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Two-Phase Heat Transfer Coefficients of R134a Condensation in Vertical Downward Flow at High Mass Flux

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1. Introduction

The transfer process of heat between two or more fluids of different temperatures in a wide variety of applications is usually performed by means of heat exchangers e.g. refrigeration and air-conditioning systems, power engineering and other thermal processing plants. In refrigeration equipment, condensers' duty is to cool and condense the refrigerant vapor discharged from a compressor by means of a secondary heat transfer fluid such as air and fluid. Examinations and improvements on the effectiveness of condenser have importance in case there is a design error which can cause the heat transfer failure occurrence in the condenser.

Industries have been trying to improve the chemical features of alternative refrigerants of CFCs due to the depletion of the earth's ozone layer. Because of the similar thermo-physical properties of HFC-134a to those of CFC-12, an intensive support from the refrigerant and air-conditioning industry has given to the refrigerant HFC-134a as a potential replacement for CFC-12. On the other hand, even though there isn't much difference in properties between these two refrigerants, the overall system performance may be affected significantly by these differences. For that reason, the detailed investigation of the properties of HFC-134a should be performed before it is applied.

A large number of researchers focused on the heat transfer and pressure drop characteristics of refrigerants over the years, both experimentally and analytically, mostly in a horizontal straight tube. The CFCs' heat transfer and pressure drop studies inside small diameter vertical either smooth or micro-fin tubes for downward condensation has been concerned comparatively little in the literature. Briggs et al. (1998) have studied on in-tube condensation using large diameter tubes of approximately 20.8 mm with CFC113. Shah (1979) used some smooth tubes up to 40 mm i.d. at horizontal, inclined and vertical positions to have a well-known wide-range applicable correlation. Finally, it has been compared by researchers

commonly for turbulent condensation conditions and it is considered to be the most predictive condensation model for the annular flow regime in a tube. Oh and Revankar (2005) developed the Nusselt theory (1916) to investigate the PCCS condenser using 47.5 mm i.d. and 1.8 m long vertical tube in their experimental study for the validation of their theoretical calculations. Maheshwari et al. (2004) simulated PCCS condensers used in water-cooled reactors using a 42.77 mm i.d. vertical tube during downward condensation presence of non-condensable gas. The prediction of heat transfer considering mass transfer along the tube length was performed by means of a computer code in their study.

Convective condensation in annular flow occurs for many applications inside tubes such as film heating and cooling processes, particularly in power generation and especially in nuclear reactors. The one of the most important flow regimes is annular flow which is characterized by a phase interface separating a thin liquid film from the gas flow in the core region. This flow regime is the most investigated one either analytically or experimentally, because of its practical significance and common usage.

Akers et al. (1959) focused on the similarity between two phase and single phase flows and developed a correlation using two-phase multiplier to predict frictional two phase pressure drop which is same rationale as the Lockhart-Martinelli (1949) two-phase multiplier. His model is known as "equivalent Reynolds number model" in the literature. According to this model, an equivalent all liquid flow, which produces the same wall shear stress as that of the two phase flow, is replaced instead of annular flow inside a tube. Many researchers benefitted from his model such as Moser et al. (1998) and Ma and Rose (2004). Moser et al. (1998) predicted heat transfer coefficient in horizontal conventional tubes developing a model. Ma and Rose (2004) studied heat transfer and pressure drop characteristics of R113 in a 20.8 i.d. vertical smooth and enhanced tubes.

Dobson et al. (1998) used zeotropic refrigerants in their condensation tests including the wide range of mass flux in horizontal tubes and benefitted from the two-phase multiplier approach for annular flow. In their study, heat transfer coefficient increased with increasing mass flux and vapor quality in annular flow due to increased shear and thinner liquid film than other flow regimes. Sweeney (1996) modified their model their model for R407C considering the effect of mass flux.

Cavallini et al. (1974) have some significant theoretical analysis on the in-tube condensation process regarding the investigation of heat transfer and pressure drop characteristics of refrigerants condensing inside various commercially manufactured tubes with enhanced surfaces using a number of correlations in the literature. Lately, Cavallini et al. (2003) prepared a review paper on the most recent condensation works in open literature including the condensation inside and outside smooth and enhanced tubes.

Valladares (2003) presented a review paper on in-tube condensation heat transfer correlations for smooth and micro-fin tubes including the comparison of experimental data belong to various experimental conditions from different researchers. Wang and Honda (2003) made a comparison of well-known heat transfer models with experimental data belong to various refrigerants from literature and proposed some models, valid for the modified annular and stratified flow conditions, for micro-fin tubes. Bassi and Bansal (2003) compared various empirical correlations and proposed two new empirical models for the determination of condensation heat transfer coefficients in a smooth tube using R134a with lubricant oil. Jung et al. (2003, 2004) did condensation tests of many refrigerants such as R12, R22, R32, R123, R125, R134a, and R142b inside a smooth tube. They paid attention not only comparison of their experimental data with various well-known correlations but also proposition on a new correlation to predict condensation heat transfer coefficients.

Generally, the condensation heat transfer coefficients and pressure drops in tubes have been computed by empirical methods. The modifications of the Dittus-Boelter single-phase forced convection correlation (1930) are used in the literature, as in Akers et al. (1959), Cavallini and Zecchin (1974), and Shah (1979). Dalkilic et al. (2010a) made a comparison of thirteen well-known two-phase pressure drop models with the experimental results of a condensation pressure drop of R600a and R134a in horizontal and vertical smooth copper tubes respectively and revealed the main parameters of related models and correlations. Dalkilic et al. (2008a) used the equivalent Reynolds number model (1998) to propose a new correlation for the two-phase friction factor of R134a and also discussed the effect of main parameters such as heat flux, mass flux and condensation temperature on the pressure drop. Dalkilic et al. (2009b) made a comparison of eleven well-known correlations for annular flow using a large amount of data obtained under various experimental conditions to have a new correlation based on Bellinghausen and Renz's method (1992) for the condensation heat transfer coefficient of high mass flux flow of R134a. The effects of heat flux, mass flux and condensation temperature on the heat transfer coefficients also exist in their paper. Dalkilic et al. (2009c) showed the significance of the interfacial shear effect for the laminar condensation heat transfer of R134a using Carey's analysis (1992), which is the improved version of Nusselt's theory (1916), and proposed a new correlation based on Bellinghausen and Renz's method (1992) for the condensation heat transfer coefficient during annular flow of R134a at low mass flux in a vertical tube. Dalkilic et al. (2008b) investigated thirty-three void fraction models and correlations from the available literature and compared them each other using relevant data. The friction factors, based on the analysis of Ma et al. (2004), are obtained from various void fraction models and correlations and a comparison was made with each other and also with those determined from graphical information provided by Bergelin et al. (1946). The presentation for the effect of void fraction alteration on the momentum pressure drop was also shown in their paper. Dalkilic et al. (2009a) compared some simple void fraction models of the annular flow pattern for the forced convection condensation of pure R134a taking into account the effect of the different saturation temperatures in high mass flux conditions. The calculated film thickness from void fraction models and correlations and those from Whalley's annular flow model (1987) were compared each other using their experimental database. Dalkilic and Wongwises (2010b) used Barnea et al. (1982)'s mathematical model, based on the momentum balance of liquid and vapour phases, in order to determine the condensation film thickness of R134a in their paper. The discussions for the effects of heat flux, mass flux and condensation temperature on the film thickness and condensation heat transfer coefficient were also made for laminar and turbulent flow conditions. Six well-known flow regime maps from the literature were found to be in good agreement for the annular flow conditions in the test tube in spite of their different operating conditions. Dalkilic et al. (2010e) used Kosky and Staub's model (1971) to predict flow pattern transitions and validate the results of void fraction models and correlations proposed in their previous publications and also show the identification of flow regimes in data corresponding to annular flow downward condensation of R134a in a vertical smooth copper tube. Furthermore, investigation of twelve number of well-known flow regime correlations from the literature is performed to identify the flow regime occurring in the test tube. Dalkilic et al. (2010d) calculated the average predicted heat transfer coefficient of the refrigerant using Kosky and Staub's model (1971) and the Von Karman universal velocity distribution correlations by means of different interfacial shear stress equations valid for annular flow in horizontal and vertical tubes in order to validate

Chen et al.'s annular flow theory (1987). The discussions for the effects of heat flux, mass flux and condensation temperature on the pressure drop were also made in their paper. A new correlation including dimensionless parameters such as the equivalent Reynolds number, Prandtl number, R number (ρ - μ ratio), Lockhart and Martinelli parameter, Bond number and Froude number was proposed in their paper using a large number of data points for the determination of turbulent condensation heat transfer coefficient. Dalkilic et al. (2010c) proposed a new experimental approach on the determination of condensation heat transfer coefficient in a vertical tube by means of von Karman's universal velocity distribution and Kosky and Staub's annular flow film thickness model (1971). They benefitted from thirteen numbers of frictional pressure drop models and thirty five numbers of void fraction models in their model. Dalkilic and Wongwises (2009d) reported a detailed review of research on in-tube condensation by reason of its significance in refrigeration, air conditioning and heat pump applications. The heat transfer and pressure drop investigations for the in-tube condensation were included and almost all relevant research subjects were summarized, such as condensation heat transfer and pressure drop studies according to tube orientation (horizontal, vertical, inclined tubes) and tube geometry (smooth and enhanced tubes), flow pattern studies of condensation, void fraction studies, and refrigerants with oil. Besides to the above, various other conference papers (2008, 2009, 2010) were used to support and validate their proposed models and correlations in their papers. It can be seen from above studies, Dalkilic and Wongwises studied in-tube condensation process comprehensively using their experimental facility whose test section is working as a double-tube heat exchanger.

In spite of the existence of some available information in the literature, there still remains room for further research. As a result, the major aim in the present chapter of the book is to investigate the appliance of well-known empirical annular flow correlations to the annular flow condensation at high mass flux in a vertical double tube heat exchanger. The independency of annular flow heat transfer empirical correlations from tube orientation (1987) and the general applicability for a vertical short tube is also proved in this chapter.

2. Data reduction

Fig. 1 shows the steady-state physical model of downward film condensation in a vertical tube. Nusselt-type analysis is valid under some assumptions such as: laminar film flow; saturated state for the vapour of R134a; condensed film of R134a along the tube surface; constant physical properties corresponding to inlet pressure and temperature conditions; no entrainment. It should be noted that there is an interfacial shear effect at the interface occurred due to the much greater vapour velocity than the film velocity. The solution of the problem can be started from the calculation of the force balance in Eq. (1) for the differential element in the control volume neglecting the inertia and downstream diffusion contributions as shown in Dalkilic et al.'s study (2009c).

The force balance for the differential element in the control volume can be expressed as follows:

$$\rho_l g dx dy dz + \tau_\delta (y + dy) dx dz + P(z) dx dy = \tau_\delta (y) dx dz + P(z + dz) dx dy \quad (1)$$

It should be noted that the solution of the investigated case in this chapter will be different from Nusselt's solution (1916) due to the high mass flux condition of the condensate flow in the tube.

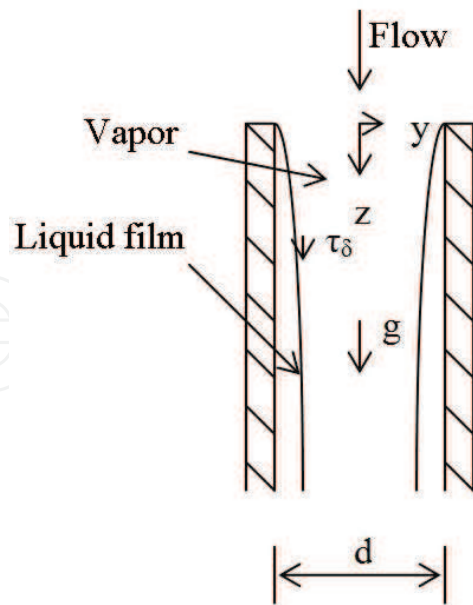


Fig. 1. System model for analysis of downward condensation

2.1 Experimental heat transfer coefficient

The details of determination of the experimental heat transfer coefficient by means of experimental setups can be seen in many papers given in references.

$$h_{\text{exp}} = \frac{Q_{\text{TS}}}{A_i(T_{\text{ref,sat}} - T_{w,i})} \quad (2)$$

where h_{exp} is the experimental average heat transfer coefficient, Q_{TS} is the heat transfer rate in the test section, $T_{w,i}$ is the average temperature of the inner wall, $T_{\text{ref,sat}}$ is the average temperature of the refrigerant at the test section inlet and outlet, and A_i is the inside surface area of the test section:

$$A_i = \pi dL \quad (3)$$

where d is the inside diameter of the test tube. L is the length of the test tube.

2.2 Uncertainties

The uncertainties of the Nusselt number and condensation heat transfer coefficient in the test tube, which belongs to a refrigerant loop consisting of an evaporator, test section and condenser, varied from $\pm 7.64\%$ to $\pm 10.71\%$. The procedures of Kline and McClintock (1953) were used for the calculation of all uncertainties. Various uncertainty values of the study can be seen from Table 1 in detail.

Based on this usual method, suppose that a set of measurement is made and the uncertainty in each may be expressed with same odds. These measurements are then used to calculate some desired result of the experiments. The result P is a given function of the independent variables $x_1, x_2, x_3, \dots, x_n$. Thus:

$$P = P(x_1, x_2, x_3, \dots, x_n) \quad (4)$$

Parameters	Uncertainty
$T_{ref, sat}$ (°C)	0.19
x_i	±6.96-8.24%
ΔT (K)	±0.191
$(T_{w,out}-T_{w,in})_{TS}$ (K)	±0.045
$(T_{w,in}-T_{w,out})_{ph}$ (K)	±0.13
m_{ref} (g s ⁻¹)	±0.023
$m_{w,TS}$ (g s ⁻¹)	±0.35
$m_{w,pre}$ (g s ⁻¹)	±0.38
q_{TS} (W m ⁻²)	±6.55-8.93%
q_{pre} (W m ⁻²)	±12-14.81%
h_{ref} (W m ⁻² K ⁻¹)	±7.64-10.71%.
ΔP (kPa)	±0.15

Table 1. Uncertainty of experimental parameters

Let w_p be the uncertainty in the result, and $w_1, w_2, w_3, \dots, w_n$ be the uncertainties in each independent variables. If the uncertainties in the independent variables are given with some odds, then the uncertainty in the result can be given as followed:

$$w_p = \pm \left[\left(\frac{\partial P}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial P}{\partial x_2} w_2 \right)^2 + \left(\frac{\partial P}{\partial x_3} w_3 \right)^2 + \dots + \left(\frac{\partial P}{\partial x_n} w_n \right)^2 \right]^{1/2} \quad (5)$$

2.2.1 Vapor quality (x_{in})

The vapor quality entering the test section is calculated from an energy balance, which gives the total heat transfer rate from hot water to liquid R134a in the evaporator as the sum of sensible and latent heat transfer rates, on the pre-heater:

$$x_{in} = \frac{1}{i_{ref,fg}} \left[\frac{Q}{m_{ref,T}} - C_{p,ref} (T_{ref,sat} - T_{ref,pre,in}) \right] \quad (6)$$

$$w_{x_{in}} = \pm \left[\left(\frac{\partial x_{in}}{\partial Q} w_Q \right)^2 + \left(\frac{\partial x_{in}}{\partial m_{ref,T}} w_m \right)^2 + \left(\frac{\partial x_{in}}{\partial T_{ref,sat}} w_{T_{ref,sat}} \right)^2 + \left(\frac{\partial x_{in}}{\partial T_{ref,pre,in}} w_{T_{ref,pre,in}} \right)^2 \right]^{1/2} \quad (7)$$

$$w_{x_{in}} = \pm \left[\left(\frac{1}{m_{ref,T} i_{ref,fg}} w_Q \right)^2 + \left(\frac{Q}{i_{ref,fg} m_{ref}^2} w_{m_{ref,T}} \right)^2 + \left(\frac{C_{p,ref}}{i_{ref,fg}} w_{T_{ref,sat}} \right)^2 + \left(\frac{C_{p,ref}}{i_{ref,fg}} w_{T_{ref,pre,in}} \right)^2 \right]^{1/2} \quad (8)$$

2.2.2 Heat flux (Q)

The outlet quality of R134a from the evaporator is equal to the quality of R134a at the test section inlet. The total heat transferred in the test section is determined from an energy balance on the cold water flow in the annulus:

$$Q = m_w C_p w (T_{w,out} - T_{w,in}) \quad (9)$$

$$w_Q = \pm \left[\left(\frac{\partial Q}{\partial m_w} w_{m_w} \right)^2 + \left(\frac{\partial Q}{\partial T_{w,in}} w_{T_{w,in}} \right)^2 + \left(\frac{\partial Q}{\partial T_{w,out}} w_{T_{w,out}} \right)^2 \right]^{1/2} \quad (10)$$

2.2.3 Average heat transfer coefficient (h_i)

The refrigerant side heat transfer coefficient in Eq. (2) is determined from the total heat transferred in the test section which is a vertical counter-flow tube-in-tube heat exchanger with refrigerant flowing in the inner tube and cooling water flowing in the annulus:

$$w_{h_i} = \left[\left(\frac{\partial h_i}{\partial Q} w_Q + \frac{\partial h_i}{\partial (T_{ref,sat} - T_{ref,w,i})} w_{(T_{ref,sat} - T_{ref,w,i})} \right)^2 \right]^{1/2} \quad (11)$$

2.2.4 Nusselt number (Nu)

Nusselt number can be expressed as follows:

$$Nu = \frac{h_i L}{k_l} \quad (12)$$

$$w_{Nu} = \pm \left[\left(\frac{\partial Nu}{\partial h_i} w_{h_i} \right)^2 \right]^{1/2} \quad (13)$$

2.2.5 Temperature difference

The temperature measurements for the determination of outer surface temperature of the test tube are evaluated as an average value for the uncertainty analysis and calculation procedure can be seen as follows:

$$T_0 = \frac{T_1 + T_2 + T_3 + \dots + T_{10}}{10} \quad (14)$$

$$w_{T_0} = \pm \left[\left(\frac{\partial T_0}{\partial T_1} w_{T_1} \right)^2 + \left(\frac{\partial T_0}{\partial T_2} w_{T_2} \right)^2 + \dots + \left(\frac{\partial T_0}{\partial T_{10}} w_{T_{10}} \right)^2 \right]^{1/2} \quad (15)$$

$$w_{T_0} = \pm \left[\left(\frac{1}{10} w_{T_1} \right)^2 + \left(\frac{1}{10} w_{T_2} \right)^2 + \dots + \left(\frac{1}{10} w_{T_{10}} \right)^2 \right]^{1/2} \quad (16)$$

3. Heat transfer correlations in this chapter

The comparison of convective heat transfer coefficients with Shah correlation (1979) has been done by researchers commonly for turbulent condensation conditions especially in vertical tubes (2000) and it is found to be the most comparative condensation model during annular flow regime in a tube (2002). Shah correlation (1979) is based on the liquid heat transfer coefficient and is valid for $Re_1 \geq 350$. A two-phase multiplier is used for the annular flow regime of high pressure steam and refrigerants (2000, 2002) in the equation.

Dobson and Chato (1998) used a two-phase multiplier to develop a correlation for an annular flow regime. A correlation is also provided for a wavy flow regime. The researchers used commonly their correlations for zeotropic refrigerants. Its validated mass flux (G) covers the values more than $500 \text{ kg m}^{-2} \text{ s}^{-1}$ for all qualities in horizontal tubes.

Sweeney (1996) developed the Dobson and Chato model (1998) using zeotropic mixtures for annular flow.

Cavallini et al. (1974) used various organic refrigerants for the condensation inside tubes in both vertical and horizontal orientations and developed a semi empirical correlation as a result of their study.

Bivens and Yokozeki (1994) modified Shah correlation (1979) benefitted from various flow patterns of R22, R502, R32/R134a, R32/R125/R134a.

Tang et al. (2000) developed the Shah [(1979) equation for the annular flow condensation of R410A, R134a and R22 in i.d. 8.81 mm tube with $Fr_{so} > 7$.

Fujii (1995) modified the correlation in Table 2 for shear-controlled regimes in smooth tubes. There is also another correlation belong to Fujii (1995) for gravity controlled regimes.

Chato (1961) used a two-phase multiplier to modify Dittus-Boelter's correlation (1930) for an annular flow regime.

Traviss et al. (1972) focused on the flow regime maps for condensation inside tubes. Their correlation was suggested for the condensation of R134a inside tubes specifically. Their model takes into account of the variation in the quality of the refrigerant using Lockhart-Martinelli parameter.

Akers and Rosson (1960) developed the Dittus-Boelter (1930)'s single-phase forced convection correlation. Their correlation's validity range covers the turbulent annular flow in small diameter circular tubes and rectangular channels.

Tandon et al. (1995) developed the Akers and Rosson (1960) correlation for shear controlled annular and semi-annular flows with $Re_g > 30000$. There is another correlation of Tandon et al. (1995) for gravity-controlled wavy flows with $Re_g < 30000$.

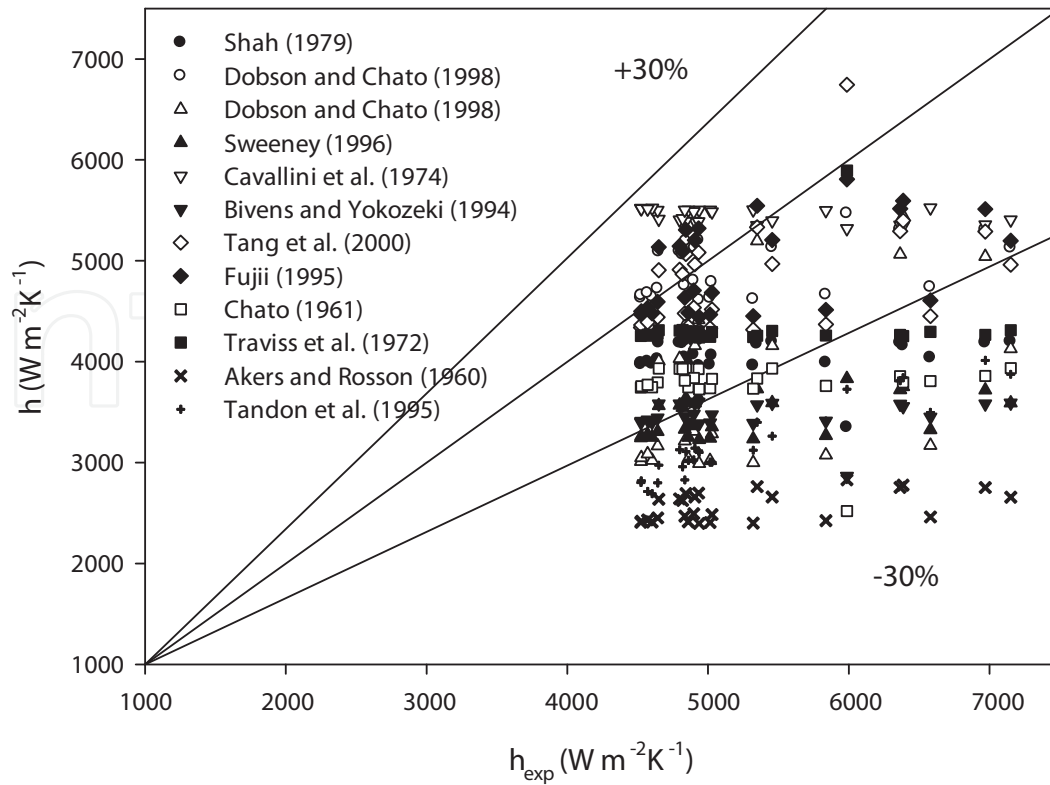
4. Results and discussion

It should be noted that it is possible for researcher to identify the experimental data of condensation by means of both flow regime maps and sight glass at the inlet and outlet of the test section. Dalkilic and Wongwises (2009b, 2010b, 2010e) checked the data shown in all figures and formulas that they were in an annular flow regime by Hewitt and Robertson's (1969) flow pattern map and also by sight glass in their experimental setup. The vapor quality range approximately between 0.7-0.95 in the 0.5 m long test tube was kept in order to obtain annular flow conditions at various high mass fluxes of R134a.

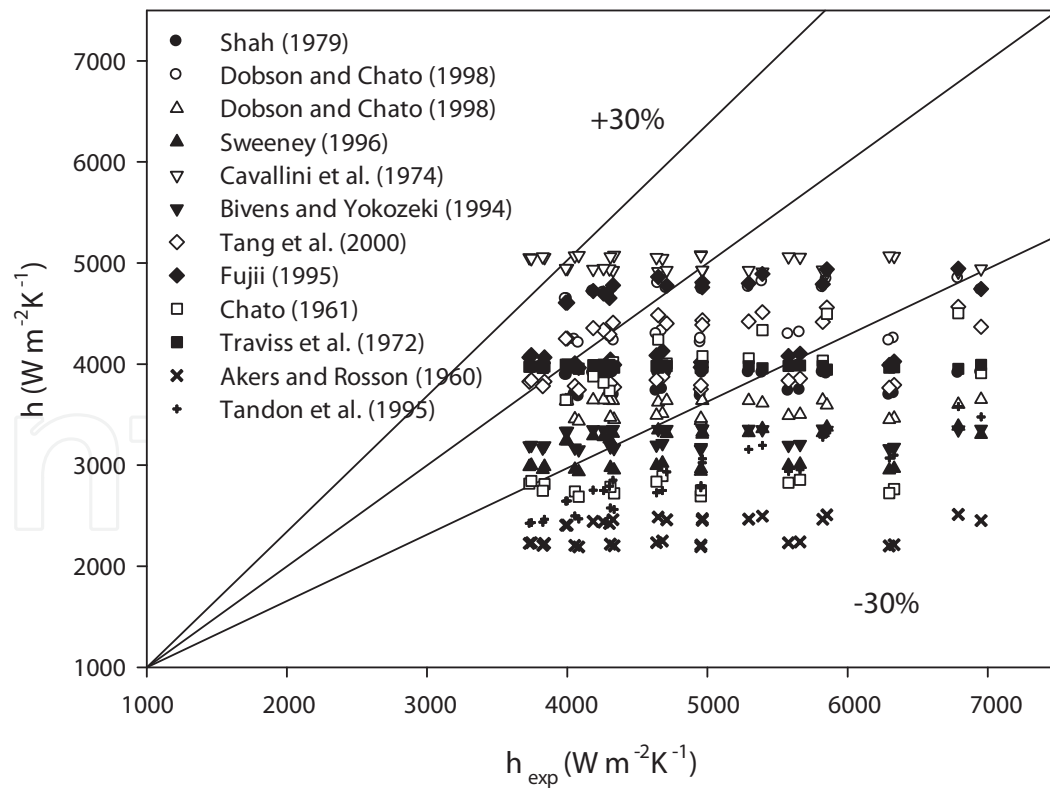
In Table 2, the list of correlations is evaluated to show the similarity of annular flow correlations which are independent of tube orientation (horizontal or vertical). Chen et al. (1987) also developed a general correlation to discuss this similarity in their article by relating the interfacial shear stress to flow conditions for annular film condensation inside tubes.

Researcher	Model/Correlation
Shah (1979)	$Re_l \geq 350 \quad Re_l = \frac{Gd(1-x)}{\mu_l} \quad h_{shah} = h_{sf} \left(\frac{1.8}{Co^{0.8}} \right) \quad P_{red} = \frac{P_{sat}}{P_{critic}}$ $h_{sf} = h_l(1-x)^{0.8} \quad Co = \left(\frac{1}{x} - 1 \right)^{0.8} \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \quad h_l = \frac{k_l}{d} \left(0.023 \left(\frac{Re_l}{1-x} \right)^{0.8} Pr_l^{0.4} \right)$
Dobson and Chato (1998)	<p>For $G > 500 \text{ kg m}^{-2} \text{ s}^{-1}$ $Nu = 0.023 Re_l^{0.8} Pr_l^{0.4} \left[1 + \frac{2.22}{X^{0.89}} \right]$ $Ga = \frac{\rho_l(\rho_l - \rho_g)gd^3}{\mu_l^2}$</p> $X = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_g} \right)^{0.1} \quad Fr_{so} = c_3 Re_l^{c_4} \left(\frac{1+1.09X^{0.039}}{X} \right)^{1.5} \frac{1}{Ga^{0.5}}$ $Re_l = \frac{Gd(1-x)}{\mu_l} \quad \text{For } Re_l > 1250 \text{ and } Fr_{so} > 18 \quad Nu = 0.023 Re_l^{0.8} Pr_l^{0.3} \frac{2.61}{X^{0.805}}$
Sweeney (1996)	$Nu = 0.7 \left(\frac{G}{300} \right)^{0.3} Nu_{Dobson-Chato}$
Cavallini et al. (1974)	$Nu_l = 0.05 Re_{eq}^{0.8} Pr^{0.33} \quad Re_{eq} = Re_g \left(\mu_g / \mu_l \right) \left(\rho_l / \rho_g \right)^{0.5} Re_l \quad Re_g = Gdx / \mu_g$
Bivens and Yokozeki (1994)	$Nu = Nu_{Shah} \left(0.78738 + \frac{6187.89}{G^2} \right) \quad P_{red} = P_{sat} / P_{critic}$ $Nu_{Shah} = 0.023 Re_l^{0.8} Pr_l^{0.4} \left[1 + \frac{3.8}{P_{red}^{0.38}} \left(\frac{x}{1-x} \right)^{0.76} \right]$
Tang et al. (2000)	$Nu = 0.023 Re_l^{0.8} Pr_l^{0.4} \left[1 + 4.863 \left(\frac{-\ln(P_{red})}{1-x} \right)^{0.836} \right]$ $Re_l \leq 1250 \quad c_3 = 0.025 \quad c_4 = 1.59 \quad Fr_{so} = c_3 Re_l^{c_4} \left(\frac{1+1.09X^{0.039}}{X} \right)^{1.5} \frac{1}{Ga^{0.5}}$ $Re_l > 1250 \quad c_3 = 1.26 \quad c_4 = 1.04$
Fujii (1995)	$Nu_l = 0.0125 \left(Re_l \sqrt{\rho_l / \rho_g} \right)^{0.9} \left(\frac{x}{1-x} \right)^{0.1x+0.8} Pr_l^{0.63}$
Chato (1961)	$Nu = 0.023 Re_l^{0.8} Pr_l^{0.4} \left[\frac{2.47}{X^{1.96}} \right]$
Traviss et al. (1972)	$Nu = Re_l^{0.9} Pr_l \frac{F_1(X)}{F_2(Re_l, Pr_l)} \quad F_1(X) = 0.15 \left[\frac{1}{X} + \frac{2.83}{X^{0.476}} \right]$ $Re_l > 1125 \quad F_2 = 5Pr_l + 5\ln(1 + 5Pr_l) + 2.5\ln(0.00313 Re_l^{0.812})$
Akers and Rosson (1960)	$Nu = 0.0265 Re_{eq}^{0.8} Pr_l^{1/3} \quad Re_{eq} = \frac{G_{eq}d}{\mu_l} \quad G_{eq} = G \left[(1-x) + x \left(\rho_l / \rho_g \right)^{0.5} \right]$
Tandon et al. (1995)	$Re_g > 30000 \quad Nu = 0.084 Re_g^{0.67} Pr_l^{1/3} \left[\frac{1}{Ja_l} \right]^{1/6} \quad Ja_l = \frac{Cp_l \Delta T_{sat}}{i_{fg}}$

Table 2. Annular flow heat transfer correlations and models



(a)



(b)

Fig. 2. Comparison of experimental condensation heat transfer coefficient vs. various correlations for $G=300 \text{ kg m}^{-2} \text{ s}^{-1}$ (a) $T_{\text{sat}}=40 \text{ }^\circ\text{C}$ and (b) $T_{\text{sat}}=50 \text{ }^\circ\text{C}$

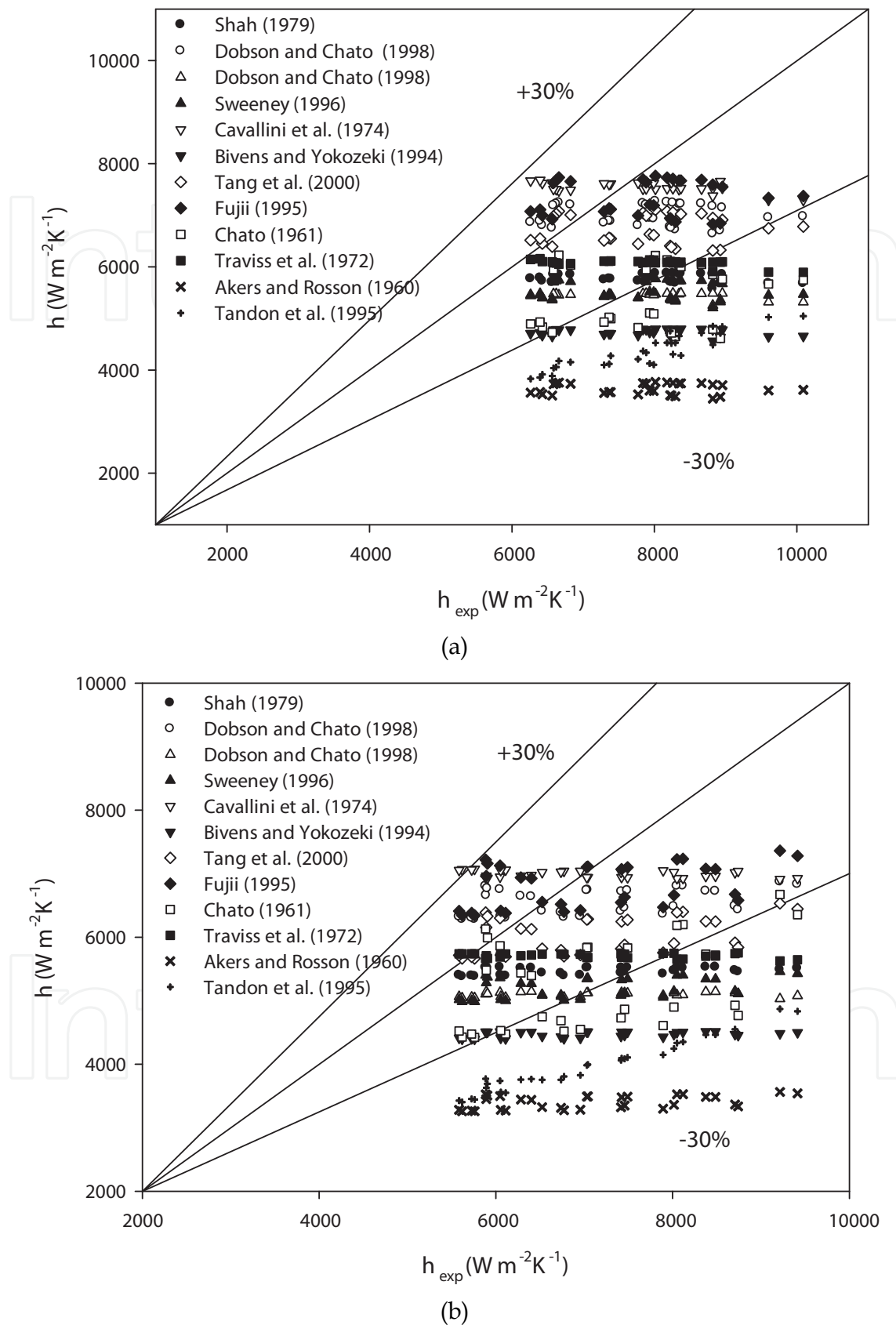


Fig. 3. Comparison of experimental condensation heat transfer coefficient vs. various correlations for $G=456\text{ kg m}^{-2}\text{ s}^{-1}$ (a) $T_{sat}=40\text{ }^{\circ}\text{C}$ and (b) $T_{sat}=50\text{ }^{\circ}\text{C}$

The comparison of experimental heat transfer coefficients with various annular flow correlations are shown in Figs. 2-3 in a 30% deviation line for the condensation temperatures of 40 and 50 °C and mass fluxes of 300 and 456 kg m⁻²s⁻¹ respectively. It can be clearly seen from these figures that the Dobson and Chato (1998) correlation, Cavallini et al. (1974) correlation, Fujii (1995) correlation are in good agreement with the experimental data. In addition to this, the majority of the data calculated by Shah (1979) correlation, Dobson and Chato (1998) correlation, Tang et al. (2000) correlation, Traviss et al. (1972) correlation fall within ±30%, whereas, Tandon et al. (1995) correlation and Akers and Rosson (1960) correlation are found to be incompatible with the experimental data. Nonetheless, the Chato (1961) correlation, the Sweeny (1996) correlation, and the Bivens and Yokozeki (1994) correlation are found to have poor agreement with experimental data.

In the literature, some correlations are developed for gravity-controlled regimes such as Fujii (1995) and Tandon et al. (1995) correlations. In this chapter, the correlations, proposed for gravity-controlled regimes, are not found to be in good agreement with the data as expected for annular flow regime. These kinds of correlations do not exist in this chapter due to their large deviations and validity for wavy flow. Chen et al. (1987) reported that the vapor shear stress acting on the interface of vapor-liquid phases affects the forced convective condensation inside tubes especially at high vapor flow rates. On that account, Fujii (1995) and Tandon et al. (1995)'s shear-controlled correlations were used to predict condensation heat transfer coefficient of R134a. Furthermore, Valladares (2003) obtained similar results on these explanations in this chapter for the condensation of various refrigerants in horizontal tubes by Valladares (2003).

5. Conclusion

In this chapter of the book, the method to determine the average heat transfer coefficient of R134a during condensation in vertical downward flow at high mass flux in a smooth tube is proposed. The comparison between the various annular flow heat transfer correlations and experimental heat transfer coefficients is shown with ±30% deviation line. It can be noted that the Dobson and Chato (1998) correlation, the Cavallini et al. (1974) correlation, the Fujii (1995) correlation are found to have the most predictive results than others in an 8.1 mm i.d. copper tube for the mass fluxes of 300 and 456 kg m⁻²s⁻¹ and condensation temperatures of 40 and 50 °C. As a result of the analysis in this chapter, it is proven that annular flow models are independent of tube orientation provided that annular flow regime exists along the tube length and capable of predicting condensation heat transfer coefficients inside the vertical test tube although most of these correlations were developed for the annular flow condensation in horizontal tubes.

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7. Nomenclature

A	surface area, m ²
C _p	specific heat, J kg ⁻¹ K ⁻¹
d	internal tube diameter, m
Fr	Froude number
G	mass flux, kg m ⁻² s ⁻¹
Ga	Galileo number
g	gravitational constant, m s ⁻²
h	convective heat transfer coefficient, W m ⁻² K ⁻¹
i	enthalpy, J kg ⁻¹
i _{fg}	latent heat of condensation, J kg ⁻¹
Ja	Jakob number
k	thermal conductivity, W m ⁻¹ K ⁻¹
L	length of test tube, m
m	mass flow rate, kg s ⁻¹
Nu	Nusselt number
P	pressure, MPa
Pr	Prandtl number
Re	Reynolds number
T	temperature
Q	heat transfer rate, W
q	mean heat flux, kW m ⁻²
y	wall coordinate
x	mean vapor quality
y	radial coordinate
z	axial coordinate
X	Lockhart-Martinelli parameter
w	uncertainty

Greek Symbols

ΔT	vapor side temperature difference, T _{ref,sat} - T _{wi} , °C
ρ	density, kg m ⁻³
μ	dynamic viscosity, kg m ⁻¹ s ⁻¹
τ	shear stress, N m ⁻²

Subscripts

eq	equivalent
exp	experimental
g	gas/vapor
i	inside
in	inlet
l	liquid
out	outlet
pre	preheater
red	reduced
ref	refrigerant

sat	saturation
so	Soliman
T	total
TS	test section
w	water
w_i	inner wall
δ	film thickness

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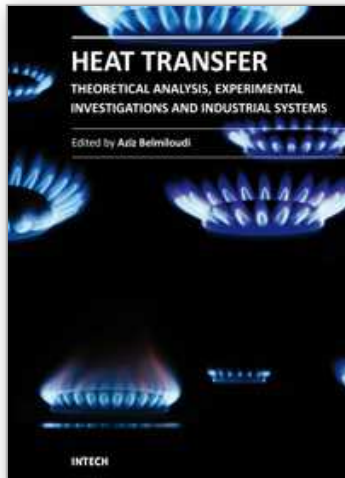
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