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Radia Eldeeb
University of Maryland, College Park, United States of America, radiame@umd.edu

Vikrant Aute
University of Maryland, College Park, United States of America, vikrant@umd.edu

Reinhard Radermacher
University of Maryland, College Park, United States of America, raderm@umd.edu

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Radia ELDEEB[1], Vikrant AUTE[2]*, Reinhard RADERMACHER[3]

Center for Environmental Energy Engineering
Department of Mechanical Engineering, University of Maryland
College Park, MD 20742, USA
[1]Tel: 301-405-7314, [2]Tel: 301-405-8726, [3]Tel: 301-405-5286
[1]Email: radiame@umd.edu, [3]Email: raderm@umd.edu

* Corresponding Author Email: vikrant@umd.edu

ABSTRACT

Plate heat exchangers are used in a wide variety of applications from air-conditioning, refrigeration, food processing, and chemical industry to energy generation systems. Plate heat exchangers are favored because of their compactness, flexible thermal sizing, close approach temperature, pure counter-flow operation, and enhanced heat transfer performance. This paper presents a literature review of available correlations for heat transfer and pressure drop calculations during condensation in brazed plates. Condensation heat transfer in plate heat exchangers can be preliminarily evaluated using the classic Nusselt equation for laminar film condensation on a vertical plate. However, condensation performance is influenced by many factors such as fluid properties, plate geometry, and mass flow rate, and therefore, it is difficult to obtain an ideal correlation which accounts for all these factors. The heat transfer coefficients of different refrigerants, including alternative lower GWP refrigerants R32, D2Y60, and L41a, are computed using heat transfer correlations found in literature. Generally, R32 shows the most favorable heat transfer performance, followed by L41a. An overview on an already existing model that is capable of predicting the performance of plate heat exchangers with generalized multi-fluid and multi-pass configurations is also presented. The model is validated against experimental data for water-to-R134a condenser. A total of sixteen experimental datasets are used. The heat capacity predicted by the model is within ±5% of the measured heat capacity, while most of the predicted outlet temperatures are within ±2 K of measured values.

1. INTRODUCTION

Plate heat exchangers (PHEs) are used in a wide variety of applications including but not limited to air-conditioning, refrigeration, food processing, chemical industry, marine, extraction and refining processes, and energy generation systems. PHEs are characterized by high effectiveness of heat transfer which is highly required in the competitive heat exchanger industry. They are also characterized by compactness, flexible thermal design where the number of plates can be easily varied according to the thermal demand, and good temperature control (Wang *et al.*, 2007).

Brazed plate heat exchanger (BPHE) is a type of PHE which consists of a pack of metal plates and two end plates which are brazed where two or more fluids flow in between the plates and exchange thermal energy. It is highly compact and can be used for high temperature and high pressure applications including water-cooled evaporators and condensers refrigeration applications as well as process water heating and heat recovery in various applications (Shah & Sekulic, 2003). The maximum operating temperature and pressure of BPHE are 225 °C and 30 bar, respectively (Wang *et al.*, 2007).

BPHEs are used with heat pumps, industrial chillers, drinking water processing, heat recovery systems, oil cooling, indoor heating including floor heating, solar thermal systems, cooling machines and motors, economizers, and other applications. One of the applications of brazed plate condensers is the rejection or recovery of heat to water in refrigeration in residential, industrial, and automotive applications. Brazed plate condensers are characterized by high thermal efficiency, compact size, light weight, durability, high working pressures and temperatures, and high corrosion resistance.

2. CONDENSATION IN BRAZED PLATE HEAT EXCHANGERS

Since it is difficult to maintain drop-wise condensation on metal surfaces, plate condenser heat transfer is dominated by filmwise condensation. Condensation on vertical plates is either gravity-controlled or shear controlled. A theoretical study of laminar condensation over cooled metal surfaces has been performed by the pioneering work of Nusselt (1916). Nusselt assumed that the temperature gradient caused by steam condensing on a cooled metal surface is linear and exists only in the film formed on the surface, while the vapor temperature is constant at the saturation temperature. This assumption might be valid due to the thinness of the film formed and the thin space between the BPHE plates. Nusselt derived a correlation for average heat transfer coefficient for gravity-controlled laminar flow condensation for a vertical surface as

$$h=\left[\frac{\rho_{liq}{}^{2}g\gamma k_{liq}^{3}}{4\mu_{liq}L\left(T_{sat}-T_{w}\right)}\right]^{1/4} \tag{1}$$

Later in 1952, Bromley improved Nusselt's correlation by including the effect of heat capacity on the heat transfer coefficient calculation indicating that the effect of a finite heat capacity of the condensate film increases the heat transfer coefficient through thinning the film due to the presence of a flatter temperature gradient in the film (Bromley, 1952). Rohsenow (1956) further improved Bromley's correlation by rather accounting for non-linear temperature distribution within the condensate film. However, Rohsenow concluded from the analysis that if the value of $c\left(T_{sat}-T_{w}\right)/\gamma$, where c is the liquid specific heat, is small, which is the case for a great number of applications, then Nusselt's assumption of linear temperature distribution is valid. Chen (1961) added to the previous studies the effect of shear stress by including a non-zero negative velocity gradient at the liquid-vapor interface which is only important for fluids with low Prandtl numbers.

The aforementioned correlations are only good for gravity-controlled laminar film condensation with no waves, and cannot be applied in wavy and turbulent condensation regions. Since condensation performance is influenced by fluid property, mass flow rate, plate geometry, and other factors, it is difficult to obtain a general correlation that predicts all situations. Thus, theoretical performance evaluation is very challenging and more experimental effort is required. Moreover, specific correlations might be developed for specific situations for common working fluids used in condensation applications such as water, and common refrigerants.

Yan *et al.* (1999) investigated heat transfer and pressure drop of the condensation of R134a in a vertical BPHE experimentally for a chevron angle of 60˚. They showed that even at lower mass flux of R134a of 100 kg/m^2·s, the condensation heat transfer coefficient for PHEs is about 25% higher than that for circular pipe with a mass flux of 130 kg/m^2·s obtained in similar measuring conditions by Eckels and Pate (1991). The correlations they developed take into account the effect of mass flux, imposed heat flux, the vapor quality, and the condensation pressure. The experimental results showed that the condensation heat transfer coefficient slightly increases with mass flux, increases sharply with vapor quality for lower mass flux, slightly increases with average imposed mass flux, and almost not affected by condensation pressure. On the other hand, the frictional pressure drop increases more significantly with mass flux, increases with average imposed mass flux, and decreases with condensation pressure. The condensation heat transfer correlation given by Equation (8) agreed with experimental values within 15%, while the friction factor given by Equation (9) agreed within 13.3%. A similar experimental setup, with similar plate type, was used by Kuo *et al.* (2005) to investigate heat transfer and pressure drop of the condensation of R410A. Their experimental results show that the heat transfer coefficient of R410A increases linearly with the mean vapor quality, increases with the mass flux and the imposed heat flux, but is unaffected by the saturation pressure. The friction factor is slightly affected by the heat flux and saturation pressure, while it significantly increase with mass flux.

Han *et al.* (2003) conducted experiments on R410A and R22 to investigate the effect of chevron angle on the heat transfer and pressure drop in a BPHE. Chevron angles of 45˚, 35˚, and 20˚ were used. They varied the mass flux, the condensation temperature and the vapor quality. Unlike the previous correlations, they included the effect of the plate geometry in the heat transfer and pressure drop calculations. They concluded that both the heat transfer coefficient and the pressure drop increase with mass flux and vapor quality, while they decrease with saturation temperature and chevron angle. They developed a correlation for heat transfer that agrees within 20% of their experimental results. Longo (2008) performed experimental tests on R134a condensation inside a small BPHE with herringbone plates with a corrugation angle of 65˚. Longo investigated the effects of mass flux, saturation temperature, and superheating on condensation performance. As previously concluded (Yan *et al.*, 1999, Kuo *et al.*, 2005), the heat transfer coefficient showed weak sensitivity to the change in the condensation temperature. Longo indicated that at refrigerant mass flux smaller than 20 kg/m^2·s, Nusselt laminar film condensation correlation can be applied (Nusselt, 1916) as condensation is mainly gravity-controlled and the heat transfer coefficient is unaffected by an increase in the mass flux. However, for higher mass flux, forced convection condensation takes place and the heat transfer coefficient increases by 30% by doubling the mass flux (Longo, 2008). Longo suggests that the transition from gravity-controlled condensation to forced convection condensation takes place at a liquid Reynolds number from 200 to 300 which is the transition from laminar to turbulent flow regime in PHEs with high corrugation angle (Shah & Focke, 1988). The forced convection condensation heat transfer coefficient obtained agreed within 20% with the equation proposed by Akers *et al.* (1959) for forced convection condensation inside tubes. Palmer *et al.* (2000) measured the average Nusselt numbers during evaporation and condensation of R22, R290, R290/600a, and R32/152a in the presence of lubricant oils inside a BPHE at low mass fluxes. Based on the measured data, two correlations were developed to account for the presence of different mineral oils. The first correlation correlated 95% of the data within 25%, while the second correlation correlated 80% of the data within 25%. However, the correlations were developed using typical system operating conditions allowing the correlations to be used for actual system design.

Jokar *et al.* (2006) investigated the condensation performance of R134a inside a BPHE that is used as an automotive refrigeration system and applied dimensional analysis to develop new heat transfer correlation. However, the correlation only agreed within 25% of the authors' experimental data. They also introduced a friction factor correlation which did not match their experimental results very well with only 57% of the condensation pressure drop data within 50%. Mancin et al. (2012) studied partial condensation of R410A and R407C inside two BPHE geometries with different aspect ratios and number of channels. Their experimental results showed that the heat transfer coefficient increases with vapor quality and decreases with the wall to saturation temperature difference. They also concluded that the heat transfer coefficient increases with mass flux at a given outlet vapor quality only at higher values of mass flux (≈ 40 kg/m^2·s), while it is unaffected at lower values as previously concluded by Longo (2008). Shah (1979) developed a correlation from a wide range of experimental data which included water, refrigerants, and organics condensing in horizontal, vertical, and inclined pipes. This equation is widely accepted in engineering calculations as many of the experimental data were obtained from the literature (Wang *et al.*, 2007). Wang *et al.* (2000) studied the condensation of steam in different PHEs. They proposed a modified Boyko-Kruzhilin (1967) equation to predict the local heat transfer coefficient. Kumar (1992) reported some experimental results for the condensation of R22 and ammonia in different types of PHEs. By comparing the results, Kumar recommended the use of Nusselt (1916) equation for gravity-controlled low mass flux flows, and the Carpenter and Colburn equation (Carpenter & Colburn, 1951) for shear-controlled higher mass flux flows. A summary of condensation heat transfer and pressure drop correlations in literature is given in Table 1. However, none of this work has been compared to experimental data of other refrigerants to be introduced as a generalized condensation heat transfer correlation. More extensive work is required in order to obtain more generalized performance estimation methods for condensation in PHEs.

Table 1: Condensation heat transfer and pressure drop correlations

Investigator	Correlation		Comments
Kuo *et al.* (2005)	$\begin{cases} h = 0.2092\left(\dfrac{k_{liq}}{D_h}\right)\mathrm{Re}_{liq}^{0.78}\,\mathrm{Pr}_{liq}^{1/3}\left(\dfrac{\mu_m}{\mu_w}\right)^{0.14}\left(0.25Co^{-0.45}Fr_{liq}^{0.25}+75Bo^{0.75}\right) \\ Co=\left(\dfrac{\rho_g}{\rho_{liq}}\right)\cdot\left(\dfrac{1-x_m}{x_m}\right)^{0.8} \end{cases}$	(2)	R410A, Chevron plate, $\beta = 60°$ $50 < G < 150$
	$f = 21,500\,\mathrm{Re}_{eq}^{-1.14}\,Bo^{-0.085}$	(3)	

Investigator	Correlation		Comments
Han *et al.* (2003)	$\begin{cases} h = Ge_1\left(\frac{k_{liq}}{D_h}\right)\mathrm{Re}_{eq}^{Ge_2}\ \mathrm{Pr}^{1/3}, \\ \mathrm{Re}_{eq} = \frac{G_{eq}D_h}{\mu_{liq}}, G_{eq} = G\left[1 - x_m + x_m\left(\frac{\rho_{liq}}{\rho_g}\right)^{1/2}\right] \\ Ge_1 = 11.22\left(\frac{b}{D_h}\right)^{-2.83}\left(\frac{\pi}{2}-\beta\right)^{-4.5}, \\ Ge_2 = 0.35\left(\frac{b}{D_h}\right)^{0.23}\left(\frac{\pi}{2}-\beta\right)^{1.48} \end{cases}$	(4)	R410A, R22, Chevron plate, $\beta = 20°, 35°, 45°$ $20 < G < 35$
	$\begin{cases} f = Ge_3\ \mathrm{Re}_{eq}^{Ge_4}, \\ Ge_3 = 3521.1\left(\frac{b}{D_h}\right)^{4.17}\left(\frac{\pi}{2}-\beta\right)^{-7.75}, \\ Ge_4 = -1.024\left(\frac{b}{D_h}\right)^{0.0925}\left(\frac{\pi}{2}-\beta\right)^{-1.3} \end{cases}$	(5)	
Thonon and Bontemps (2002)	$\begin{cases} h = 1564 h_{lo}\ \mathrm{Re}_{eq}^{-0.76}, \text{ for pure hydrocarbons} \\ h_{lo} = 0.347\left(\frac{k_{liq}}{D_h}\right)\mathrm{Re}^{0.653}\ \mathrm{Pr}^{0.33}\ \text{(Thonon } et\ al.\text{1999)} \end{cases}$	(6)	Pure hydrocarbons, mixtures of hydrocarbons Chevron plate, $100 < Re < 2000$ $\beta = 45°$
Wang *et al.* (2000)	$\begin{cases} h = h_{liq}\left(\frac{\rho_{liq}}{\rho_m}\right)^{\left(a + b\cdot\mathrm{Re}_{liq}^{c}\right)} \\ h_{liq} = 0.023\left(\frac{k_{liq}}{D_h}\right)\mathrm{Re}_{liq}^{0.8}\ \mathrm{Pr}_{liq}^{0.4} \end{cases}$	(7)	Water, Chevron plate, $\beta = 30°, 45°, 60°$ $300 < Re < 2000$ $20 < G < 120$
Yan *et al.* (1999)	$h = 4.118\left(\frac{k_{liq}}{D_h}\right)\mathrm{Re}_{eq}^{0.4}\ \mathrm{Pr}_{liq}^{1/3}$	(8)	R134a, Chevron plate, $\beta = 60°$ $500 < Re < 1000$ $60 < G < 120$
	$\begin{cases} f = 94.75\left(\frac{p_m}{P_c}\right)^{0.8} Bo^{0.5}\ \mathrm{Re}^{-0.4}\ \mathrm{Re}_{eq}^{-0.0467} \\ Bo = \frac{q_w''}{G\gamma} \end{cases}$	(9)	
Shah (1979)	$h = h_{liq}\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{p^{0.38}}\right]$	(10)	Wide variety of refrigerants in horizontal, inclined, and vertical pipes
Akers *et al.* (1959)	$h = 5.03\left(\frac{k_{liq}}{D_h}\right)\mathrm{Re}_{eq}^{1/3}\ \mathrm{Pr}_{liq}^{1/3}$	(11)	Used by Longo (2008) for R134a, Herringbone plate, $65°$, $20 < G < 240$

3. CONDENSATION HEAT TRANSFER CORRELATIONS COMPARISON

In order to investigate the heat performance of different refrigerants using some of the correlations given in Table 1, the heat transfer coefficient using the different correlations is computed for R410A, and lower GWP refrigerants R32, D2Y60, and L41a with varying vapor quality. The results are shown in Figure 1. All calculations are done for a saturation temperature of 45°C, for a hydraulic diameter of 6.42 mm, a chevron angle of 60°, a corrugation pitch of 10 mm, heat flux of 15 kW/m^2, and a mass flux of 60 kg/s·m^2.

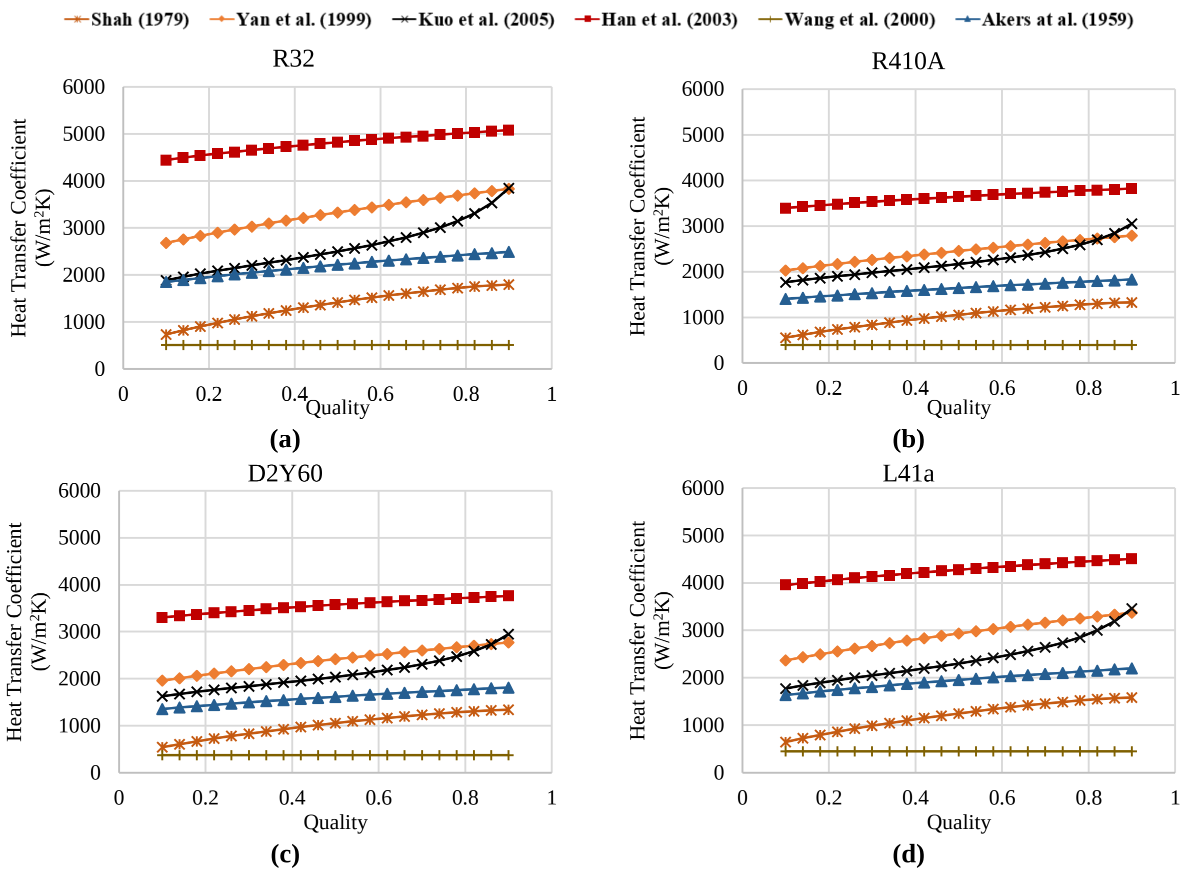

Figure 1: Heat transfer coefficient at different vapor quality calculated using different correlations for (a) R32, (b) R410A, (c) D2Y60, and (d) L41a.

Generally, the heat transfer coefficient increases as the vapor quality increases, with Kuo *et al.* (2005) showing a significant increase at higher values of vapor quality. Han *et al.* (2003) gives the highest predicted values while Wang *et al.* (2000) gives the lowest predicted values for the heat transfer coefficients. This might be because Han *et al.* (2003) developed their correlation based on much lower values of mass flux, and thus, their correlation might be only applicable for mass fluxes in the range of 20-35 kg/s·m^2. On the other hand, Wang *et al.* (2000) developed their correlation using steam condensation, which possesses a large latent heat value, and no other refrigerant was used in their investigation although different geometries and chevron angles were used in their study. They also did not investigate the effect of the vapor quality on the heat transfer performance and their correlation varies with Reynolds number. Thus, their correlation might be limited only to steam condensation with varying Reynolds number. Wang *et al.* (2000) also used much lower mass flux values in their experiments than the mass flux used in this calculation.

Shah (1979) predicts the heat transfer coefficient fairly, although it is not developed for PHEs. However, it has a wide acceptability in the industry (Wang *et al.*, 2007). The heat transfer coefficients predicted by Shah (1979) are close to that predicted by Akers *et al.* (1959), which is suggested by Longo (2008), at high vapor qualities. Akers *et al.* (1959) coincides with the predicted values of Kuo *et al*. (2005) for R32 and L41a at low vapor quality. Yan *et al.* (1999) slightly over predicts the heat transfer coefficients compared to other correlations except for that of Han *et al.* (2003).

R32 has the highest predicted heat transfer coefficient, using any of the correlations at any given quality. This is because R32 has the highest latent heat, smallest molecule size, and the highest liquid thermal conductivity of all the presented refrigerants. L41a has higher heat transfer coefficient values than R410A and D2Y60, but lower than that of R32. This is also because L41a has the highest latent heat and liquid thermal conductivity after R32. However, L41a and D2Y60, as other refrigerant mixtures, have the drawback of diffusion-controlled two-phase flow which might affect their thermal performance negatively. D2Y60 also has a temperature glide of about 5.2 °C at this condensing temperature, while L41a has a temperature glide of about 1.7 °C, and thus D2Y60 is more affected by diffusion-controlled condensation. R410A and D2Y60 have similar latent heat and liquid thermal conductivity. Thus, R410A and D2Y60 show similar heat transfer performance with R410A having slightly higher values than D2Y60. However, other considerations must be taken into account for the mixtures of D2Y60 and L41a such as the effect of the diffusion-controlled two-phase process as well as the effect of enhanced mixing inside BPHE. Generally, further studies are required to accurately predict the heat performance of such refrigerants and refrigerant mixtures in BPHEs.

4. BRAZED PLATE HEAT EXCHANGER MODEL

The BPHE model used in this analyses is based on the work by Qiao *et al.* (2013). The model is based on a finite volume approach that divides the entire plate heat exchanger into multiple slices. Each slice spans multiple channels and the performance is evaluated using wall temperature linked equations. At the heat exchanger level, all the slices are iterated upon using a successive substitution approach. The model accounts for and calculates varying channel wall temperature and allows for complex geometry and multiple flow passes. The model can handle more than two different fluid streams undergoing simultaneous phase change. The model was validated with single phase water-water and two-phase ammonia and R22 boiling experiments (Qiao *et al.*, 2013). However, the model was not validated with condensation experiments which will be presented in Section 5. The model is further developed in the current work to include pressure drop during condensation. The model divides the entire PHE of N channels into M slices of equal size. The control volume upon which the governing equations are applied is called a segment, thus the BPHE has $N \times M$ segments. The schematic of the segment control volume is shown in Figure 2. For condensation, the total pressure drop is given by

$$\Delta P_{cond} = \Delta P_f + \Delta P_g + \Delta P_{acc} + \Sigma \Delta P_{ports} + \Sigma \Delta P_{man} \tag{12}$$

Where ΔP_f is the frictional pressure drop in the segment, ΔP_g is the gravitational pressure drop, ΔP_{acc} is the flow acceleration pressure drop, and $\Delta P_{ports}, \Delta P_{man}$ are pressure drops in ports and manifolds, respectively. The estimation of ΔP_f for two-phase flow is generally complicated and there isn't a unified calculation method in literature. A commonly used method is based on the Lockhart-Martinelli model (Lockhart & Martinelli, 1949) for water flow in horizontal tubes. However, more studies on frictional pressure drop for two-phase flow in PHEs are required. The gravitational pressure drop for two-phase flow is given by

$$\Delta P_g = \pm \int_0^L \left[\alpha \rho_v + (1-\alpha) \rho_{liq} \right] g dz \tag{13}$$

Where the term is positive for vertical up flow and negative for down flow. The two-phase void fraction, α, defined as the fraction of the channel flow cross-sectional area that is occupied by the vapor phase is estimated by (Zivi, 1964)

$$\alpha = \left[1 + \frac{1-x}{x} \left(\rho_v / \rho_{liq} \right)^{2/3} \right]^{-1} \tag{14}$$

Where x is the vapor quality. ΔP_{acc} for two-phase flow can be estimated from Lockhart & Martinelli (1949)

$$\Delta P_{acc} = G^2 \left\{ \left[\frac{(1-x_e)^2}{\rho_{liq} (1-\alpha_e)} + \frac{x_e^2}{\alpha_e^2 \rho_v} \right] - \left[\frac{(1-x_i)^2}{\rho_{liq} (1-\alpha_i)} + \frac{x_i^2}{\alpha_i^2 \rho_v} \right] \right\} \tag{15}$$

Where G is the mass flux of the fluid. Finally, the pressure drop in the inlet and outlet manifolds due to the expansion and contraction of the fluid is calculated using the empirical correlation (Shah & Sekulic, 2003)

$$\Sigma \Delta P_{man} = 1.5 \times \frac{G^2 N}{2\rho_i} \tag{16}$$

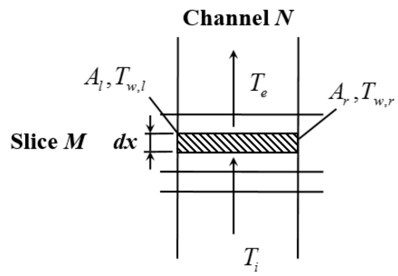

Figure 2: Schematic of segment control volume.

All the thermodynamic properties are calculated based on NIST REFPROP 9.1 (Lemmon *et al.*, 2013). Since the phase change of the fluid only occurs at the boundary of a segment, the number of slices must be chosen carefully such that the location of the phase change is predicted accurately, and thus an accurate calculation of the heat capacity is achieved. The effect of the number of segments in two-phase flow on the heat capacity calculation is shown in Figure 3. As shown, for two-phase computation, as the number of segments increases, the calculated heat capacity is more accurate but the computational time increases as well.

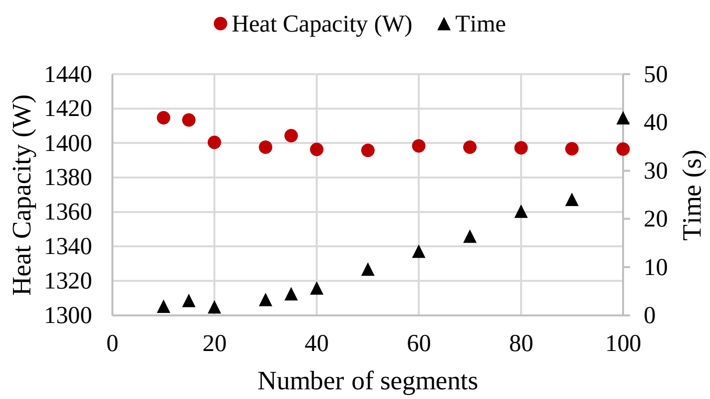

Figure 3: Effect of number of segments in two-phase flow on heat capacity calculation and computational time.

5. VALIDATION

4.1. System Description

A brazed plate heat exchanger used to condense R134a using water is used in a hybrid absorption vapor compression cycle in an experimental setup used to evaluate the performance of a desorber in such a system (Mandel 2013). The plate heat exchanger has 6 thermal plain plates made of SS316L, with 4 water channels and 3 R134a channels flowing in a counter flow pass arrangement. Since the experiments focus on the desorber, the pressure is only measured for R134a at the inlet to the condenser, and hence pressure drop information are not available. Also, the refrigerant contained a percentage of oil within 5%. A total of 16 tests are used in the validation.

4.2. Results

Figure 4 shows the predicted outlet temperature of water and R134a as compared to the experiments. The outlet temperature of water varies within ±1 K, while most of the outlet temperature of R134a varies within ±2 K. The deviation of few R134a outlet temperatures from the experimental data is probably due to the presence of oil in the refrigerant.

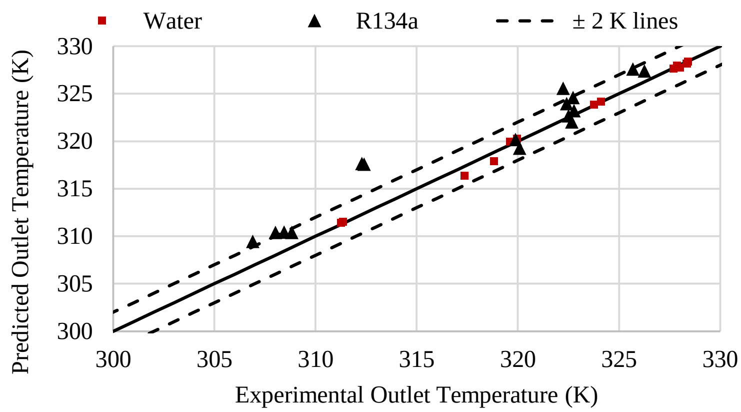

Figure 4: Comparison between simulation and experimental outlet temperature.

Figure 5 shows the predicted heat capacity as compared to the experimental heat capacity of the condenser. All the predicted heat capacity values vary within 5% of the experimental heat capacity showing very good agreement. This shows the ability of the model to predict the condensation performance accurately. However, the variation might be caused due to the presence of oil, and experiment uncertainty.

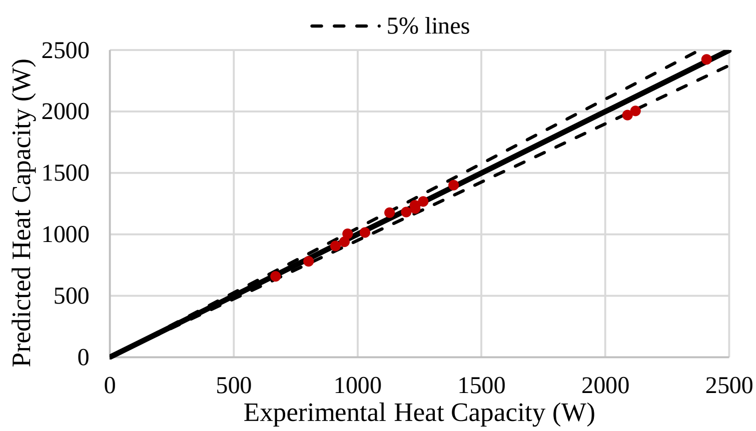

Figure 5: Comparison between predicted and experimental heat capacity.

6. CONCLUSIONS

A literature review of available correlations for heat transfer and pressure drop calculations during condensation in BPHEs is presented. Correlations predicting the condensation heat performance in PHEs are limited, while correlations for refrigerant mixtures are scarce. The heat transfer coefficients of R410A, R32, D2Y60, and L41a are computed using seven different heat transfer correlations from literature. Generally, R32 shows the most favorable heat transfer performance, followed by L41a. More general correlations for predicting heat transfer performance of pure refrigerant and refrigerant mixtures in BPHE are required. A previously introduced model is validated against experimental data for water-to-R134a plate condenser. A total of sixteen experimental datasets are used. The heat capacity predicted by the model is within ±5% of the measured heat capacity, while most of the predicted outlet temperatures are within ±2 K of measured values.

NOMENCLATURE

A	heat transfer area, m^2	L	length, m
b	channel spacing, m	M	number of slices
Bo	Boiling number	N	number of channels
c	specific heat, J/kg·K	p	pressure, Pa
Co	Convection number	Pr	Prandtl number
D_h	hydraulic diameter, m	q"	heat flux, W/m^2
f	friction factor	Q	heat capacity, W
Fr	Froude number	Re	Reynolds number
g	gravitational acceleration, m/s^2	T	temperature, K
G	mass flux, $kg/(m^2 \cdot s)$	x	vapor quality
h	convective heat transfer coefficient, $W/(m^2 \cdot K)$	ΔP	pressure drop, Pa
k	thermal conductivity, W/(m·K)		

Greek Symbols

α	void fraction
β	chevron angle, °
γ	latent heat of vaporization, J/kg
μ	dynamic viscosity, Pa·s
ρ	density, kg/m^3

Subscripts

acc	acceleration	l	left
c	critical	liq	liquid
cond	condensation	m	mean
e	exit	man	manifold
eq	equivalent	ports	ports
f	friction	r	right
g	vapor	sat	saturation
i	inlet	w	wall

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