film and thus to an increase of heat transfer coefficients. The shorter the tube length, the thinner the liquid film was at bottom of the tube. For d_i=10.7mm, the heat transfer coefficient of 300mm increased by an average of about 11.57% than 400mm, and 400mm increased by about 18.46% than 500mm, 500mm increased by about 16.70% than 600mm. However, the same results can be obtained whatever the inner diameter was, 400mm tube had the largest average growth rate of heat transfer coefficients. It indicated that entrance effect played an increasingly important role in condensation heat transfer of vertical tube with the mass flux for the length of tube less than 500mm. The influence of entrance effect on average condensation heat transfer coefficients would weaken with the length of tube, especially when L ≥ 500mm in the experimental condensation. So, it was advised that more experimental data should be gained to find a correlation considering entrance effect, which was similar to the single phase flow entrance region correlation.

Film thickness

According to the physical model of film condensation in the vertical tube and the assumptions of Dalkilic [7], annular film condensation heat transfer model of R410A in vertical tube was built. The interfacial shear stress was considered because it played an important role in short vertical tube.

The pressure drop was tested in this experiment and Fig. 3 showed the pressure drop of R410A in short vertical tube of the length of 300, 400, 500, 600 mm with 8.02mm inner diameter. It can be seen that pressure drop was so small that nearly was negligible.

According to Nusselt theory, the mass and energy balance equation of liquid film was simplified as:

$$\delta^4 + \frac{4}{3}\frac{\tau_\delta}{\rho_l g}\delta^3 - 4\frac{\mu_l \lambda_l \left(T_\delta - T_w\right) z}{\rho_l^2 g r'} = 0 \tag{8}$$

Where the interfacial shear stress τ_δ was defined by[18]:

$$\tau_\delta = A\frac{\mathrm{Re}_l^2}{\mathrm{Re}_{to}^{0.2}}\left(\frac{\mathrm{Re}_l}{\mathrm{Re}_{to}}\right)^{-1.6}\left(1.25 + 0.39\frac{\mathrm{Re}_l}{\mathrm{Re}_{to}}\right)^{-2} \tag{9}$$

$$A = \frac{0.252\mu_l^{1.177}\mu_v^{0.156}}{d_i^2 g^{2/3}\rho_l^{0.553}\rho_v^{0.78}} \tag{10}$$

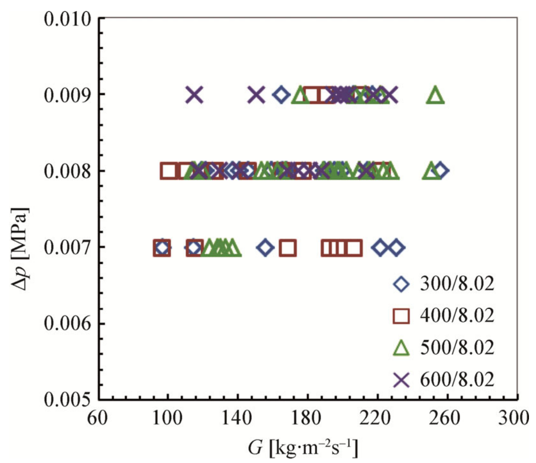


Fig. 3 Pressure drop of R410A in short vertical tube of the length of 300, 400, 500, 600 and 8.02 mm inner diameter

The condensate flow rate can be calculated as:

$$m_l = \pi d_i \rho_l \frac{\delta^2}{\mu_l}(1/3\rho_l g\delta + \frac{\tau_\delta}{2}) \tag{11}$$

After giving the inner wall temperature distribution and boundary conditions, the change of film thickness along the tube can be calculated by Eq. (8). If a linear temperature distribution in the film region was assumed, the local film heat transfer coefficient can be expressed as:

$$h_z = \lambda_l / \delta_z \tag{12}$$

Fig. 4 showed the outer wall temperature curves of four different lengths of 300, 400, 500, 600mm (8.02 mm ID) at the mass fluxes of 114±1 kg $m^{-2}s^{-1}$ and 38 °C saturated vapor of R410A. The temperature decreased nonlinearly along the four tubes under the same boundary condition.

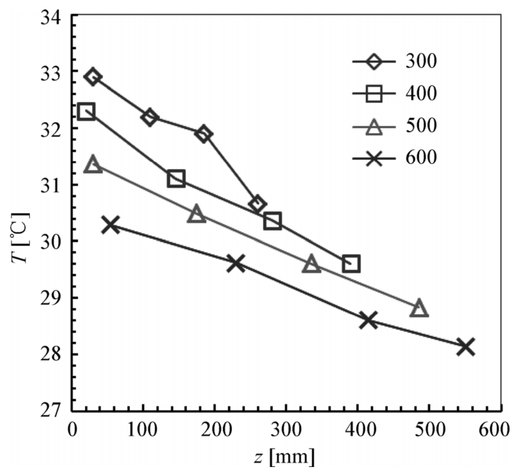


Fig. 4 Temperature distribution of four lengths of tubes of the length of 300, 400, 500, 600 mm and 8.02 mm ID at the mass fluxes of 114±1 kg $m^{-2}s^{-1}$

Fig. 5, 6, 7 showed the film thickness, liquid flow rate and heat transfer coefficients along the tube obtained from the Eq. (8), (11) and (12), respectively. The film thickness nearly linear increased and local heat transfer coefficient nonlinearly decreased along the tube. The condensation heat transfer coefficient was high at the entrance in which entrance effect played an important role. Then it decreased quickly along the tube, especially at the bottom of longer tube. The shorter the tube length, the thinner the film thickness and the larger the heat transfer coefficients. Even at the same location for each tube, the shorter tube had better condensation heat transfer performance. It also indicated that the temperature distribution along the tube had a very important role on condensation heat transfer performance. It was obvious that the tube with length of 300mm had a fairly good heat transfer performance mainly due to the effect of entrance effect. On the other hand, heat transfer coefficients decreased faster at the bottom of tube although driving

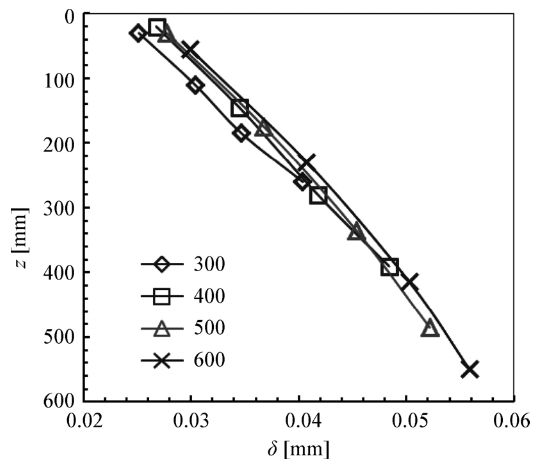


Fig. 5 Film thickness of four different lengths tubes at mass fluxes of 114±1 kg $m^{-2}s^{-1}$(L=300, 400, 500, 600 mm and 8.02 ID)

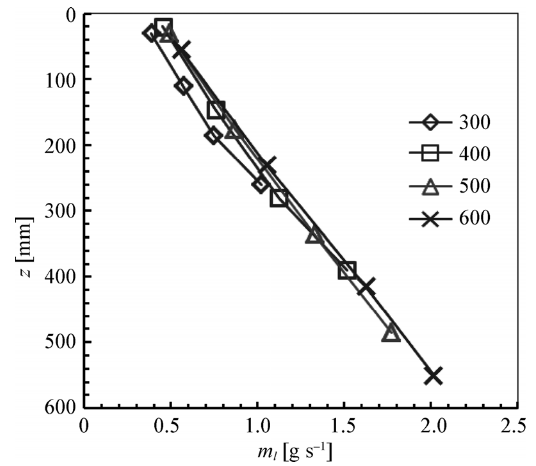


Fig. 6 Liquid flow rate of four different lengths tubes (L=300, 400, 500, 600 mm and 8.02 mm ID) at mass fluxes of 114±1 kg $m^{-2}s^{-1}$

temperature difference increased and condensing rate gradually decreased.

Figs. 8, 9, 10 and Fig. 11 showed the outer wall temperature, liquid flow rate, film thickness and condensation heat transfer coefficients for different temperature difference ($\Delta T_{re,ln}$) between the refrigerate vapor and inner wall at mass flux of 114±1 kg $m^{-2}s^{-1}$ (L=400mm, di=8.02mm), respectively. The two curves of temperature change along the tube had nearly the same trend (Fig. 8), and the average condensation heat transfer coefficient with temperature difference 4.6℃ was 4.4% larger than that with 6.7℃. It indicated that temperature difference, as the driving force, had a considerable effect on heat transfer performance.

Comparison with heat transfer correlations

Several in–tube condensation heat transfer correlations have been published in the literature[13]. Dalkilic et al[14]

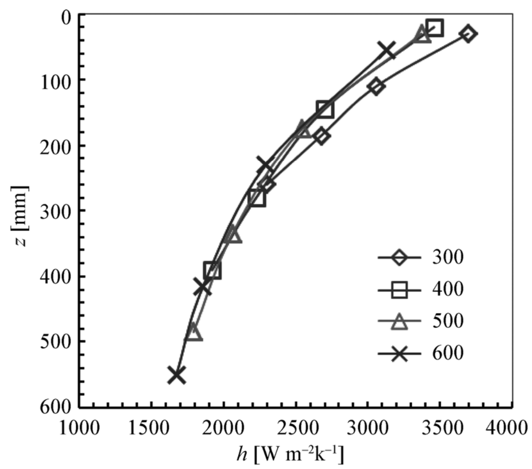


Fig. 7 Condensation heat transfer coefficients of four different lengths tubes at mass fluxes of 114±1 kg $m^{-2}s^{-1}$ (L=300, 400, 500, 600 mm and 8.02 ID)

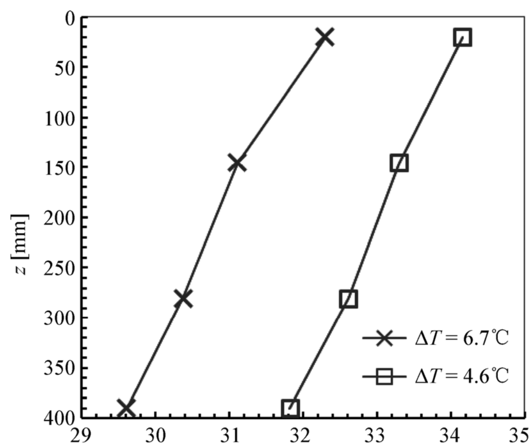


Fig. 8 Outer wall temperature distribution for different temperature difference at mass flux of 114±1 kg $m^{-2}s^{-1}$ (L=400mm, 8.02mm ID)

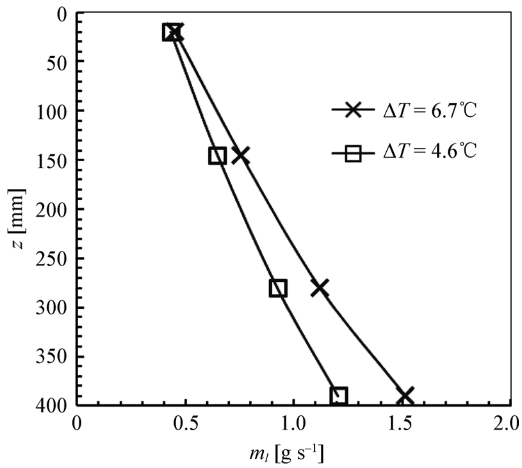


Fig. 9 Liquid flow rate for different temperature difference at mass flux of 114±1 kg $m^{-2}s^{-1}$ (L=400 mm, 8.02mm ID)

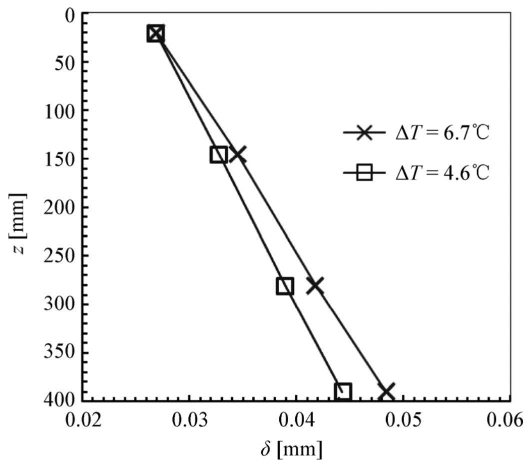


Fig. 10 Film thickness for different temperature difference at mass flux of 114±1 kg $m^{-2}s^{-1}$ (L=400 mm, 8.02mm ID)

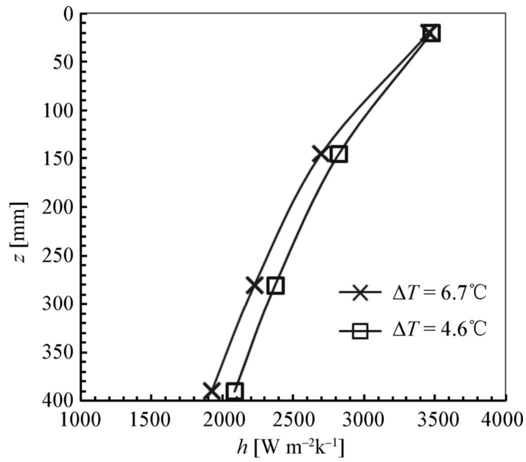


Fig. 11 Condensation heat transfer coefficients for different temperature difference at mass flux of 114±1 kg $m^{-2}s^{-1}$ (L=400 mm, 8.02 mm ID)

compared eleven well-known correlations for annular flow using a large amount of data obtained from various experimental conditions and stated that annular flow models were independent of tube orientation.

Four correlations were selected to compare with the experimental data:

(1) Shah[15] correlation was one of the most widely used two-phase multiplier correlations for years, which include water, R-11, R-12, R-22, R-113, methanol, ethanol, benzene, toluene, and trichloroethylene condensing in horizontal, vertical, and inclined pipes of diameters ranging from 7 to 40mm.

$$Nu = 0.023\left(Gd_i/\mu_1\right)^{0.8} pr_1^{0.4}\left[1+\frac{3.8}{p_{reduce}^{0.38}}\left(\frac{x}{1-x}\right)^{0.76}\right] \tag{13}$$

(2) Tandon et al.[16] presented a correlation depending on the dominated force. For shear controlled flows with $Re_v > 30000$:

$$Nu = 0.084\mathrm{Re}_v^{0.67}\mathrm{Pr}_1^{1/3}\left(\frac{1}{Ja_1}\right)^{1/6} \tag{14}$$

For gravity controlled flows with $Re_v < 3000$:

$$Nu = 23.1\mathrm{Re}_v^{1/8}\mathrm{Pr}_1^{1/3}\left(\frac{1}{Ja_1}\right)^{1/6} \tag{15}$$

(3) Dobson and Chato's[17] annular flow correlation developed by using a two phase multiplier approach could be used with a zeotropic refrigerant, which was thought as independent of tube orientation[11].

$$Nu = 0.023\mathrm{Re}_l^{0.8}\mathrm{Pr}_l^{0.4}\left(1+\frac{2.22}{x_{tt}^{0.89}}\right) \tag{16}$$

where

$$\mathrm{Re}_1 = \frac{Gd_i\left(1-x\right)}{\mu_1} \tag{17}$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{18}$$

(4) Chen et al.[18] developed a correlation based on analytical and empirical results for annular film condensation inside tubes, which incorporated the effects of interfacial shear stress, interfacial waviness, and turbulent transport in the condensate film.

$$Nu = 0.018\left(\frac{\rho_l}{\rho_v}\right)^{0.39}\left(\frac{\mu_v}{\mu_l}\right)^{0.078} \mathrm{Re}_l^{0.2}(\mathrm{Re}_{to}-\mathrm{Re}_l)^{0.7}\mathrm{Pr}_l^{0.65} \tag{19}$$

Fig. 12 showed the comparison results of the experimental Nusselt number with the predicted Nusselt number of four correlations, and total 146 experimental data points from eight vertical tubes were used. Deviation ranges and mean deviation of the four correlations for the Nusselt number were presented in Table 2. The mean deviation of the four correlations was defined as:

$$\Delta\mathrm{dev} = \left(1/n\right)\sum\left(\left|Nu_{pre}-Nu_{\exp}\right|/\mathrm{Nu}_{\exp}\right) \tag{20}$$

Table 2 Deviation ranges and mean deviation of correlations

Correlation	Deviation range	Mean deviation
Shah [15]	+20% ~ -60%	32.37%
Dobson-Chato[17]	+100% ~ -20%	46.40%
Tandon [16]	+40% ~ -23%	22.46%
Chen [18]	+40% ~ -40%	22.34%

Correlation development

Finally, a new empirical correlation revised from Chen[18] correlation was developed based on the experimental data. It was used to predict condensation heat transfer of R410A in short vertical tube at different mass fluxes, as shown in Fig.13. According to the performance of condensation heat transfer in short vertical tube, the effect of entrance effect was considered in the new correlation:

$$Nu = 0.9\left(\frac{\rho_l}{\rho_v}\right)^{0.39}\left(\frac{\mu_v}{\mu_l}\right)^{0.078} \mathrm{Re}_l^{0.2}\left(\mathrm{Re}_{to}-\mathrm{Re}_l\right)^{0.7}\mathrm{Pr}_l^{0.65}\left(d/L\right) \tag{21}$$

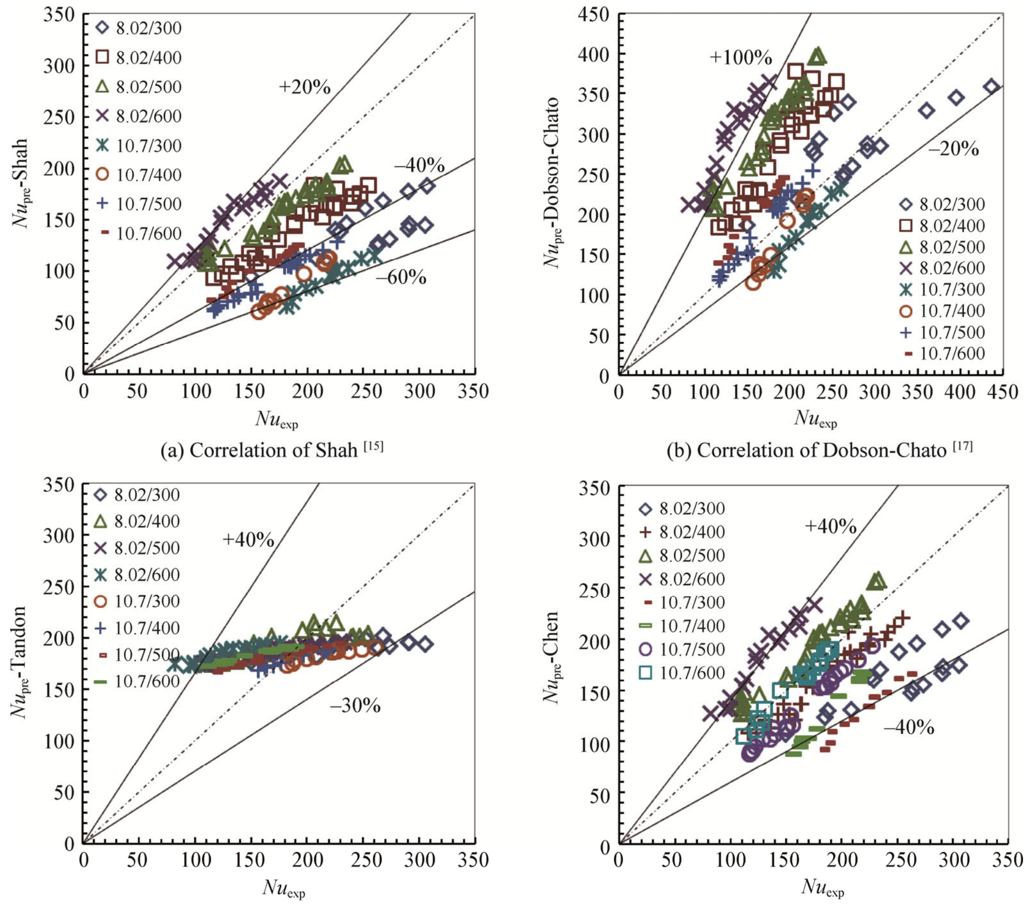
The new correlation had a deviation range of +15% to -20% and mean deviation 12.7%.

Conclusion

The condensation heat transfer of R410A in short vertical tube (8.02 and 10.7 ID, L=300, 400, 500, 600mm) at mass fluxes ranging from 58-246 kg $m^{-2}s^{-1}$ and saturation temperature of 38 °C was experimentally investigated.

Entrance effect plays an important role in condensation heat transfer in vertical tube. The influence of entrance effect on average condensation heat transfer coefficients will be weakened with the length of tube, especially when $L \geq 500$mm in present condition.

A modified physical model based on the assumptions of Dalkilic[7] was built. The model could be used to calculate the film thickness, condensate flow rate and condensation heat transfer coefficients in vertical tube. The changes of film thickness and condensation heat transfer coefficients along the tube were affected considerably by temperature distribution. The heat transfer coefficient reached the highest values at the entrance region mainly due to the entrance effect, and rapidly decreased at the bottom of tube.



(a) Correlation of Shah [15]

(b) Correlation of Dobson-Chato [17]

(c) Correlation of Tandon [16]

(d) Correlation of Chen [18]

Fig. 12 Comparison of the experimental Nusselt number (L=300, 400, 500, 600 mm and 8.02, 10.7 mm ID) with that of predicted by four correlations.

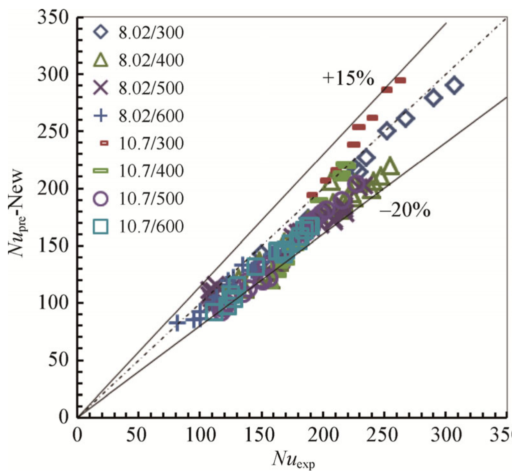


Fig. 13 Comparison of the experimental Nusselt number with that of predicted by new correlations by revision of Chen correlation

A modified correlation based on Chen was proposed for short vertical tube, and the effect of entrance effect was considered. It was with mean deviation of 12.7% in this study.

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References

[1] Akio Miyara, Kengo Nonaka, Mitsunori Taniguchi, Condensation heat transfer and flow pattern inside a herringbone-type micro-fin tube, International Journal of Refrigeration, vol.23, pp. 141–152, (2000).

[2] Yi-Yie Yan, Tsing-Fa Lin, Condensation heat transfer and pressure drop of refrigerant R134a in a small pipe, International Journal of Heat and Mass Transfer, vol. 42, pp. 697–708, (1999).

[3] S. Lips, J.P. Meyer, Experimental study of convective condensation in an inclined smooth tube, Part I: Inclination effect on flow pattern and heat transfer coefficient, International Journal of Heat and Mass Transfer, vol.55, pp.395–404, (2012).

[4] S.N. Sapali, Pradeep A.Patil, Heat transfer during condensation of HFC-134a and R404A inside of a horizontal

smooth and micro-fin tube, Experimental Thermal and Fluid Science, vol. 34, pp.1133–1141, (2010).

[5] A.S. Dalkilic, S. Wongwises, Intensive literature review of condensation inside smooth and enhanced tubes, International Journal of Heat and Mass Transfer, vol.52, pp. 3409–3426(2009).

[6] Seungmin Oh, Shripad T. Revankar, Analysis of the complete condensation in a vertical tube passive condenser, International Communications in Heat and Mass Transfer, vol.32, pp. 716–727, (2005).

[7] A.S. Dalkilic, S. Yildiz, S, Wongwises, Experimental investigation of convective heat transfer coefficient during downward laminar flow condensation of R134a in a vertical smooth tube, International Journal of Heat and Mass Transfer, vol.52, pp. 142–150(2009).

[8] A.S. Dalkilic, I. Teke, S. Wongwises, Experimental analysis for the determination of the convective heat transfer coefficient by measuring pressure drop directly during annular condensation flow of R134a in a vertical smooth tube, International Journal of Heat and Mass Transfer, vol.54, pp. 1008–1014, (2011).

[9] A.S. Dalkilic, O. Agra, I. Teke, S. Wongwises, Comparison of frictional pressure drop models during annular flow condensation of R600a in a horizontal tube at low mass flux and of R134a in a vertical tube at high mass flux, International Journal of Heat and Mass Transfer, vol.53, pp. 2052–2064, (2010).

[10] Jonathan A. Olivier, Leon Liebenberg, John R. Thome, Josua P. Meyer, Heat transfer, pressure drop, and flow pattern recognition during condensation inside smooth, helical micro-fin, and herringbone tubes, International Journal of Refrigeration, vol.30, pp. 609–623, (2007).

[11] A. Cavallini, D. Del Col, G.A. Longo, L. Rossetto, Experimental investigation on condensation heat transfer and pressure drop of new HFC refrigerants (R134a, R125, R32, R410A, R236ea) in a horizontal tube, International Journal of Refrigeration, vol.24, pp. 73–87, (2001).

[12] Hoo-Kyu Oh, Chang-Hyo Son, Condensation heat transfer characteristics of R-22, R-134a and R-410A in a single circular microtube, Experimental Thermal and Fluid Science, vol.35, pp. 706–716, (2011).

[13] O. GARCIA-VALLADARES, Review of In-Tube Condensation Heat Transfer Correlations for Smooth and Microfin Tubes, Heat Transfer Engineering, vol.24, pp. 6–24, (2003).

[14] A.S. Dalkilic, S. Laohalertdecha, S. Wongwises, Experimental investigation of heat transfer coefficient of R134a during condensation in vertical downward flow at high mass flux in a smooth tube, International Communications in Heat and Mass Transfer, vol.36, pp. 1036–1043, (2009) .

[15] M. M. Shah, A general correlation for heat transfer during film condensation inside pipes, International Journal of Heat and Mass Transfer, vol.22, pp. 547–556, (1979).

[16] Tandon, T. N., Varma, H. K., Gupta, C. P., Heat transfer during forced convection condensation inside horizontal tube, International Journal of Refrigeration, vol.18, pp. 210–214, (1995).

[17] Dobson M. K., Chato J. C., Condensation in smooth horizontal tubes, Journal of Heat Transfer, vol.120, pp. 193–213, (1998).

[18] Chen, S.L., Gerner, F.M., Tien, C.L., General film condensation correlations, Experimental Heat Transfer, vol.1, pp. 93–107, (1987).