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Experimental Investigation on Heat Transfer Coefficient during Upward Flow Condensation of R410A in Vertical Smooth Tubes

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This paper presents an experimental investigation on condensation of R410A upward flow in vertical tubes with the same inner diameter of 8.02mm and different lengths of 300 mm, 400 mm, 500 mm and 600mm. Condensation experiments were performed at mass fluxes of 103-490 kg $m^{-2}s^{-1}$. The saturation temperatures of experimental condition were 31℃, 38℃and 48℃, alternatively. The average vapor quality in the test section is between 0.91 and 0.98. The effects of tube length, mass flux and condensation temperature on condensation were discussed. Four correlations used for the upward flow condensation were compared with the experimental data obtained from various experimental conditions. A modified correlation was proposed within a ±15% deviation range.

Keywords: Condensation, R410A, Flow upward, Heat transfer coefficient, Refrigerant

Introduction

Convective condensation plays an important role in many industrial applications such as air-conditioning, power engineering and other thermal processing plants. Increasing the condenser efficiency in these applications can save material and reduce the running cost. Moreover, it can reduce the impact on environment.

Heat transfer characteristics of refrigerants condensation in horizontal smooth and enhanced tubes have been widely studied in the past decades [1-3]. Also, refrigerants condensation in vertical downward flow in smooth tubes and channels has been studied in recent years [3-7]. However, only a few studies deal with condensation in vertical upward flow, especially for refrigerants as working substances. Condensation in vertical upward flow usually occurs in reflux condensers and rectification system. In such conditions vapor flows counter currently to the condensation liquid. The shear stress at the interface of liquid and vapor obstructs the liquid flow downward, which is different from the co-current condensation conditions. The effect of the shear stress is different on the condensate film thickness, which decreases under co-current condensation, but increases under counter-current condensation conditions along the vapor flow direction. The situation is further complicated by the two-phase interfacial shear stress [8]. The flow patterns are also affected by the two-phase interfacial shear stress, so the heat transfer characteristics are much different from that in horizontal tube and in vertical tube with vapor flows downward.

Seban et al. [9] reviewed the Nusselt solution for laminar film condensation on a vertical plate with upward vapor flow and commented the possible condensate flow. He also presented the same laminar film condensation problem in the vertical tube with vapor flow upward.

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Nomenclature

$\dot{m}$	mass flow rate, kg/s	R	thermal resistance, K/W
G	mass flux, kg m^{-2} s^{-1}	***Greek symbols***	
x	vapor quality	ρ	Density, kg m^{-3}
C_p	specific heat capacity at constant pressure, J kg^{-1} K^{-1}	μ	Dynamic viscosity, kg m^{-1} s^{-1}
$\dot{q}$	heat transfer rate, W	ψ	h_{model}/h_l defined by shah's equations
h	heat transfer coefficient, W m^{-2} K^{-1}	***Subscripts***	
h_{fg}	latent heat of condensation, J kg^{-1}	pr	Pre-condenser
T	temperature, ℃	o	outlet
A	tube inner surface area, m^2	i	inlet
Fr_{so}	Soliman's modified Froude number	w	water
Fr_l	Froude number, $G^2 / \rho_l^2 gd$	r	Refrigerant/R410A
Re	Reynolds number	l	liquid
X	Lockhark Martinelli parameter	g	gas/vapor
Ga	Galileo number	ave	average
P	pressure, Mpa	wi	inner wall
Pr	Prandtl number	wo	outer wall
$P_{r\,educe}$	reduced pressure, P/P_{critical}	sat	saturation
Nu	Nusselt number	t	test section
k	thermal conductivity, W m^{-1} K^{-1}	sf	superficial
Co	parameter in Shah correlation	exp	experimental
L	Length of the test tube, m		

Y. Liao et al. [8] developed a mechanistic model using the heat and mass transfer analogy approach for local heat transfer in reflux condensation of flowing vapor and non-condensable gases counter-current to laminar liquid film. The model could satisfactorily predict the magnitude and trend of local heat transfer coefficients along the tube under a wide variety of reflux condensation conditions, with the root mean square error of about 30% comparing to a large number of experimental data points. Also S. Fiedler et al. [10] presented a physical model for laminar reflux condensation of pure saturated vapor flow upward in an inclined small diameter tube. The calculated values for the film thickness and the mean heat transfer coefficient were in good agreement with the experimental data. The deviation was less than 15%.

Thumm et al. [11] investigated film condensation of water in a vertical tube with counter-current vapor flow. The effects of film Reynolds number, Prandtl number and interfacial shear stress on heat transfer were studied independently. The results showed that a reduction in heat transfer coefficients was found for Re_l<10 and an increase for 20<Re_l<670 with the increasing values of interfacial shear stress. However, the influence of the shear stress disappeared for turbulent film. He also presented a correlation as a function of shear stress and film Reynolds number.

Gross et al. [12] investigated the effect of conjugated shear stress and Prandtl number on reflux condensation heat transfer inside a vertical tube. In his study, the liquid film (0.68<Re_l<200) and the vapor (1000<Re_g<16500) flowed downward and upward respectively. The results showed that the shear stress impeded the heat transfer only when the film was very thin and in the smooth laminar range. But in the laminar-wavy range and developing turbulence the heat transfer coefficients were found to increase with the shear stress. The heat transfer was enhanced with the increase of liquid Prandtl number.

Gross et al. [13] found that the decrease and subsequent re-increase of heat transfer coefficients was obtained with increasing shear stress for counter-current vapor-flow.

Lips and Meyer [14] studied experimentally the effect of the inclination angle of an 8.38mm inner diameter smooth tube for condensing flow of R134a which included the condition of vapor flow upward. They found that the heat transfer coefficient decreased for upward flow. For high mass fluxes and high vapor qualities, the shear forces were predominant; the flow remained annular and the condensation heat transfer coefficient was constant, whatever the angle of inclination.

Akhavan-Behabadi et al. [15] investigated the effect of the inclination angle of an 8.92mm inner micro-fin tube for condensing flow of R134a. The results showed that the heat transfer coefficients were higher for downward flows than for upward flows.

As a conclusion, experimental studies on condensation heat transfer coefficients of refrigerant in short tubes with vapor flow upward were strongly limited, especially on the condition that condensation liquid was separated at the bottom of the tube which was usually seen as an efficient method to increase heat transfer. So the aim of this work is to achieve a better understanding of this kind of flow in different length of tubes and develop a more accurate correlation to predict the heat transfer coefficients in this condition.

Experimental set-up

A schematic diagram of the test apparatus is shown in Fig.1. The set-up consisted of a test line and a by pass line. The refrigerant was circulated by a compressor. The test line consisted of a pre-condenser, test section and a post-condenser. The by-pass line consisted of a bypass condenser used to control the mass flow through the test section. The lines were combined at the liquid-storage tank. The liquid of the refrigerant flowed through the expansion valve and entered the evaporator. Eventually, the refrigerant returned to the compressor to complete the cycle.

Under the test section, there was a gas-liquid separator used to gather the condensation liquid of refrigerant condensed in the test section and the pre-condenser and avoid the gas of refrigerant getting through this line.

The test section was a vertical tube-in-tube heat exchanger with refrigerant flowing upward in the inner tube and cooling water flowing at the same direction in the annulus. The inner tube was made from smooth copper with inner diameters of 8.02 mm. The outer tubes were made from plexiglass with inner diameter of 26mm. The detailed information on the heat exchanger and the location of the thermocouples are showed in Fig.2.

In all sections of the test apparatus, T-type thermocouples were used to measure temperature. Ten thermocouples were located on the outer surface of the copper tube at five points along the test tube. The uncertainty of the temperature measurements is ±0.1℃. The pressures in different parts of the test apparatus were measured by pressure transducer. The mass flux was measured by Coriolis mass flow meter in the range of 10-100kg/h. The uncertainty of the mass flow meter is ±0.2kg/h.

Data reduction

The mass flow rate

Part of the refrigerant is condensed in the pre-condenser and the condensation liquid will get into the gas-liquid separator, so the real mass flow rate of the refrigerant flowing through the test section is as follows:

$$\dot{m}_r = x_{pr.o}\dot{m} \quad (1)$$

$\dot{m}$ is the mass flow rate measured by the Coriolis mass flow meter. $x_{pr.o}$ is the outlet vapor quality of the pre-condenser.

The vapor quality of the pre-condenser outlet

The refrigerant of the inlet of the pre-condenser is superheated, so the total heat transfer rate from cooling water to vapor of R410A in the pre-condenser is the sum of the latent and sensible heat transfer rates:

$$\dot{q}_{w.pr} = \dot{q}_{sensible} + \dot{q}_{latent} = \dot{m}_{w.pr}C_{P.w.pr}(T_{w.o} - T_{w.i})_{pr} \quad (2)$$

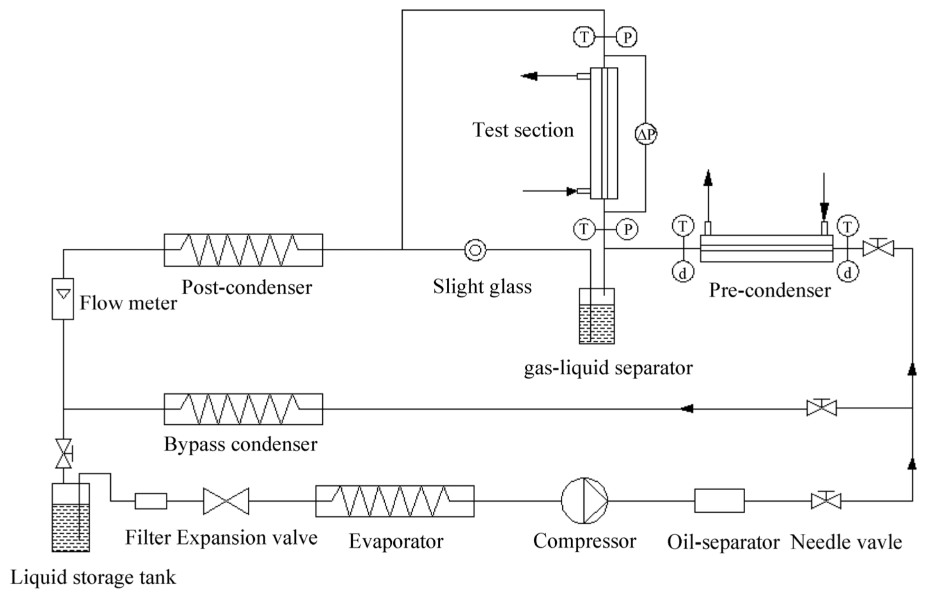


Fig. 1 Schematic diagram of experimental set-up

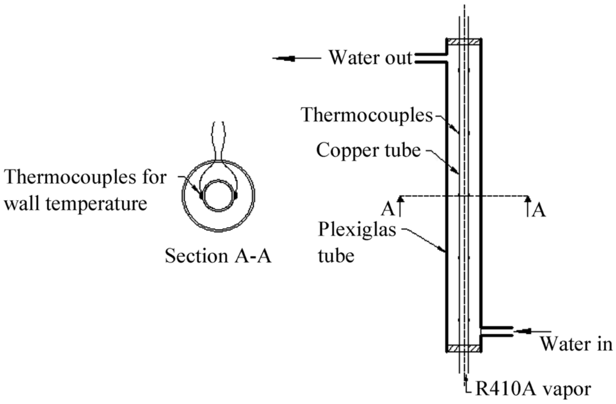


Fig. 2 Schematic diagram of test section

$$\dot{q}_{sensible} = \dot{m}C_{P.r.pr}(T_{r.i} - T_{r.sat})_{pr} \tag{3}$$

$$\dot{q}_{latent} = \dot{m}(1 - x_{pr.o})h_{fg} \tag{4}$$

$\dot{q}_{w.pr}$ is the total heat transfer rate in the pre-condenser. $\dot{q}_{sensible}$ and $\dot{q}_{latent}$ are the sensible and latent heat transfer rate respectively. $(T_{w.o}-T_{w.i})_{pr}$ is the temperature difference of the water between outlet and inlet position of the pre-condenser. $(T_{r.i}-T_{r.sat})_{pr}$ is the temperature difference of the refrigerant between inlet position of pre-condenser and the saturation temperature in the pre-condenser. $(1-x_{pr.o})$ is the liquid quality of outlet position of the pre-condenser. $\dot{m}_{w.pr}$ is the mass flow rate of the water entering the pre-condenser. h_{fg} is the latent heat of condensation.

According to equations (2), (3) and (4), the outlet vapor quality of R410A from the pre-condenser is calculated as:

$$x_{pr.o} = \frac{1}{h_{fg}}\left[\frac{\dot{q}_{w.pr}}{\dot{m}} - C_{P.r.pr}\left(T_{r.i} - T_{r.sat}\right)pr\right] \tag{5}$$

The vapor quality of the test section

The liquid from the pre-condenser goes into the gas-liquid separator, hence the vapor quality of the test section inlet is regarded below as:

$$x_{t.i} = 1 \tag{6}$$

The vapor quality alternation in the test section is

$$\Delta x = \frac{\dot{q}_t}{\dot{m}_r h_{fg}} \tag{7}$$

$\dot{q}_t$ is the heat transfer rate n the test section.

The average vapor quality in the test section is

$$x_{ave.t} = \frac{2 - \Delta x}{2} \tag{8}$$

The average condensation heat transfer coefficient

$$h_{exp} = \frac{\dot{q}_t}{A(T_{r.sat} - T_{wi})} \tag{9}$$

$T_{r.sat}$ is the saturation temperature of the refrigerant in the test section and obtained by averaging the saturation temperature measured at the inlet and outlet of the test section. T_{wi} is the inter-wall temperature and obtained by the outer-wall temperature of the tube, T_{wo}, the thermal resistance of the wall of the copper tube, R_w:

$$T_{wi} = T_{wo} + \dot{q}_t R_w \tag{10}$$

Where R_w is as follows:

$$R_w = \ln(\frac{d_o}{d_i})/2\pi k_{cu}L \tag{11}$$

T_{wo} was calculated by averaging the outer-wall temperature measured at the five points along the test tube. The equation is given by:

$$T_{wo} = \frac{1}{n}\sum_{1}^{n} T_{wo.j} \tag{12}$$

A is the inner surface of the test section:

$$A = \pi d_i L \tag{13}$$

Where d_i is the inner diameter of the test tube, and L is the length of the test tube.

Result and discussion

In the present work，flow condensation of R410A with vapor flow upward inside vertical smooth tubes with the inner diameter 8.02mm and 300mm, 400mm, 500mm and 600mm lengths was measured at the condensation temperatures of 31℃, 38℃ and 48℃. Experiments were carried out for mass flux range of 103-490kg m^{-2} s^{-1}. The modified Froude number proposed by Soliman [16] was used to predict the flow patterns. The Froude number represents a balance between inertial and gravitational forces on the liquid film. Soliman [16] argued that the appropriate velocity and length scale were the actual liquid velocity and the film thickness. However, these parameters were not known based solely on the mass flux, the vapor quality and the fluid properties. He obtained the equations based on the relations for two-phase pressure drop in annular flow. The equation of Soliman's modified Froude number is given as follows:

$$Fr_{so} = c_1 Re_l^{c_2}(\frac{1+1.09X^{0.039}}{X})^{1.5}\frac{1}{Ga^{0.5}} \tag{14}$$

where

$Re_l \leq 1250$ $c_1=0.025$ $c_2=1.59$

$Re_l > 1250$ $c_1=1.26$ $c_2=1.04$

Where the Galileo number is defined as follows:

$$Ga = \frac{\rho_l(\rho_l - \rho_g)gd^3}{\mu_l^2} \tag{15}$$

Soliman [16] proposed that wavy flow was observed for $Fr_{so}<7$, and annular flow was observed for $Fr_{so}>7$ by compare with data in tubes of 4.8mm to 25 mm inner diameter, and with water, refrigerants and acetone. However, Dobson [17] and Dobson et al.[18] have not observed the annular flow until around Fr_{so} =18. In the present work, the Froude number is between 18.25 and 98.86, so the data points can be regarded as in the annular flow regime in this experiment.

The effects of mass flux and saturation temperature

Fig.3 shows the average heat transfer coefficients of R410A at saturation temperature of 31℃, 38℃ and 48℃ with various mass fluxes in different length of tubes. The condensation heat transfer coefficients increase with mass flux and decrease with saturation temperature. The results reveal that mass transfer resistance is a strong function of mass flux. With the increase of mass flux, the average condensation heat transfer coefficients increase due to the enhancement of momentum exchange which can lead to a reduction of thermal conduction resistance in the liquid layer. Traviss et al.[19] argued that the shear stress at the liquid–vapor interface increases with the vapor velocity resulting in an increase of momentum, thus leads to the enhancement of the heat transfer. Although the liquid–vapor interface shear stress obstructs the flow of liquid film which results in thickening of the liquid layer, this force can also enhance the turbulence in the tube which leads to an enhancement of heat transfer.

The condensation heat transfer coefficients decrease with saturation temperature. This is due to the different thermodynamic properties of the refrigerants at three saturation temperatures. On the one hand, at the higher saturation temperature, the corresponding saturation pressure and vapor density will be higher, leading to higher ratio of the vapor density to the liquid density. Thus, at given mass flux, the vapor velocity will decreases with the increase of saturation temperature. On the other hand, the liquid film thermal conductivity decreases with saturation temperature. The liquid film is the main thermal resistance in the tube. The main thermophysics properties of R410A at different saturation temperatures are listed in Table 1.

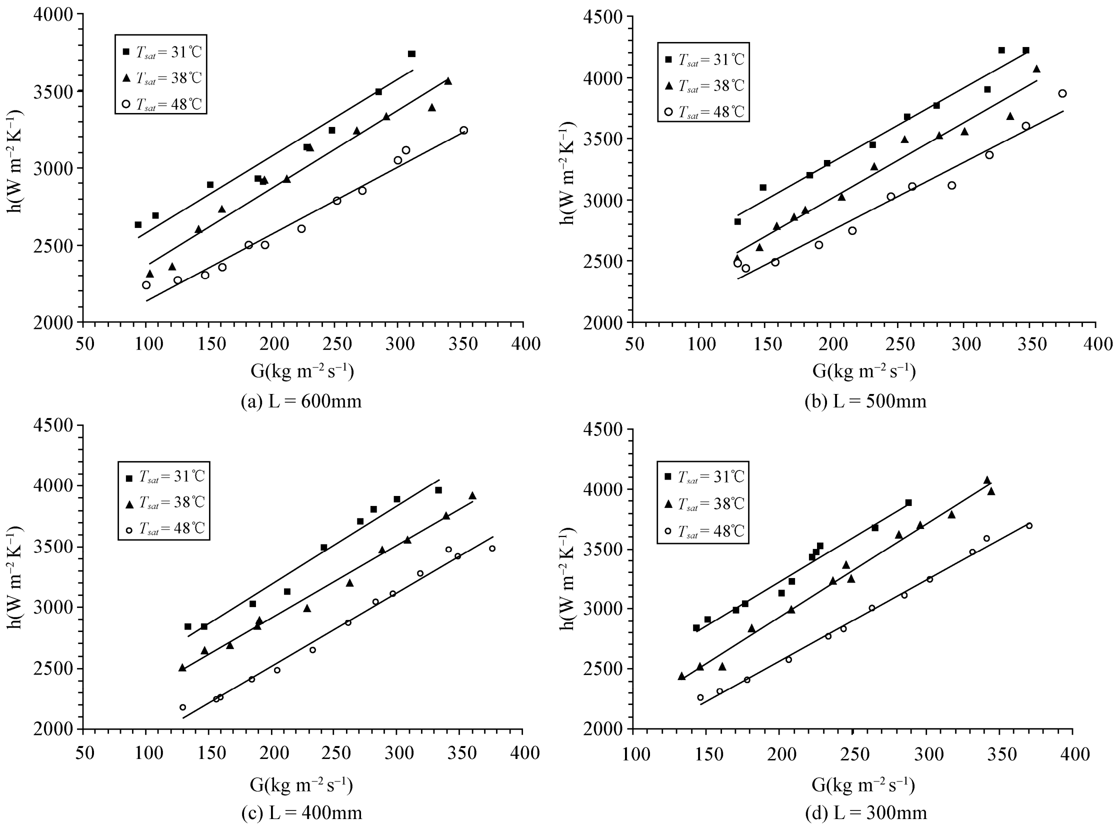


Fig. 3 Effects of mass flux and saturation temperature on condensation heat transfer coefficient

Table 1 Properties of R410A at saturation temperature 31℃, 38℃ and 48℃

T_{sat} (℃)	P_{sat} (MPa)	ρ_l (kg/m^3)	ρ_g (kg/m^3)	μ_l (uPa·s)	μ_g (uPa·s)	Pr_l	k_l (mW/m·K)
31	1.9385	1027.3	77.593	108.89	14.101	2.2634	85.743
38	2.3102	987.52	95.722	98.708	14.692	2.2848	81.978
48	2.9321	921.75	130.47	84.609	15.784	2.3947	76.710

The effect of tube length

The comparison of the average heat transfer coefficient for tube length of 300mm, 400mm, 500mm and 600mm at saturation temperatures 31℃, 38℃ and 48℃ was given in Fig.4. The results reveal that the effect of tube length is not obvious. The gravity plays a strong effect on liquid film in vertical tube comparing with that in horizontal tube. On the condition that condensation in vertical tube with vapor flows upward, the liquid film flows downward. So at the entrance of the tube, the thickness of film is the thickest. As a result, the traditional entrance effect has no effect on that flow condition. In addition the change of the vapor quality is less than 0.15 in all mass fluxes range. With the effect of interfacial shear stress caused by counter-current vapor flow, the average film thickness may not be the most important factor on condensation heat transfer. As a result, the short-tube effect is not obvious in this flow condition.

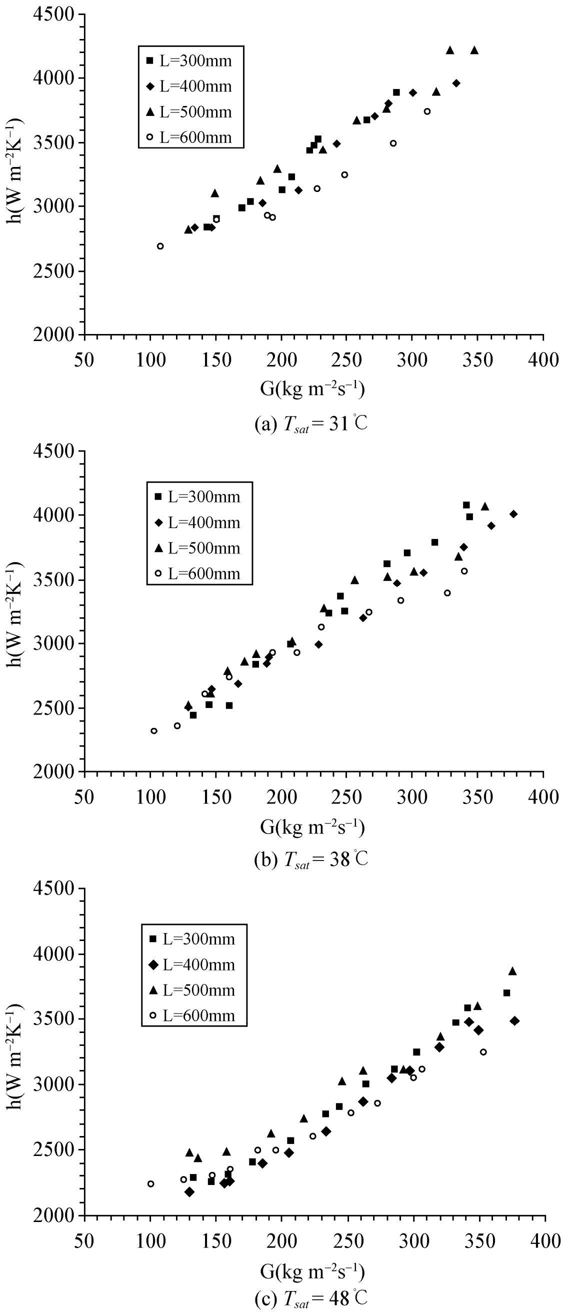


Fig. 4 Comparison of average heat transfer coefficient vs. various tubes length and T_{sat}=31℃ (*a*), T_{sat}=38℃ (b) and T_{sat}=48℃ (*c*)

Comparison with existing correlations

Condensation heat transfer of refrigerants in vertical tubes with vapor flow downward has been studied for many years. But study on condensation in vertical tubes with vapor flow upward was very limited. This experiment condition was not continuous and stable in the tubes. In the present work, a gas-liquid separator was set at the entrance of the tube so that the experiment could be conducted continually and stably. Among all the correlations, the correlation proposed by Shah [20] and the correlation suggested by Dobson and Chato [21] were developed for annular flow regimes and were independent of tube orientations. Bivens and Yokozeki [22] modified Shah [20] correlation for various flow patterns of R22, R502, R32/R134a, R32/R123/R134a. Tang et al. [23] modified Shah [17] equation for annular flow pattern of R410A, R134a and R22 with the Fr_{so}>7 as the indicated flow patterns. The detailed recommendation of these correlations are showed in Table 2.

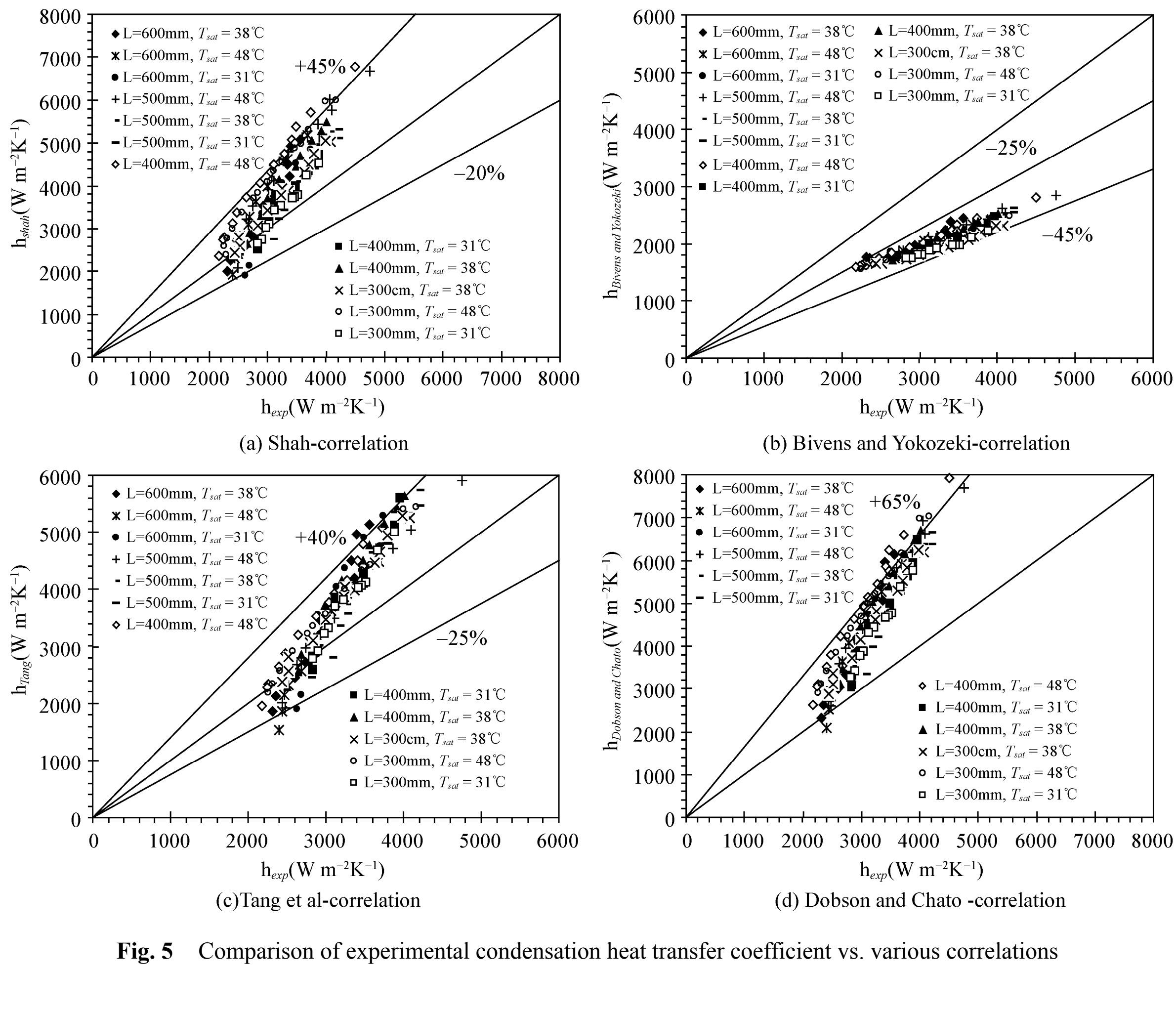
The comparison of the experimental condensation heat transfer coefficients with the four correlations were given in Fig.5. It shows that correlations proposed by Shah [20], Dobson and Chato [21] and Tang et al. [23] over-predicted almost all the experimental data. The Bivens and Yokozeki [22] correlation lower predicted all the experimental data. The Shah [20] and Tang et al. [23] correlations had the best predictability of experiment with the deviation from -20% to 45% and -25% to 40%, which was still very large for use in practice. The statistical analysis results were listed in Table 3. Correlations by Shah [20], Bivens and Yokozeki [22], Tang et al. [23] and Dobson and Chato [21] showed average deviations of 20.2, -35.74, 33.82 and 43.17 and mean deviations of 22.7, 35.74, 19.78 and 43.56 respectively. Since the Shah [20] correlation gave the best performance, the following work would modify the Shah correlation.

Correlation development concerning heat transfer coefficient

Shah [20] correlation developed for saturated boiling heat transfer was modified to improve the predictability with the 144 data points of smooth tube under the present experimental conditions. Shah [20] had `found that in the

Table 2 Models for Flow Condensation of Pure Refrigerants inside Tubes

Authors	Model/correlation	Application conditions
Shah[20]	$Nu = 0.023\left(Gd/\mu_l\right)^{0.8} Pr_l^{0.4}\left[1+\frac{3.8}{P_{\mathrm{reduce}}^{0.38}}\left(\frac{x}{1-x}\right)^{0.76}\right]$ $P_{reduce} = P_{sat}/P_{critical}$	Annular flow
Bivens and Yokozeki [22]	$h_{Bivens} = h_{shah}F \qquad F = 0.78738 + 6187.89/G^2$	Various flow patterns
Tang et al. [23]	$Nu = 0.023Re_l^{0.8}Pr_l^{0.4}[1+4.863((-\ln(P_{reduce})\frac{x}{1-x})^{0.836}]$	Annular flow $Fr_{so} > 7$
Dobson and Chato [21]	$Nu = 0.023Re_l^{0.8}Pr_l^{0.4}[1+\frac{2.22}{X_{tt}^{2.22}}]\cdots\cdots\cdots(a)$ $Nu = \frac{0.23Re_{vo}^{0.12}}{1+1.11X_{tt}^{0.58}}[\frac{GaPr_l}{Ja_l}]^{0.25} + (1-\frac{\theta_l}{\pi})Nu_{forced}\cdots(b)$ 式中：$Nu_{forced} = 0.0195\mathrm{Re}_l^{0.8}\mathrm{Pr}_l^{0.4}\phi_l(X_{tt})$ $\phi_l(X_{tt}) = \sqrt{1.376+\frac{c_1}{X_{tt}^{c2}}}$ $X_{tt} = (\frac{1-x}{x})^{0.9}(\frac{\rho_g}{\rho_l})^{0.5}(\frac{\mu_l}{\mu_g})^{0.1}$ *For* $0<Fr_l \le 0.7$, $c_1=4.172+5.48Fr_l-1.564Fr_l^2 \qquad c_2 = 1.773-0.169Fr_l$ *For* $Fr_l > 0.7$ $c_1=7.24 \qquad c_2 = 1.655$	(a) Annular flow $G\ge500$kg/m^{2}s $0<x<1$ $G<500$kg/m^{2}s $Fr_{so}>20$ (b) wave-annular flow $G<500$kg/m^{2}s $Fr_{so}<20$



(a) Shah-correlation (b) Bivens and Yokozeki-correlation

(c)Tang et al-correlation (d) Dobson and Chato -correlation

Fig. 5 Comparison of experimental condensation heat transfer coefficient vs. various correlations

Table 3 Deviations of various correlations against the experiment data

Correlations	Average deviation	Mean deviation
Shah	20.2	22.7
Bivens and Yokozeki	-35.74	35.74
Tang et al.	33.82	19.78
Dobson and Chato	43.14	43.56

Average deviation $= \frac{1}{n}\sum_1^n [\frac{(h_{cal}-h_{exp})\times 100}{h_{exp}}]$;

Mean deviation $= \frac{1}{n}\sum_1^n ABS[\frac{(h_{cal}-h_{exp})\times 100}{h_{exp}}]$

absence of bubble nucleation and as long as the entire pipe surface kept wetting by the liquid, the following equation may apply to all flow orientations.

$$\psi = 1.8 / Co^{0.8} \tag{16}$$

Where

$$\psi = h_{model} / h_{sf} \tag{17}$$

$$Co = (\frac{1}{x}-1)^{0.8}(\frac{\rho_g}{\rho_l})^{0.5} \tag{18}$$

Where the superficial heat coefficient of the liquid phase was calculated as:

$$h_{sf} = h_l(1-x)^{0.8} \tag{19}$$

Where h_l is the heat transfer coefficient assuming all the mass flowing as liquid and was given by the Dittus-Boelter single-phase forced convection correlation as:

$$h_l = 0.023 Re_l^{0.8} Pr_l^{0.4} k_l / d \tag{20}$$

Shah [20] argued that equation (20) could apply to condensation in all flow orientations. The liquid film would be formed in the process of condensation whenever vapor contacts the tube surface and thus the tube circumference would always be wetted at all flow rates and in all flow orientations.

By analyzing the experiment data points in term of Co and ψ, the equation (16) was not fit in our work. So the following equation was developed in term of Co and ψ, which was found to give the best fit:

$$\psi = 4.32 / Co^{0.67} \tag{21}$$

This new correlation was developed using equations (17), (18), (19), (20) and (21).

Fig.6 shows the development of the equation (21) using the linear fitting.

The comparison of the proposed condensation heat transfer coefficient correlation with the experimental heat transfer coefficient was given in Fig.7. Almost all the measured data fall within ±15% by means of proposed correlation for all mass fluxes (103-490kg m^{-2} s^{-1}) and saturation temperatures 31℃, 38℃ and 48℃ of R410A.

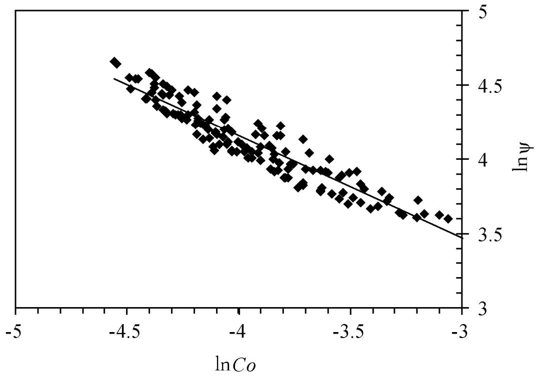


Fig. 6 Development of the equation (21) in term of Co and ψ

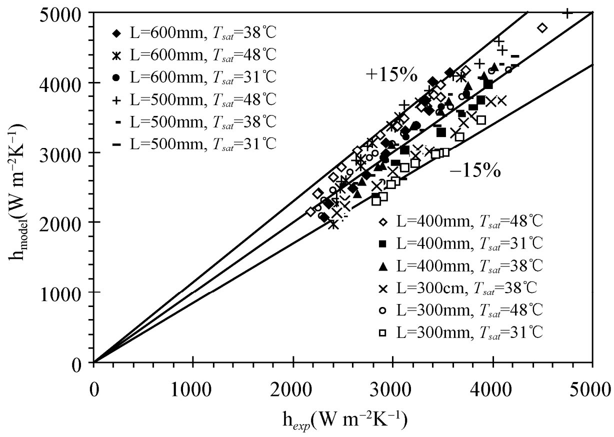


Fig. 7 Comparison of experimental condensation heat transfer coefficient vs. the correlation

Conclusion

Average heat transfer coefficient of R410A was investigated during condensation in vertical tubes with vapor flow upward in different length of smooth tubes. The study was conducted at condensation temperatures 31℃, 38℃ and 48℃ with mass fluxes between 103-490kg m^{-2}s^{-1}. The accurate and repeatable condensation heat transfer coefficients data during annular flow were obtained. The Fr_{so}=18 was served as an indicator of the transition from wavy-annular flow to annular flow. The data in the present work were collected in annular flow regime. The effects of mass flux, condensation saturation temperature and tube length on heat transfer were discussed and investigated in detail. The following results were obtained.

The condensation heat transfer coefficient increases with the mass flux and decreases with the increase of condensation saturation temperature. Short-tube effect is not obvious in such flow condition since entrance effect is destroyed by the liquid film flow downward by the gravity force.

Four annular flow heat transfer correlations which were developed independent of the tube orientation were chosen to compare with experimental heat transfer coefficients. The Shah [20] correlation had the best predictability of experimental with the deviation from -20% to 45%, average deviation of 20.2 and mean deviation of 22.7.

A modified correlation for condensation heat transfer coefficients was proposed. Almost all the measured data fall within ±15% by means of proposed correlation for all mass fluxes (103-490kg m^{-2} s^{-1}) and saturation temperatures 31℃, 38℃ and 48℃ of R410A.

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References

[1] Colombo L P M, Lucchini A, Muzzio A: Flow patterns, heat transfer and pressure drop for evaporation and condensation of R134A in microfin tubes, International Journal of Refrigeration, vol.35, pp.2150–2165, (2012).

[2] Jung D, Cho Y, Park K: Flow condensation heat transfer coefficients of R22, R134a, R407C, and R410A inside plain and microfin tubes, International Journal of Refrigeration, vol.27, pp.25–32, (2004).

[3] Liebenberg L, Meyer J P：A review of flow pattern-based predictive correlations during refrigerant condensation in horizontally smooth and enhanced tubes, Heat Transfer Engineering, vol.29, pp. 3–19, (2008).

[4] Dalkilic A S, Laohalertdecha S, Wongwises S: 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).

[5] Dalkilic A S, Teke I, Wongwises S: 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).

[6] Dalkilic A S, Laohalertdecha S, Wongwises S: Effect of void fraction models on the film thickness of R134a during downward condensation in a vertical smooth tube, International Communications in Heat and Mass Transfer, vol.36, pp.172–179, (2009).

[7] Xu W, Jia L, Visualization on flow patterns during condensation of R410A in a vertical rectangular channel, Journal of Thermal Science, vol.23, pp.269–274, (2014).

[8] Liao Y, Guentay S, Suckow D, et al: Reflux condensation of flowing vapor and non-condensable gases countercurrent to laminar liquid film in a vertical tube, Nuclear Engineering and Design, vol.239, pp.2409–2416, (2009).

[9] Seban R A, Hodgson J A: Laminar film condensation in a tube with upward vapor flow, International Journal of Heat and Mass Transfer, vol.25, pp.1291–1300, (1982).

[10] Fiedler S, Auracher H: Experimental and theoretical investigation of reflux condensation in an inclined small diameter tube, International Journal of Heat and Mass Transfer, vol.47, pp.4031–4043, (2004).

[11] Thumm S, Philipp C, Gross U: Film condensation of water in a vertical tube with countercurrent vapor flow, International Journal of Heat and Mass Transfer, vol.44, pp.4245–4256, (2001).

[12] Gross U, Philipp C: Conjugated shear stress and Prandtl number effects on reflux condensation heat transfer inside a vertical tube, International Journal of Heat and Mass Transfer, vol.49, pp.144–153, (2006).

[13] Gross U, Storch T, Philipp C, et al: Wave frequency of falling liquid films and the effect on reflux condensation in vertical tubes, International Journal of Multiphase Flow, vol.35, pp.398–409, (2009).

[14] Lips S, Meyer J P. 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).

[15] Akhavan-Behabadi M A, Kumar R, Mohseni S G: Condensation heat transfer of R-134a inside a microfin tube with different tube inclinations, International Journal of Heat and Mass Transfer, vol.50, pp.4864–4871, (2007).

[16] Soliman H M: On the annular-to-wavy flow pattern transition during condensation inside horizontal tubes, The Canadian Journal of Chemical Engineering, vol.60, pp. 475–481, (1982).

[17] Dobson M K: Heat transfer and flow regimes during condensation in horizontal tubes, Air Conditioning and Refrigeration Center. College of Engineering, University of Illinois at Urbana-Champaign, USA (1994).

[18] Dobson M K, Chato J C, Hinde D K, et al: Experimental evaluation of internal condensation of refrigerants R-134a and R-12, Air Conditioning and Refrigeration Center. College of Engineering, University of Illinois at Urbana-Champaign., USA (1993).

[19] Traviss DP, Rohsenow WM, Baron AB: Forced convection condensation in tubes: a heat transfer equation for condenser design, ASHRAE Trans, vol.79, pp.157–165, (1973).

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

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

[22] Bivens D B, Yokozeki A: Heat transfer coefficients and transport properties for alternative refrigerants, (1994).

[23] Tang L, Ohadi M, Johnson A T: Flow Condensation in Smooth and Micro-fin Tubes with HCFC-22, HFC-134a and HFC-410 Refrigerants. Part II: Design Equations, Journal of Enhanced Heat Transfer, vol.7, (2000).