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A Comparative Study of Heat Transfer Coefficients for Film Condensation

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Abstract

Film condensation heat transfer has wide applications in a variety of industrial systems. A number of film condensation heat transfer correlations (FCHTCs) have been proposed. However, their predictions are often inconsistent. This paper presents a comparative study of existing FCHTCs. Totally 1214 experimental data points are obtained from 10 published papers, and 14 FCHTCs are reviewed, among which four correlations are used for horizontal flow outside smooth tubes, three for flow on vertical surfaces of plates or tubes, two for flow inside smooth tubes either vertically or horizontally, and five for horizontal flow inside smooth tubes. 13 FCHTCs are compared with the experimental data. There are three FCHTCs for horizontal flow inside smooth tubes having a mean absolute relative deviation (MARD) less than 26%, among which the best one has an MARD of 22.2%. More efforts should be made to develop better correlations.

Key words: Correlation; Heat transfer; Film; Condensation

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NOMENCLATURE

A	surface area (m^2)
Co	convective number, $(1/x\text{-}1)^{0.8}(\rho g/\rho l)^{0.5}$
cp	specific heat capacity (J/ kg℃)
D	diameter of test-section (m)
Fr	Froude number
g	gravitational acceleration (m/s^2)
G	mass flux (kg/m^{2}s)
h	heat transfer coefficient (W/m^2•K)
h_{fg}	latent heat of condensation (J/kg), $h_{fg}=h'_g-h'_l$
h'	enthalpy(J/kg)
k	thermal conductivity of condensate (W/m^2℃)
L	length of test-section (m)
Pr	Prandtl number
Re	Reynolds number
T	temperature (℃or K)
x	mean vapor quality
X_{tt}	Lockhart-Martinelli parameter

Greek Symbols

α	void fraction
θ	liquid level angle (rad)
μ	dynamic viscosity (Pa•s)
η	fin efficiency
ρ	density (kg/m^3)
φ	enlargement factor

Subscripts

b	bottom
g	vapor phase
i	inside
l	liquid phase
o	outside
r	refrigerant
s	saturation
t	top
w	tube or plate wall

INTRODUCTION

Film condensation heat transfer (FCHT) in tubes occurs in many industrial applications, including refrigeration, air conditioning, electric power generation, marine propulsion, as well as chemical process industries. There are a large number of papers (Singh et al.,2009; Fujii, 1995; Cavallini et al., 2003; Park et al., 2011; Matkovic et al., 2009; Jung et al., 2003; Sapali & Patil, 2010; Suliman et al., 2009; Laohalertdecha & Wongwises, 2010; Dalkilic et al., 2009; Aprea et al., 2003; Park et al., 2008; Wen et al., 2006; Park et al., 2011; Pega & Hrnjak, 2009) investigating the influence of condensation film on condensation heat transfer in tubes. When condensation occurs on horizontal and short vertical plates, condensate film motion is generally laminar. On vertical tubes and long vertical plates, film motion can become turbulent. Grober et al. (1961) suggested that a Reynolds number of 1600 could be used as the critical point at which the flow pattern changes from laminar to turbulent. In practice, condensation is usually laminar in shell-and-tube condensers with the vapor outside the horizontal tubes.

There are many papers which investigated experimentally the FCHT characteristics as the function of mass flux, heat flux, saturation temperature and tube diameter.

Laohalertdecha and Wongwises (2010) studied the heat transfer coefficient (HTC) of R-134a inside a horizontal smooth tube and corrugated tubes with the inner diameter of 8.7 mm, outer diameter of 9.52 mm, and test section length of 2000 mm. The measured parameter ranges are the saturation temperatures of 40, 45, and 50℃ respectively, heat fluxes of 5 and 10 kW/m^2 respectively, vapor quality of 0.1–0.9, and mass fluxes ranging from 200 to 700 kg/m^{2}s. The corrugation pitches are 5.08, 6.35, and 8.46 mm, respectively. The corrugation depth of all corrugated tubes is fixed at 1.5 mm. The results revealed that the average HTC increased with increasing mass flux and average quality. The ratio of the convective HTC of the smooth tube to that of the corrugated tube varied from 0.67 to 0.91 under the same average quality. The average condensation HTC in corrugated tubes was higher than that in the smooth tube at the same average quality.

Dalkilic et al. (2009) studied the two-phase HTC of pure HFC-134a condensing inside a smooth tube-in-tube heat exchanger. The inner tube is constructed from smooth copper tubing of 9.52 mm outer diameter, 8.1 mm inner diameter and 0.5 m length. The data ranges are average saturation condensing temperatures of 40–50 °C, the mass fluxes of 260 and 515 kg/m^{2}s, vapor quality of 0.7–0.95, and the heat fluxes of 11.3 and 55.3 kW/m^2, respectively. The result was that the average HTC of R134a decreased with decreasing mass flux and increasing condensation temperature and increased with increasing average quality.

Aprea et al. (2003) studied the local HTC of R22 and R407C in the coaxial counter-flow condenser of a refrigerating vapor compression plant of 22 mm outer diameter, 20 mm inner diameter and 6.6 m length. The experimental parameter ranges are saturation temperatures of 36.6–39.6 °C, the mass fluxes of 45.5–120kg/m^{2}s, vapor quality of 0–0.9, pressure of 15.2–14.3 bar and the heat fluxes of 5.6–38.0 kW/m^2. They believed that the HTC of R22 was always greater than that of R407C and the difference ranges from 7 to 30%. The HTC depended on the vapor quality strongly when the mass fluxes were high.

Park et al. (2008) discussed flow condensation HTCs of R22, propylene, propane, DME and isobutane on a horizontal plain copper tube of 8.8 mm inner diameter and 530 mm length. The data ranges are saturation temperature of 40 ± 0.2℃, mass fluxes of 100, 200, and 300 kg/m^{2}s, heat fluxes of 7.3–7.7 kW/m^2 and vapor quality of 0.09–0.91. Test results showed that some well-known correlations developed based on conventional fluorocarbon refrigerants predicted the present data within a mean deviation of 33%. The flow condensation HTCs for propylene, propane, DME and isobutane were about 46.8%, 53.3%, 93.5% and 61.6% larger as compared with that for R22 for a given mass flux, respectively.

Wen et al. (2006) investigated heat transfer behavior during condensation of R-600, R-600/R-290 (50wt.%/50wt.%) and R-290 in the three-line serpentine small-diameter tube bank of 3.18 mm outer diameter, 2.46 mm inner diameter and 3.85 m length. The measured parameter ranges are heat flux of 5.2 kW/m^2, mass fluxes of 205–510 kg/m^{2}s, saturation temperatures of 40℃, and vapor quality of 0.15–0.84. They found that Dobson and Chato (1998) correlation had the best predictability with an average standard deviation of 12.8%. The HTCs of R-134a were lower than those of R-600, R-290/R-600 and R-290 by 155%, 124% and 89% at the same conditions.

Park et al. (2011) studied the experimental condensation heat transfer data for the new refrigerant R1234ze(E), trans-1,3,3,3-tetraflu-oropropene, and compared with refrigerants R134a and R236fa for a vertically aligned, aluminum multi-port tube of 1.45 mm hydraulic diameter and 260 mm length. The experimental data ranges are vapor quality of 0.0–1.0, mass fluxes of 50–260 kg/m^{2}s, saturation temperature of 25–70℃, and heat fluxes of 1–62 kW/m^2. It was found that in general, some correlations under-predicted the low Nusselt number data and over-predicted the high Nusselt number data, but captured the mid-range quite well. The heat transfer performance of R1234ze (E) was relatively similar to R236fa but about 15–25% lower than that of R134a.

Cavallini et al. (2001) researched experimentally condensation HTCs inside a smooth tube of 8 mm inner diameter, 12mm outer diameter, and 1000 mm length with pure HFC refrigerants (R134a, R125, R236ea, R32) and the nearly azeotropic HFC refrigerant blend R410A. The experimental parameter ranges are saturation temperature of 30–50℃, mass fluxes of 100–750 kg/m^{2}s, vapor quality of 0.15–0.85, and heat fluxes of 4.21–93.1 kW/m^2. They found that the heat transfer performance of R125 which was enhanced by the good thermal properties of R32 was lower than that of R410A, and that high pressure refrigerants performed worse than low and mid-pressure refrigerants.

June et al. (1999) investigated the HTCs of a plain tube and low fin tube of 15.9 mm outer diameter for the low pressure refrigerants CFC11 and HCFC123 and for the medium pressure refrigerants CFC12 and HFC134a. The nominal outside diameter, fin height and fins of the fin tube are 18.9 mm, 1.214 mm and 1024/m, respectively. They found that the HTCs of HFC123, an alternative for CFC11, were 8.2–19.2% lower than those of CFC11 for all the tubes tested, and the HTCs of HFC134a, an alternative for CFC12, were 0.0–31.8% higher than those of CFC12 for all the tubes tested.

Belghazi et al. (2001) studied the condensation of pure HFC134a and different zeotropic mixtures with pure HF-C134a and HFC23 on the outside of a bundle of smooth tubes of 14.2 mm and 16.8 mm inner diameter and outer diameter. The experimental date ranges are inlet vapor temperature of 40℃, heat fluxes of 5–30 kW/m^2 and the cooling water flow rate of 120–300 l/h.

Park and Hrnjak (2009) discussed CO_2 flow condensation HTCs for micro channels of 0.89 mm inner diameter and 500 mm length at horizontal flow conditions. The parameter ranges are saturation temperatures of –15 and –25℃, mass fluxes of 200–800 kg/m^{2}s, wall sub-cooling temperatures of 2–4℃, vapor quality of 0.085–0.91, and heat fluxes of 6.17–28.7 kW/m^2. They found that measured HTCs increased with the increase of mass fluxes and vapor qualities, whereas they were almost independent of wall sub-cooling temperature changes. They compared the experimental results with the correlations given by Akers et al. (1959), Soliman et al. (1968), Traviss et al. (1973), Jaster and Kosky (1976), Shah (1979), Chen et al. (1987), Dobson and Chato (1998), Cavallini et al. (2003), Koyama et al. (2003), and Thome et al. (2003), and found that the Akers et al. (1959) correlation showed acceptable predictions for the mass fluxes from 400 to 800 kg/m^{2}s, and that the Akers et al. (1959) and Thome et al. (2003) correlations had the MARD less than 20%.

Laohalertdecha and Wongwises (2010) compared the experimental data for smooth tube with the correlations proposed by Aker et al. (1959), Traviss et al. (1973), Cavallini and Zechin (1974), Chen et al. (1987), Dobson and Chato (1998), Nualboonrueng et al. (2003), and Bassi and Bansal (2003). They found that almost all the correlations could give an agreement with the their data to be within 30%.

Dalkilic et al. (2009) compared the experimental HTCs with various annular flow correlations, such as Akers and Rosson (1960), Chato (1961), Traviss et al. (1973), Cavallini et al. (1974), Shah (1979), Bivens and Yokozeki (1994), Tandon et al. (1995), Dobson and Chato (1998), Sweeney (1998), Tang et al. (2000), and Fujii (2009). They found that the correlations of Cavallini et al. (1974), Dobson and Chato (1998), and Fujii (2009) had the best predictability, that the Chato (1961) correlation, the Bivens and Yokozeki (1994) correlation, and the Sweeny (1998) correlation had poor predictions, and that the correlation s of Traviss et al. (1972), Shah (1979), and Tang et al. (2000) had a mean deviation within ±30% for the majority of the data. The Akers and Rosson (1960) correlation and Tandon et al. (1995) correlation were incompatible with the experimental data.

Park et al. (2008) compared their experimental data against some of the well-known correlations by Akers et al. (1959), Soliman et al. (1968), Traviss et al. (1973), Cavallini and Zecchin (1974), Shah (1979), Dobson and Chato (1998), and Jung et al. (2003). They found that all correlations predicted the data reasonably well within an error range of 30%. Especially, the correlation by Jung et al. (2003) predicted both fluorocarbon and hydrocarbon refrigerants and DME data within 12%.

Park et al. (2011) compared the experimental results with the macro-scale annular flow method of Thome et al. (2003), the micro-channel heat transfer correlations of Moser et al. (1998), Koyama et al. (2003), Bandhauer et al. (2006), and Cavallini et al. (2006), and the falling film flow method of Nusselt theory. They found that the Bandhauer (2006) and Koyama et al. (2003) methods showed the most compact distribution within a narrow range around a single line tilted from the centerline of the comparison. In addition, the Bandhaueret al. (2006) method showed the best results with a MARD of 22.8%.

It can be seen from the above brief introduction that comparisons of existing correlations with experimental data made by different authors presented inconsistent results. The experimental data used for their studies were not abundant, which might be the main reason which caused the inconsistence. Therefore, a comprehensive evaluation is necessary. This paper conducts an overall review of the published literature pertaining to FCHTCs and performs an up-to-date survey of the experimental studies. Based on the experimental data obtained from the published literature, a comparative study of the existing correlations is carried out. The method used in this paper is similar to that used by Fang et al. (2011, 2012) and Fang and Xu (2011).

1. REVIEW ON FILM CONDENSATION HEAT TRANSFER CORRELATIONS

A large number of FCHTCs have been proposed to predict the heat transfer of pure substances and mixture condensing in or on horizontal and vertical tubes, or on vertical plates. Here, 14 FCHTCs are obtained through literature review, as listed in the following. In general, they were proposed and developed primarily depended on the author's own experimental data.

1.1 Film Condensation Outside Horizontal Tubes

1.1.1 Nusselt (1916) Correlation 1

Nusselt (1916) proposed the first theoretical solution for predicting FCHT of single smooth tubes. Waves and an interfacial shear effect between the phases were not considered.

Nusselt's equation for flow outside a smooth horizontal tube is shown as following:

$$h = 0.725\left[\frac{\rho_l(\rho_l-\rho_g)gh_{fg}k_l^3}{\mu_l D_o(T_{r,s}-T_w)}\right]^{\frac{1}{4}} \tag{1a}$$

$$h_{fg} = h_g' - h_l' \tag{1b}$$

where h' is the specific enthalpy of refrigerant, (J /kg).

1.1.2 Dhir and Lienhard (1971) Correlation

Dhir and Lienhard (1971) proposed an average HTC for laminar film condensation outside single horizontal tube as

$$h = 0.729\left[\frac{\rho_l(\rho_l-\rho_g)gh_{fg}k_l^3}{\mu_l D_o(T_{r,s}-T_w)}\right]^{\frac{1}{4}} \tag{2}$$

1.1.3 Incropera and DeWitt (2002) Correlation

Incropera and DeWitt (2002) proposed an average HTC of laminar film condensation on vertically aligned horizontal tube bundles as

$$h = h_D N^{-\frac{1}{4}} \tag{3}$$

where N is the number of the tubes of the bundle, and h_D is the HTC calculated with Eq. (2).

1.1.4 Beatty and Katz (1948) Correlation

Beatty and Katz (1948) were the first to develop an analytical model based on the Nusselt analysis for condensation on low-finned tubes. This correlation is acceptable for low-surface-tension fluids and laminar condensate film motion on horizontal low-fin-density tubes.

$$h = 0.689F^{0.25}\left[\frac{A_r}{A}\frac{1}{D_r^{0..25}} + 1.3\eta\frac{A_f}{A}\frac{1}{L_c^{0.25}}\right] \tag{4a}$$

where η is the fin efficiency which was assumed to be 1 for the low fin tube, A is the tube surface area at nominal diameter (m^2), A_r is the tube surface area at the base of the fins (m^2), A_f is the fin surface area (m^2), D_r is the diameter at fin root (i.e., smooth tube outer diameter), F is a constant accounting for the effect of physical properties, and L_c is the characteristic length.

$$F = \left(\frac{\rho_l^2 g k_l^3 h_{fg}}{\mu_l(T_{r,s}-T_w)}\right) \tag{4b}$$

$$L_c = \frac{\pi(D_o^2 - D_r^2)}{4D_o} \tag{4c}$$

$A=A_r+A_f$(4d)

1.2 Flow onVertical Surfaces of Plate or Tube with Large Diameter

1.2.1 Nusselt (1916) Correlation 2

Nusselt's theory for condensation can also be used for condensation outside the tubes if the tubes are large in diameter, compared with the film thickness. He assumed a linear temperature profile through a laminar film flowing downwards without entrainment on a vertical tube. According to Nusselt (1916), the HTC correlation used for vertical flow is as the following ($0 < Re < 30$):

$$h = 0.943\left[\frac{\rho_l(\rho_l-\rho_g)gh_{fg}k_l^3}{\mu_l L(T_{r,s}-T_w)}\right]^{\frac{1}{4}} \tag{5}$$

1.2.2 Longo and Gasparella (2007) Correlation

For saturated vapor condensation, Longo and Gasparella proposed a FCHTC for vertical laminar motion as

$$h=\varphi h_N \tag{6}$$

where h_N is evaluated with Eq. (5), and φ is the enlargement factor which equals 1.24 for the plate heat exchanger.

1.2.3 McAdams (1954) Correction

For turbulent flow, the McAdams (1954) correction factor can be used to consider the effects of the waviness and rippling in the film on the increase in heat transfer.

$$h = 0.0077\left[\frac{\rho_l(\rho_l-\rho_g)gk_l^3}{\mu_l^2}\right]^{\frac{1}{3}} Re_l^{0.4} \tag{7a}$$

$$Re_l = \frac{GD_i(1-x)}{\mu_l} \tag{7b}$$

1.3 Internal Flow in Round Tubes

1.3.1 Annual Flow with Uniform Film Distribution (Horizontal or Vertical)

1.3.1.1 Jung et al (2003) Correlation

Jung et al. (2003) condensed many refrigerants such as R12, R22, R32, R123, R125, R134a, and R142b inside a smooth tube, compared the experimental data with various well-known correlations, and proposed a new correlation as the following:

$$h = 22.4h_l\left(1+\frac{2}{X_{tt}}\right)^{0.81}\left(\frac{Q}{h_{fg}GA}\right)^{0.33} \tag{8a}$$

$$h_l = 0.023Re_l^{0.8}Pr_l^{0.4}\left(\frac{k_l}{D_i}\right) \tag{8b}$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1} \tag{8c}$$

$$Pr_l = \frac{\mu_l c_{pl}}{k_l} \tag{8d}$$

where A, Q, Re_l and Pr_l are the area of test section (m^2), heat transfer rate (W), the Reynolds number evaluated by Eq. (7b), and the Prandtl number, respectively.

1.3.1.2 Shah (1979) Correlation

The Shah (1979) correlation has been compared by researchers commonly for turbulent condensation conditions valid for $Re_l \geq 350$ and is considered to be the most comparative condensation model during annular flow with uniform film distribution in horizontal or vertical tubes. It has a two-phase multiplier for annular flow regime of high pressure steam and refrigerants.

$$h = h_l\left(\frac{1.8}{Co^{0.8}}\right) \tag{9a}$$

where h_l is calculated with Eq. (8b), and convective number Co is defined as

$$Co = \left(\frac{1}{x} - 1\right)^{0.8}\left(\frac{\rho_g}{\rho_l}\right)^{0.5} \tag{9b}$$

1.3.2 Horizontal Stratified Wavy Flow

1.3.2.1 Akers et al. (1959) Correlation

The characteristics of the heat transfer mechanism for forced convection condensation can be explained by the Akers et al. (1959) correlation, presented in Eq. (12), which was developed for annular flow regime.

$$h = 0.026 Pr_l^{\frac{1}{3}} Re_l^{0.8} \frac{k_l}{D_i}\left[\frac{x}{1-x}\left(\frac{\rho_l}{\rho_g}\right)^{0.5} + 1\right] \tag{10}$$

1.3.2.2 Cavallini et al. (2006) Correlation

Cavallini et al. (2006) proposed a HTC for film condensation inside horizontal tubes. The model includes two different flow categories: △T-dependent and △T-independent flow regime.

For △T-independent flow regime ($J_g > J_{tg}$)

$$h_I = h_{lo}\left[1 + 1.128 x^{0.817}\left(\frac{\rho_l}{\rho_g}\right)^{0.3685}\left(\frac{\mu_l}{\mu_g}\right)^{0.2363}\left(1 - \frac{\mu_g}{\mu_l}\right)^{2.144} Pr_l^{-0.1}\right] \tag{11a}$$

For △T-dependent flow regime ($J_g \leq J_{tg}$)

$$h_D = \left[h_I\left(\frac{J_{tg}}{J_g}\right)^{0.8} - h_{ST}\right]\left(\frac{J_g}{J_{tg}}\right) + h_{ST} \tag{11b}$$

$$h_{ST} = 0.725\left[1 + 0.741\left(\frac{1-x}{x}\right)^{0.3321}\right]^{-1}\left[\frac{k_l^3 \rho_l(\rho_l - \rho_g) g h_{fg}}{\mu_l D_i \Delta T}\right]^{0.25} + (1 - x^{0.087}) h_{lo} \tag{11c}$$

where

$$J_g = \frac{xG}{\left[g D_i(\rho_l - \rho_g)\right]^{0.5}} \tag{11d}$$

$$J_{tg} = \left[\frac{7.5}{(4.3 X_{tt}^{1.111} + 1)^{-3}} + C_T^{-3}\right]^{-\frac{1}{3}} \tag{11e}$$

$$h_{lo} = 0.023 Re_{lo}^{0.8} Pr_l^{0.4} \frac{k_l}{D_i} \tag{11f}$$

$$Re_{lo} = \frac{G D_i}{\mu_l} \tag{11g}$$

where Re_{lo} is Reynolds liquid only number. The value of C_T is 1.6 for hydrocarbon and 2.6 for other refrigerants.

1.3.2.3 Chato (1962) Correlation

Chato (1962) developed a detailed analytical model of the heat transfer for gravity-driven film condensation. In Chato's (1962) analysis the heat transfer through the liquid pool at the bottom of the tube is considered negligible compared to the conduction across the thin film on the upper portion of the tube wall.

$$h = 0.728 K_c\left[\frac{g\rho_l(\rho_l - \rho_g) k_l^3 h'_{fg}}{\mu_l (T_{r,s} - T_w) D_i}\right]^{\frac{1}{4}} \tag{12a}$$

$$h'_{fg} = h_{fg}\left[1 + 0.68 \frac{c_{pl}(T_{r,s} - T_w)}{h_{fg}}\right] \tag{12b}$$

where K_c is a constant decided for each refrigerant. Chato assumed for K_c in Eq. (12a) the value of 0.76.

1.3.2.4 Dobson and Chato (1998) Correlation

Dobson and Chato (1998) developed a correlation using a two-phase multiplier for an annular flow regime. They have also provided a correlation for a wavy flow regime. Their correlations are commonly used in the literature for zeotropic refrigerants and suited for flow in both horizontal and vertical tubes. It is recommended for $G > 500$ kg/m^{2}s for all qualities in horizontal tubes.

$$h = h_t + \left(1 - \frac{\theta}{\pi}\right) h_b \tag{13a}$$

where θ is geometrically related to the void fraction by the following formula if the area occupied by the condensate film was neglected and calculated with

$$1-\frac{\theta}{\pi}\approx\frac{\arccos(2\alpha-1)}{\pi} \tag{13b}$$

where the void fraction α can be calculated with the following Jaster and Kosky (1979) equation:

$$K_c=\alpha^{\frac{3}{4}} \tag{13c}$$

$$K_c=\frac{0.23Re_{go}^{0.12}}{0.728(1+1.11X_{tt}^{0.58})} \tag{13d}$$

$$Re_{go}=\frac{GD_i}{\mu_g} \tag{13e}$$

where Re_{lo} is the vapor only Reynolds number, and X_{tt} is the Lockhart-Martinelli parameter evaluated with Eq. (8c). The HTC for the top portion of the tube is evaluated with Eq. (12a) by replacing the K_c value given by (13d), and the HTC for the bottom is expressed as

$$h_b=0.0195\frac{k_l}{D_i}Re_l^{0.8}Pr_l^{0.4}\phi_l \tag{13f}$$

$$\phi_l=\sqrt{1.376+\frac{c_{p1}}{X_{tt}^{c_{p2}}}} \tag{13g}$$

for $0<Fr_l\leq 0.7$

$$c_{p1}=4.172+5.48Fr_l-1.564Fr_l^2 \tag{13h}$$

$$c_{p2}=1.773-0.169Fr_l \tag{13i}$$

and for $Fr_l>0.7$, $c_{p1}=7.242$, $c_{p2}=1.655$

where $$Fr_l=\frac{(G/\rho_l)^2}{gD_i} \tag{13j}$$

1.3.2.5 Singh et al. (1996) Correlation

Singh et al. (1996) proposed the following FCHTC for stratified wavy flow:

$$h=\theta h_b+(2\pi-\theta)h_t \tag{14a}$$

where θ is the liquid level angle in radians, subtended from the top of the tube to the liquid level and is approximated by

$$\theta\approx 2\cos^{-1}(2\alpha-1) \tag{14b}$$

The HTC of the bottom film was evaluated with

$$h_b=h_l\left(1+\frac{0.2332}{X_{tt}^{1.402}}\right) \tag{14c}$$

where h_l is the HTC for the liquid portion of the flow evaluated by Eq. (8b), h_l is determined by Eq. (12a) with a K_c value of 0.1271, and X_{tt} is calculated with Eq. (8c).

2. AVAILABLE EXPERIMENTAL DATA FOR FILM CONDENSATION HEAT TRANSFER

The 1214 experimental data of FCHT are collected from 10 papers and showed in Table 1. All data are presented graphically in the source papers.

Table 1
Experimental Data Sources of Heat Transfer for Film Condensation

Reference	Refrigerant	Parameter range: T_{sat}(℃)/P_{sat}(bar)/G (kg/m^{2}s)/q (kW/m^2)/x	Geometry range: D_i(mm)/D_o(mm)/L(mm)/ /Orientation/ Surface type	Number of data points
Laohalertdecha and Wongwises (2010)	R134a	40–50/*/200–700/ 5–10/*	8.7/9.52/2000/horizontal flow/ in smooth round tube	13
Aprea et al. (2003)	R22/ R407C	36.6–39.6/14.3–15.2/ 45.5–120/*/*	20/22/6600/horizontal flow/in smooth round tube	121
Park et al. (2008)	R22/C_3H_6/C_3H_8/ DME/C_4H_{10}	40/5.31–16.52/100–300/7.3–7.7/*	8.8/*/530/horizontal flow/in smooth round tube	126
Wen et al. (2006)	R-290/ R-600	40/*/205–510/5.2/ 0.15–0.84	2.46/3.18/3850/horizontal flow/in smooth round tube	64
Cavallini et al. (2001)	R134a/R125/R32/ R410A/R236ea	30–50/2.46–31.5/100–750/*/0.15–0.85	8/12/1000/horizontal flow/ in smooth round tube	230
Park and Hrnjak (2009)	CO_2	(–25)–(–15)/*/200–800/ */0.1–0.9	0.89/*/150/horizontal flow/ in smooth round tube	113
Park et al. (2011)	R1234ze(E)/ R134a/ R236fa	25–70/*/50–260/1–62/ 0.0–1.0	1.3-1.45[a]/*/260/vertical downward /in multi-port rectangular channel	206
Dalkilic et al. (2009)	R134a	40–50/*/260–515/ 11.3–55.3/*	8.1/9.52/500/vertical downward/in smooth round tube	266
Jung et al . (1999)	R11/R123/R12/ R134a	39/*/*/*/*	*/15.9/*/horizontal flow /outside of smooth or low fin tube	48
Belghazi et al. (2001)	R134a	40/*/*/5–30/*	14.2/16.8/300/ horizontal flow/outside of smooth round tube	27

[a] Hydraulic diameter.
* Data out of the applicable conditions of the correlation.

3. COMPARATIVE STUDY OF THE EXISTING CORRELATIONS AGAINST THE EXPERIMENTAL DATA

Table 2
Comparison Between Experimental Data and Correlation Predictions for Horizontal Flow Outside Smooth Tube

Data sources	Errors %	Correlations: Beatty-Katz	Dhir-Lienhard	Nusselt 1
Jung et al .(1999)	MRD	-12.7	-5.2	-5.7
	MARD	12.7	5.5	5.9
Belghazi et al. (2001)	MRD	—	-8	-8.5
	MARD	—	8	8.5
Average	MRD	-12.7	-6.6	-7.1
	MARD	12.7	6.8	7.2

"—" means that the correlation is unavailable, the same as in Tables 3 and 4.

The 1214 experimental data as indicated in Table 1 are used for the comparative study of the 14 HTCs as described above, and the results are listed in Table 2 for horizontal flow outside smooth tubes, Table 3 for flow on vertical surfaces of plates or tubes and Table 4 for horizontal flow inside smooth tubes, respectively, where the MRD is the mean relative deviation and the MARD is the mean absolute relative deviation.

$$MRD = \frac{1}{N}\sum_{i=1}^{N}\frac{y(i)_{cal} - y(i)_{exp}}{y(i)_{exp}} \tag{21}$$

$$MARD = \frac{1}{N}\sum_{i=1}^{N}\left|\frac{y(i)_{cal} - y(i)_{exp}}{y(i)_{exp}}\right| \tag{22}$$

where y_{cal} is the calculated value, y_{exp} is the experimental value, and N is the number of the data points.

The Incropera and DeWitt (2002) correlation is not included because there are no available data in the literature listed in Table 1.

Table 3
Comparison Between Experimental Data and Correlation Predictions for Flow on Vertical Surfaces of Plate or Tube

Data sources	Errors %	Correlations: Jung et al.	Longo-Gasparella	McAdams	Nusselt 2	Shah
Dalkilic et al. (2009)	MRD	-11.1	-76	-83.3	-80.7	-46.6
	MARD	23.7	76	83.3	80.7	46.6
Park et al. (2011)	MRD	—	-15.6	-31.2	-31.9	—
	MARD	—	24.7	37.4	34	—
Average	MRD	-11.1	-45.8	-57.3	-56.3	-46.6
	MARD	23.7	50.4	60.4	57.4	46.6

Table 4
Comparison Between Experimental Data and Correlation Predictions for Horizontal Flow Inside Smooth Tube

Data sources	Errors %	Correlations: Akers et al.	Cavallini et al.	Chato	Dobson-Chato	Jung et al.	Shah	Singh et al.
Aprea et al. (2003)	MRD	-38.5	-38.1	18.5	50.7	30.6	-43.5	24.1
	MARD	38.6	38.1	35.5	50.7	33	43.5	38.4
Cavallini et al. (2001)	MRD	-16.8	-15.3	-43.3	3.6	-6.4	-20.1	-40.6
	MARD	22.9	17.2	44.2	13.7	18.9	21.6	42.2
Laohalertdecha and Wongwises (2010)	MRD	-1.1	4.9	-35.3	34.4	-33.1	-19.7	-32.2
	MARD	14.8	13.1	35.3	36.9	33.1	21.8	32.2
Park et al. (2008)	MRD	3.6	0.75	-6	66.4	7.6	-7.04	-1.6
	MARD	24.2	15.2	24.3	66.4	12.9	15.6	24.5
Park and Hrnjak (2009)	MRD	22.7	26.3	23.7	71.8	36.1	16.0	29.3
	MARD	30.7	31.2	33.0	71.8	37.7	25.2	36.5
Wen et al. (2006)	MRD	-21.1	-18.2	-28	12.4	-45.8	-23	-24.7
	MARD	21.1	18.2	37	16.1	45.8	23	35.9
Average	MRD	-8.5	-6.6	-11.7	39.9	-1.8	-16.2	-7.6
	MARD	25.4	22.2	34.9	42.6	30.2	25.1	35

From the above tables, the following can be seen:

(1) For horizontal flow outside smooth tubes, the Dhir and Lienhard (1971) correlation has the best predictability of experimental data, with an MRD of –6.6% and an MARD of 6.8%, and the Nusselt (1916) 1 correlation is the next best one, with the MARD of 7.2%.

(2) For flow on vertical surfaces of plates or tubes, the Jung et al. (2003) correlation has the best predictability of experimental data, with an MRD of –11.1% and an MARD of 23.7%. However, it application is very limited.

(3) For horizontal flow inside smooth tubes, the Cavallini et al. (2006) correlation has the best predictability of experimental data, with an MRD of –6.6% and an MARD of 22.2%. The correlations of Shah (1979) and Akers et al. (1959) are the next two best ones, with the MARD of 25.1% and 25.4%, respectively.

CONCLUSIONS

(1) The 1214 data points of film condensation heat transfer (FCHT) are obtained from 10 published papers and 14 FCHTCs are reviewed, including 4 for horizontal flow outside smooth tubes, 3 for flow on vertical surfaces of plates or tubes, 2 for flow inside smooth tubes either vertically or horizontally, and 5 for horizontal flow inside smooth tubes.

(2) All the 14 FCHTCs except one are compared with the experimental data. The comparisons show that the Dhir and Lienhard (1971) correlation and the Nusselt (1916) 1 correlation have the is the best predictability of experimental data for horizontal flow outside smooth tubes, that the Jung et al. (2003) correlation has the best predictability of experimental data for flow on vertical surfaces of plates or tubes with limited applications, and that the correlations of Cavallini et al. (2006), Shah (1979), and Akers et al. (1959) are best for horizontal flow inside smooth tubes.

(3) For horizontal flow inside smooth tubes, Cavallini et al. (2006) has highest accuracy with the best one, with an MRD of –6.6% and an MARD of 22.2%. More efforts need to be made to develop better correlations.

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