CONDENSATION IN HORIZONTAL SMOOTH ROUND TUBE

BY

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THESIS

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ABSTRACT

From thermodynamic point of view, first droplet of liquid during condensation appears at quality one and the last vapor disappears at quality zero. This is because thermodynamic assumes equilibrium and temperature to be uniform. Therefore, conventional modeling of condensation typically contains three regions: superheated (SH), two-phase (TP) and subcooled (SC) regions. However, since phase change requires temperature gradient, the first droplet in heat exchanger occurs as soon as the temperature of inner wall drops below saturation temperature even though the bulk enthalpy indicates the flow still in superheated region. Once the liquid appears, the heat transfer and pressure drop mechanism deviates from single phase analysis and switches into two-phase behaviors. According to this characterization, a fourth and fifth zone were proposed to be modeled in heat exchangers, classified as condensing superheated region (CSH) and condensing subcooled region (CSC). Previous experimental data has indicated the existence of these additional regions. Preliminary visualization directly confirmed the presence of liquid-film layer in SCH region. A correlation was made to calculate the heat transfer coefficient (HTC) in CSH region. However, flow characteristics in CSH region are almost never discussed, and little of the physics behind heat transfer mechanism in this region was explained. Therefore, the early-stage condensation flow regimes and void fractions predicted from existing flow maps and correlations are either missing or physically incorrect. As for heat transfer correlation in CSH region, despite it being accurate, the approach is highly empirical, and thus hard to be understood and generalized.
This study aims at better understanding the physics from flow characterization and heat transfer measurement, which can be later used for the development of a more physical model for void fraction, heat transfer and pressure drop. The paper presents flow visualization, liquid film thickness measurement and heat transfer results, showing the effect of mass flux and heat flux on the onset of condensation, film distribution, flow regime and HTC. The result of flow visualization reveals that the onset of condensation is always in SH region, as annular flow and the flow regime is strongly affected by mass flow but heat flux. Based on the flow visualization in CSH region, a new diabetic flow map is proposed to best represent the flow characters and predict flow regimes in CSH region. Film thickness measurement in CSH demonstrates that void fraction drops below one at the onset of condensation, and mass flux has greater effect on slip ratio than heat flux. Besides, having film distribution suggests an opportunity to more objectively determine the flow regime and develop a more physical model knowing more details about the two-phase structure inside the tube. Experimental HTC data shows that mass flux can affect HTC in TP region but not in CSH region, while heat flux can affect HTC in CSH region but not TP region. The counter intuitive results in CSH region gives rise to a new heat transfer coefficient based on the local heat transfer across liquid film, which is named as film heat transfer coefficient ($HTC_f$). The comparison between HTC and $HTC_f$ illustrates the difference between two interpretations and shows that $HTC_f$ is a better representative of heat transfer process in CSH region.
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<td>h</td>
<td>Specific enthalpy</td>
<td>(J kg(^{-1}))</td>
</tr>
<tr>
<td>HTC</td>
<td>Heat transfer coefficient</td>
<td>(W m(^{-2}) K(^{-1}))</td>
</tr>
<tr>
<td>P</td>
<td>Pressure</td>
<td>(Pa)</td>
</tr>
<tr>
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</tr>
<tr>
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<td>Density</td>
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</tr>
<tr>
<td>θ</td>
<td>Stratify angle</td>
<td>(rad)</td>
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<tr>
<td>g</td>
<td>Gravitational acceleration</td>
<td>(m s(^{-2}))</td>
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<td>λ</td>
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<td>(W K(^{-1}) m(^{-2}))</td>
</tr>
<tr>
<td>G</td>
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<td>(kg m(^{-2}))</td>
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<td>(kW m(^{-2}))</td>
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<tr>
<td>f</td>
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<td></td>
</tr>
<tr>
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<td>Refrigerant</td>
<td></td>
</tr>
<tr>
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<td>Saturated</td>
<td></td>
</tr>
<tr>
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<td>Wall</td>
<td></td>
</tr>
<tr>
<td>f</td>
<td>Film</td>
<td></td>
</tr>
<tr>
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<td>Liquid</td>
<td></td>
</tr>
<tr>
<td>V</td>
<td>Vapor</td>
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**Subscripts**

- \(b\): Evaluated at bulk temperature
- \(r\): Refrigerant
- \(sat\): Saturated
- \(w\): Wall
- \(f\): Film
- \(L\): Liquid
- \(V\): Vapor
CHAPTER 1: INTRODUCTION

There are generally two types of correlations for the heat transfer predictions when it comes to in-tube heat exchanger designs: single phase and two phase correlations. Dittus-Boelter and Gneillinski correlations that assume single phase turbulent flow are considered to have highly accurate predictions in single phase region. In two-phase region, Cavallini and Thome are generally used to satisfactorily predict the HTC. However, discontinuities exist between single phase and two phase correlations at quality one and zero. The underlying reason is the assumption of thermodynamic equilibrium that does not hold in real heat exchangers. To properly treat these discontinuities, two additional zones are proposed as CSH and CSC regions. The heat transfer in CSH region was described earlier by Kondo and Hrnjak (2011a, 2011b, 2012). A correlation was proposed by combining the existing single-phase and two-phase correlations in CSH region. Then, Agarwal and Hrnjak (2013) refined the correlation and introduced the pressure drop behavior in CSH region. After that, Meyer and Hrnjak (2014) further described the pressure drop behavior in CSH region and confirmed the existence of liquid layer in superheated region through flow visualization. Even though the HTC correlation shows satisfactory prediction in CSH region, little of the physics behind heat transfer behavior in this region is explained and a more physical model is needed for the generalization of the correlation and to make it more accurate.

This study focuses on the flow characterization and heat transfer mechanism in CSH, as well as unifying the correlations in CSH and TP regions. The goal is to eventually develop a more physical and generalized model. The goal is approached by first understanding the physics behind
condensation through each zone. To better understand the physics, under different working conditions, heat transfer data is taken, two phase flow structure is visualized and film thickness distribution is measured simultaneously. After establishing the connection between heat transfer, flow regime and film structure, the flow characteristics and heat transfer mechanism of condensation will be better understood and building a more physical model will become possible.

The remainder of this document discloses the detailed information regarding this study. Chapter 2 first presents the background information from a brief literature review on heat transfer, void fraction and film thickness measurement techniques. Chapter 3 presents the flow characterization from flow visualization and film thickness measurement under different working conditions. The effect of mass flux and heat flux on onset of condensation, film thickness and flow regime is discussed. The limitations of conventional approach for flow regime prediction is addressed by the proposal of a new flow map and its validation. Void fraction in CSH region is also discussed. Chapter 4 shows the HTC data and discusses the effect of mass flux and heat flux on HTC in combine with the results from Chapter 3. A new heat transfer coefficient \((HTC_f)\) is proposed to unify the heat transfer coefficient of condensation in CSH and TP region. Discussions are made to differentiate it from conventional heat transfer coefficient and identify the physical meaning embedded in this new heat transfer coefficient. Chapter 5 concludes all the work and make suggestions to the future directions. More details of the research is grouped in the appendices.
CHAPTER 2: LITERATURE REVIEW

Over the years, many researcher have been looking at subject of condensation. Large number of correlations were built with improving understandings on fluid properties, flow regime, void fraction heat transfer and pressure drop. The following literature review focuses on previous studies that are crucial for understanding the process of condensation, intending to illustrate a brief description of correlations and techniques that might be useful in this study.

2.1 Single phase heat transfer

Dittus-Boelter correlation is one of the earliest and most famous single phase heat transfer correlations for in-tube turbulent flow. The correlation is valid for Prandtl’s number between 0.6 and 120, Reynold’s number larger than 10000. The correlation was later improved by researchers like Colburn (1933), Sieder and Tate (1936) and so on.

Gnielinski’s correlation is considered the most accurate single phase heat transfer correlation for turbulent flow in tubes. The correlation is valid for Prandtl’s number between 0.5 and 2000, Reynold’s number between 3000 and $5 \times 10^6$. Both correlations are considered well rounded and the accuracies are much higher than two phase flow correlations.

Dittus-Boelter correlation:

$$Nu_D = 0.027Re_D^{4/5}Pr^{0.3}$$

Gnielinski’s correlation:
\[
\text{Nu}_D = \frac{\left(\frac{f}{8}\right) (Re_D - 1000)Pr}{1 + 12.7 \left(\frac{f}{8}\right)^{0.5} (Pr^{2/3} - 1)}
\]

\[
f = (0.79\ln(Re_D) - 1.64)^{-2}
\]

2.2 Two phase heat transfer

The correlation for two phase heat transfer is considered much more complicated and inaccurate compared to single phase correlation. Two correlations are widely used and introduced below.

Thome et al. (2003) proposed a correlation for horizontal tubes based on flow regimes. If the flow regime is determined to be stratify-wavy or stratified flow, the correlation regards the flow at the bottom of the tube to be turbulent flow and the top of the tube to be falling film. Otherwise the heat transfer is pure turbulent convective. Thus the expression of heat transfer coefficient for the bottom of the tube takes the form of either pure convective heat transfer coefficient of turbulent film or a combination of turbulent film and Nusselt falling film. Based on void fraction correlations, the stratified angle can be determined, then the local heat transfer coefficient can be calculated. This correlation uses flow regimes and existing physical models as its base-structure and is thus considered more physical compared to other correlations.

Cavallini et al. (2006) proposed a correlation for smooth round tube dividing flow into two categories: \(\Delta T\)-dependent and \(\Delta T\)-independent, which requires only one transition criteria.
For each category, there is a correlation calculating the corresponding heat transfer coefficient. This correlation is relatively more empirical and easier to use. Good agreement was achieved for many refrigerants and working conditions for both correlations.

Thome’s correlation:

\[
\alpha_{tp} = \frac{\alpha_f r \theta + \alpha_c (2\pi - \theta) r}{2\pi r}
\]

\[
\alpha_c = \frac{0.003 Re_0^{0.74} Pr_0^{0.5} \lambda_l f_i}{\delta}
\]

\[
\alpha_f = 0.665 \left[ \frac{\rho_l (\rho_l - \rho_v) g h_{LV} \lambda_l^{3/3}}{\mu_l dq} \right]^{1/3}
\]

Cavallini’s correlation:

\[
J_G^T = \left\{ \left[ \frac{7.5}{(4.3X_{tt}^{1.111} + 1)} \right]^{-3} + C_T^{-3} \right\}^{-1/3}
\]

\[
J_G = \frac{xG}{[gd \rho_G (\rho_L - \rho_G)]^{0.5}}
\]

\(\Delta T\)-independent \( (J_G > J_G^T)\):

\[
\alpha_A = \alpha_{LO} \left[ 1 + 1.128 x^{0.817} \left( \frac{\rho_L}{\rho_G} \right)^{0.3685} \left( \frac{\mu_L}{\mu_G} \right)^{0.2363} \left( 1 - \frac{\mu_G}{\mu_L} \right)^{2.144} \left( Pr_L \right)^{-0.1} \right]
\]

\(\Delta T\)-dependent \( (J_G \leq J_G^T)\):

\[
\alpha_D = \left[ \alpha_A \left( \frac{J_G^T}{J_G} \right)^{0.8} - \alpha_{STRAT} \right] \left( \frac{J_G}{J_G^T} \right) + \alpha_{STRAT}
\]

\[
\alpha_{LO} = \frac{0.023 Re_0^{0.8} Pr_0^{0.4} \lambda_l}{d}
\]
\[ \alpha_{STRAT} = 0.725 \left\{ 1 + 0.741 \left[ \frac{1 - x}{x} \right]^{0.3321} \right\}^{-1} \left[ \frac{L_p (\rho_L - \rho_G) g h \mu L d \Delta T}{\mu_L d \Delta T} \right]^{0.25} + (1 - x^{0.087}) \alpha_{LO} \]

2.3 Heat transfer in CSH region

Kondo and Hrnjak (2011a, 2011b, 2012) described the existence of liquid film while the bulk flow is still in superheated region due to the temperature gradient inside of the tube. This phenomenon also explained the discontinuity between conventional single phase correlation and two phase correlation at quality one and zero. Kondo-Hrnjak correlation was then made to try to account for the effect brought from liquid film. The correlation took asymptotic approach to make sure at the onset of condensation and quality one the expression would be exactly Gnielinski's correlation and Cavallini’s correlation. The ratio between the two correlations for other states of the flow is calculated using the ratio between two temperature differences: temperature difference between bulk and saturation as well as temperature difference between saturation and wall. The prediction of heat transfer coefficient agrees very well with experimental data.

Kondo-Hrnjak correlation:

\[ \alpha = \frac{\alpha_{Gnielinski}(T_r - T_{sat}) + \alpha_{Cavallini}(T_{sat} - T_w)}{T_r - T_w} \]

Agarwal and Hrnjak (2013) refined the correlation by taking the difference between area occupied by vapor and liquid into account. The film structure is assumed to be Nusselt falling film. The values of HTC calculated from this approach do not vary much from that from Kondo-
Hrnjak correlation. A detailed implementation of calculating HTC in each region is also described.

Meyer and Hrnjak (2014) directly visualized the flow in CSH region and confirmed the existence of liquid-film layer. A brief description of the possible reason for heat transfer and pressure drop behavior in CSH region is also brought up.

2.4 Film thickness measurement techniques

The thickness of liquid film is an important parameter for two-phase flow. The measurement of film thickness is generally divided into two categories: direct measurement and indirect measurement.

Critical angle method was developed by Hurlburt and Newell (1996) as an optical technique using total reflection. The technique has advantages includes non-intrusive, low cost, fast temporal response, easy to use and little calibration. Shedd and Newell (1998) calibrated and described the principles of the method. Basically, by using the Snell’s law with known refractive indices of vapor and liquid, the critical angle where total reflection happens can be determined and thus can be related to the thickness of the liquid film. The principle of this method was developed for finding the film thickness on a flat surface.

Wujak and Hrnjak (2011) described using critical angle method on curved surface. Optical model was made to relate the length of minor axis with thickness of the film.
2.5 Void fraction correlations

Void fraction has a strong relation with heat transfer and flow characteristics. Numerous correlation of void fractions were built. Some most widely used ones include homogenous, Zivi, Smith and Lockhart-Martinelli. Homogenous model assumes vapor and liquid to travel at the same speed and thus the form is most simplified. Zivi (1964) proposed his correlation modified from homogenous model. By applying the principle of minimum entropy generation, the slip ratio could be calculated. Zivi’s correlation predicts the lower bound of void fraction for most cases while homogenous model is responsible for the upper bound. Lots of void fraction correlation takes the form of modified homogenous model with a slip ratio determined either analytically or experimentally. Graham and Newell (1997) looked specifically into the void fraction during condensation of refrigerant and developed a correlation based on Froude rate parameter. Nino, Hrnjak and Newell (2002) developed a flow regime based model for refrigerant void fraction in microchannels.

\[ \alpha = \frac{1 + \frac{1}{x} \rho_g S}{\rho_f} \]

Zivi’s correlation:

\[ S = \frac{\rho_g}{\rho_f} \]

Smith’s correlation:

\[ S = K + (1 - K) \left[ \frac{\rho_f}{\rho_g} + K \left( \frac{1 - x}{x} \right) \right]^\frac{1}{2} \]
CHAPTER 3: FLOW CHARACTERIZATION OF CONDENSATION

The existence of condensation in superheated region is described by Kondo and Hrnjak (2011a, 2011b, 2012) and Agarwal and Hrnjak (2013) through measurement of heat transfer coefficient and pressure drop in superheated region. The transition behavior of heat transfer coefficient and pressure drop in condensing superheated region is considered as an indirect indication of condensation in superheated region. Meyer and Hrnjak (2014) further proved the existence of liquid film in superheated region from flow visualization and film thickness measurement of one refrigerant in one working condition. These studies confirmed that condensation can happen in superheated region, yet the physics behind the heat transfer and pressure drop measurements needs to be better explained. Moreover, there is still no prediction regarding the flow regimes in CSH region, and little discussion about the flow regime at early stages of condensation. In order to address these issues and better understand the physics, flow visualization, film thickness and heat transfer measurement were performed simultaneously at different working conditions for the purpose of comparison. This chapter focuses on the flow visualization and film thickness measurement. The visualization of flow studies the effect of working conditions on the flow characters, including the onset of condensation and flow regime with emphasis in CSH region. The film thickness measurement is refined from Meyer and Hrnjak’s work. It quantitatively supports the visualization by giving void fraction and a more objective determination of flow regimes in condensing superheated region. Besides, the visualization and film thickness measurement can be used as a strong tool for the modeling of heat transfer coefficient, which will be discussed in detail in Chapter 4.
3.1 Experimental facility

3.1.1 Visualization

The visualization section is a coaxial heat exchanger made out of glass. Water was used as secondary fluid to cool the refrigerant, thus the visualization was diabetic. High-speed video was recorded at 1000 frames per second and the resolution was 512 by 512 pixels. Phantom Cine Viewer software from Vision Research was used to process the video. The bulk enthalpy of refrigerant was taken as the inlet enthalpy of visualization section.

3.1.2 Film thickness measurement

The film thickness measurement is based on the principle proposed by Hurlburt and Newell (1996) and refined by Shedd and Newell (1998). With this method, a focused light beam is shined on the surface of duct tape wrapped around a glass tube. The light is assumed to be
diffused uniformly to each direction. The light appearing on the tape is the brightest at the spot where light is shined on (location of light source). Generally speaking, the further away from the source, the lower the brightness of light is. However, at some point the incident angle will reach critical angle for total reflection. Since all of the light is reflected when total reflection happens, there is a sudden increase of brightness. The location of the jump in brightness forms an ellipse whose size is dependent on the thickness of the liquid film layer inside the tube. The major diameter is chosen to be related to the film thickness because the major diameter denotes the axial direction of the glass tube where there is no curvature. Hence the optical model can be simplified as a flat plate. The ellipse is caught on a web camera and a correlation is built to convert major diameter to film thickness. The major diameter is related to a film thickness via an optical model described as below.

\[
Major\ Diameter = 4 \times \left( \frac{\delta_{film}}{\sqrt{n_f^2 - 1}} + \frac{\delta_{wall}}{\sqrt{n_w^2 - 1}} \right)
\]

To get the film distribution around the tube, four photos of ellipse are taken at each angle of 0°, 60°, 90°, 135°, 180°, 225°, 270°, and 300° clockwise from the bottom of the tube. The film thicknesses calculated at each angle are then averaged.

3.2 Film thickness measurement calibration

Wujek and Hrnjak (2011) developed an optical model for film thickness measurement in round tube relating minor diameter of the ellipse with film thickness. Meyer and Hrnjak (2014) continued to use this model in their work. However, neither has a complete calibration of their device before experiment, which made their results less reliable. In addition, the minor diameter will be affected by the curvature of the tube and tube diameter, making the model much more
complicated and inaccurate. Therefore, a calibration was performed to relate major diameter and film thickness in order to make the measurement more accurate. The calibration of the optical film thickness measurement was done by simultaneously measuring a liquid film in a half glass tube using two different methods: critical angle method and contact needle method. HFE 7100 was used here as calibration fluid due to its similar properties with refrigerants. The result using optical model agrees well with needle contact method within 10 percent discrepancy.

Figure 2: Film thickness measurement schematic

Figure 3: Film thickness measured from critical angle method vs contact needle method
3.3 Visualization and discussion

![Figure 4: Screen shots from high-speed videos of R134a condensing at different working condition](image)

### 3.3.1 Onset of condensation

From the visualization, the location of condensation onset varies with different working conditions. When heat flux is fixed, the higher the mass flux, the later (lower enthalpy) when condensation happens; when mass flux is fixed, the higher the heat flux, the earlier (higher enthalpy) the condensation happens. To summarize, the state of flow and single phase heat transfer dictate the onset of condensation. According to the videos and experimental data, whenever the wall temperature drops to saturation temperature, there will be liquid refrigerant on the tube wall.
3.3.2 Flow regime at early stages of condensation

When the wall temperature drops below saturation temperature, liquid starts to appear on the tube wall and in very short time, liquid film will be formed. Due to the shear or gravitational effect, the film will be removed or dragged downwards but condensate continues to form everywhere on the tube wall. Since at the beginning, liquid film is very thin and surface tension prevails, the flow regime has to be annular. The visualization confirms that the condensation always starts in annular flow regime and the process of liquid film formation (formation of drops, partial film into complete film) is extremely short in time.

3.3.3 Effect of mass flux and heat flux on flow regimes

From the visualization, it can be seen that mass flux has a clear effect on flow regime. The higher the mass flux, the larger portion of the condensation process is in annular regime. On the contrary, the lower the mass flux, the larger portion of the condensation process is in stratify regime. Meanwhile, the film thickness is thinner at the same enthalpy for higher mass flux since the condensation rate (heat flux) is the same for each different mass flux. The higher mass flux creates greater shear at interface and pulls more liquid away. Unlike mass flux, heat flux has little effect on flow regime. At the same mass flux, all three cases with different heat flux displayed almost identical flow regime at each enthalpy.
3.3.4 New flow map for the entire condensation process

For conventional condensation flow maps, the onset of condensation is always at quality one, the flow is not necessarily in annular regime at the beginning of condensation, and most of them are developed under adiabatic conditions. To address the above-mentioned issues, a new flow map is proposed below. It is modified from a flow map proposed by Hajal, Thome and Cavallini (2003). The new flow map satisfactorily predicts the flow regime across the entire condensation process and corrects the problems in conventional flow maps. To draw the new flow map, follow the method proposed by Hajal et al. and make the following modifications:

1) Define enthalpy at the onset of condensation to be superficial quality one. Keep enthalpy of thermodynamic quality zero as superficial quality zero. Adjust each enthalpy to its superficial quality and use the superficial quality just as it is thermodynamic quality in the equations.

2) Skip the process of finding minimum $G_{mist}$ and use the value as calculated for higher superficial quality instead of using minimum $G_{mist}$ as stated in the paper.

3) After finding $G_{wavy,min}$, calculate new $G_{wavy}$ when superficial quality is larger than that of $G_{wavy,min}$ using $G_{wavy} = G_{wavy,min}(1 - x)^{0.5}$, making $G_{wavy}$ approach zero as superficial quality approaches one.

4) Changing the last term in equation for $G_{strat}$ from $(+20x)$ to $(+20 - 40x^2)$, making $G_{strat}$ approach zero as superficial quality approaches one.
Figure 5: Modified flow map vs. flow visualization at different working conditions

Figure 6: Modified flow map vs. conventional flow map at G=200 kg m$^{-2}$, Q=10 kW m$^{-2}$
3.4 Film thickness measurement

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</table>

**Figure 7: Comparison between flow visualization and film distribution**

- **G=200 [kg m⁻² s⁻¹]**
- **G=150 [kg m⁻² s⁻¹]**
- **G=100 [kg m⁻² s⁻¹]**
- **G=50 [kg m⁻² s⁻¹]**
- **Q=15 [kW m⁻²]**
- **Q=5 [kW m⁻²]**

- R134a
- Q=10 [kW m⁻²]
- P=1.319 MPa
- Tₑₜ=50°C
3.4.1 Film distribution in early stages of condensation

Fig. 7 shows the comparison between liquid-film distributions recorded from flow visualization and determined from critical angle method. It is obvious that the distributions determined from critical angle method catches the trend of growth of liquid film. By altering mass flux and fixing heat flux, the results from film thickness measurement demonstrates that with higher mass flux, the onset of condensation is delayed, the film is thinner at same enthalpy and the flow is more in annular flow regime than stratify flow regime. By fixing mass flux and altering heat flux, the results from film thickness measurement demonstrates that with higher heat flux, the onset of condensation is advanced, the film is thicker at same enthalpy and the flow regime is almost the same. The conclusion drawn from film thickness measurement is the same with that from flow visualization.

Even though the critical angle method has lots of advantages such as non-intrusive, low cost and fast temporal response, the film thickness measurement has its limits. The range of film thickness this method can measure is limited. First, the accuracy of this method relies on the accurate of image recording and processing. If the film thickness is too small, the system error will create huge relative uncertainty, making the result less trustful. Also, the interface between wall and liquid refrigerant creates a second ellipse from its total reflection. There is a chance for those two ellipses to be too close to distinguish the one from liquid-vapor interface. Last but not least, wavy structure would make the measurement unstable. In this study, four pictures are taken at a same condition and the average value is considered the representative of film thickness.
However, if the flow becomes too wavy, the measurement might not physically make sense. Therefore, the range of film thickness in this study is selected to be 0 to 0.8 mm.

3.4.2 Void fraction in CSH and early stage of TP

![Figure 8: Void fraction of R134a condensing at same heat flux and different mass flux](image)

*Figure 8: Void fraction of R134a condensing at same heat flux and different mass flux*
Fig. 8 and Fig. 9 illustrate the void fraction determined from film thickness measurement and modified void fraction model with different working conditions. Similar to conventional flow map, conventional void fraction model assumes equilibrium, thus void fraction is always one at equality one and zero at quality zero. The experimental data suggests void fraction fall below one before quality one depending on working conditions.

Fig. 8 shows that void fraction increases and slip ratio decreases when mass flux increases. Even though result shows that mass flux affects the slip ratio of flow in condensation, most void fraction correlations do not have mass flux as input. Fig. 9 shows that void fraction decreases and slip ratio does not change when heat flux increases. Here, the void fraction correlation is modified in the same way as flow map by defining a superficial quality one at the onset of condensation. Slip ratios are chosen to fit the experimental data. It is very important to
be able to predict slip ratio as a function of fluid properties, superficial quality and mass flux in the future.
CHAPTER 4: HEAT TRANSFER IN CONDENSATION FROM SUPERHEATED VAPOR

Condensation in smooth round tube happens in a variety of scenarios such as air conditioning and refrigeration systems, power plants cooling systems and so on. A good heat transfer model of smooth round tube not only gives designers a useful tool to size their heat exchangers in different working conditions, but also provides physical insight of heat transfer mechanism and serves as a baseline model for other geometries. Conventionally, the first droplet of condensation happens when the bulk quality of the flow is one and the condensation ends at quality zero. Therefore, almost all the two phase heat transfer model applies to quality ranging from one to zero. However, the implicit assumption of thermal equilibrium does not hold in real heat exchangers where temperature gradient is necessary for condensation to happen. This is the reason why there is almost always a discontinuity between two phase and single phase correlations. To bridge this discontinuity, Kondo and Hrnjak (2011a, 2011b, 2012) proposed a correlation that asymptotically combined the two phase and single phase correlation to make a smooth transition of heat transfer coefficient (HTC) between two regions, and they define the new region where the transition happens to be condensing superheated region (CSH). However, because of the empirical nature of the correlation, the model proposed by Kondo and Hrnjak takes a form where the heat is transferred in parallel from core vapor and liquid film, which is counter intuitive because thermal resistance between core vapor and liquid film should be in series. Also, the peak of heat transfer coefficient consistently occurs at quality equal to one suggests some counteracting factors balancing at quality one, which cannot be found. Therefore, a better understanding of the heat transfer mechanism in CSH is needed before a more physical
model can be proposed. This chapter focuses on the heat transfer coefficient at different working conditions. By linking heat transfer coefficient with flow regimes and film distribution discussed in Chapter 2, a new definition of heat transfer coefficient, described as “film heat transfer coefficient” is proposed as a preparation for the new physical model.

4.1 Experimental facility

As can be seen in the experimental schematic drawing, liquid refrigerant is pumped into an electric heater where refrigerant is heated until it becomes superheated vapor. In the mixing chamber, the refrigerant is mixed and the state is determined by measuring pressure and temperature. Then the state of the flow is adjusted in a precooler whose secondary fluid is water. Through the energy balance from the water side, the condition of test section inlet is determined. The test section consists of a 150 mm horizontal smooth copper tube whose inner diameter is 6.1 mm. Heat transfer rate is determined from the energy balance of water side. Wall temperature is determined from the twelve thermocouples welded onto the tube wall. Heat transfer coefficient calculated from heat flux and temperature difference is considered to be quasi-local. After the test section, refrigerant flows through the film thickness measurement section and visualization section described in Section 3. The refrigerant is further cooled into subcooled liquid and fed into the pump.
4.2 Heat transfer result and discussion

Fig. 11 shows HTC against enthalpy for R134a condensing at 1.319 MPa with same heat flux and different mass flux. In TP region, it is apparent that with higher mass flux, the HTC is higher. This is reasonable because with higher mass flux, first of all, there is stronger advection, which takes away more heat. Secondly, the liquid vapor interaction is stronger for higher mass flux, which creates more mixing. Moreover, from Chapter 3 it is evident that higher mass flux means thinner film, whose thermal resistance is lower. In CSH region, however, the mass flux does not have any effect on HTC. Since the same logic applies also to CSH region, the behavior seems to be strange. The reason will be discussed later in this Chapter.
Fig. 11 shows HTC against enthalpy for R134a condensing at 1.319 MPa with same mass flux and different heat flux. In TP region, HTC is not affected by heat flux. This is reasonable because even though higher heat flux means higher condensation rate, thicker film and higher thermal resistance, it has been shown in Section 3 that flow regime is not dependent on heat flux. Since flow regime is one of the most important factors that alter the heat transfer behavior, heat flux should not alter heat transfer behavior a lot, either. In CSH region, again, the heat transfer behaves oddly because it shows that the higher heat flux, the higher HTC. In theory, if the heat flux has any effect on HTC, the higher the heat flux, the lower the HTC should be, since the thermal resistance is higher. Additionally, for both cases, either fixed heat flux or fixed mass flux, HTC always peak at quality one. There is, however, no physical explanation suggesting two counteracting parameters that could consistently give the highest HTC at that quality.
The reason for the odd behavior of HTC actually lies in the mathematically definition of HTC itself. Conventionally, HTC is defined as heat flux divided by difference between bulk and wall temperature as is shown in equation below. In TP region, bulk temperature is automatically saturation temperature everywhere. In CSH region, however, the bulk temperature decreases dramatically with enthalpy and the temperature difference of superheat of bulk is generally much higher than subcool of wall. Therefore, in CSH region, the subcool is usually neglected. Since the superheat is dictated only by the state of the flow, the HTC is essentially only a function of heat flux. This explains why HTC increases as heat flux increases and is not dependent on mass flux in CSH region. As for the peak, in CSH region, superheat of bulk consistently decreases as refrigerant condenses more and more. Hence the HTC decreases as enthalpy decreases. In TP region, since superheat is zero, HTC consistently decreases as condensation marches on because subcool of wall becomes more and more.
\[
HTC = \frac{Q}{T_b - T_w} = \frac{Q}{(T_b - T_{sat}) + (T_{sat} - T_w)} = \frac{Q}{T_{superheat} + T_{subcool}} \left\{ \begin{array}{l}
\approx \frac{Q}{T_{superheat}} \quad \text{(in CSH)} \\
= \frac{Q}{T_{subcool}} \quad \text{(in TP)}
\end{array} \right.
\]

4.3 Film heat transfer coefficient

In section 4.2, the heat transfer measurement shows the counterintuitive trend of conventional HTC, which is caused by the different definition of driving force, the temperature difference. In TP region, the bulk temperature is consistently saturation temperature, while in CSH region, the bulk temperature can be much higher. However, if the driving force is redefined as the temperature difference between liquid vapor interface temperature as long as liquid is present, the expression of driving force in TP and CSH region will be identical. After refining the driving force, the HTC becomes local, which is focusing on the heat transfer across the liquid film. Since whatever heat that is removed from the refrigerant, it has to go through the liquid film into the wall, physically the new definition makes sense also. Note that by focusing on the heat transfer across the film, the scenario becomes much more like boundary layer problem, whose HTC at the very beginning approaches infinite. Physically this is incorrect because we neglect scenarios such as nucleation of droplets or partial film and the definition of driving force will deviate from that of single phase correlations, in which bulk temperature is used. Mathematically, however, it is correct because when the condensation starts, an infinite HTC means wall temperature is saturation temperature, which is exactly the case. Moreover, it is always heat transfer rate that matters instead of HTC. The conventional HTC seems to be low in CSH region while the driving force is actually very high, meaning the heat transfer in this region
could also be very high. The newly defined HTC is called film heat transfer coefficient, denoted $HTC_f$.

$$HTC_f = \frac{Q}{T_{sat} - T_w}$$

Fig. 13 shows the comparison between conventional (bulk) HTC and film HTC. It is obvious that film HTC could catch the effect of mass flux to heat transfer behavior and forms a smooth curve throughout the entire condensation process. It shows that with higher mass flux, film HTC is higher because of thinner liquid film, stronger advection as well as more mixing. The onset of condensation is located at where film HTC approaches infinity.

![Figure 13: Comparison between conventional HTC and film HTC with different mass flux](image)

Similarly, when heat flux is fixed, unlike conventional HTC, film HTC forms a smooth curve all the way through. The effect of heat flux can be seen at the early stages of condensation.
due to different location of condensation onset as well as film thickness described in Section 3. Hence the higher heat flux, the lower film HTC at the beginning. The conventional HTC, however, shows opposite trend. Later as the state of flow gets closer to TP region, the effect of film thickness is outweighed by that of flow regime. Since the flow regime of condensation does not depend on heat flux, the film HTC will not be affected by heat flux then.

![Figure 14: Comparison between conventional HTC and film HTC with different heat flux](image)

Film HTC unifies the heat transfer performance in CSH and TP region, making it possible in the future to use one model to predict the film HTC throughout the entire condensation process. The model should predict infinite film HTC at the onset of condensation in CSH region and film HTC will sharply decrease due to the increase of film thickness. Meanwhile, the accumulation of liquid at bottom of the tube increases the stratification angle, meaning turbulent convective HTC takes more and more portion of total film HTC compared to
falling film HTC. It would be better if the model could take the other flow regimes (other than annular and stratify flow regimes) into consideration. Compared to conventional heat transfer correlations, the model should be able to catch the condensation in CSH region. The values for film HTC should be consistent with that predicted by conventional heat transfer correlations in TP region.
CHAPTER 5: CONCLUSION

5.1 Summary

The research is divided into two parts. The first part is flow characterization in order to understand the flow regime, film distribution as well as void fraction with emphasis in CSH region. The second part is study of heat transfer in order to develop more physical model with the knowledge from the first part. At this stage, the focus is on the preparation of model development.

For flow characterization, the problem is approached in two different methods: flow visualization and film thickness measurement. The flow visualization shows that condensation always happens in superheated region, as annular flow. The onset of condensation depends on heat flux, mass flux and flow properties while the flow regime is only a dependent on mass flux and flow properties. Based on the observation and understanding of the physics, a new flow map is proposed to predict the flow regime from the onset of condensation in CSH region all the way to the end of condensation at quality zero. The accuracy of prediction is further confirmed by the visualization. The film thickness measurement technique is described and calibrated. The result shows that the technique is capable of measuring the film distribution around the tube wall. Void fraction is derived from the film thickness data and shows that mass flux has more effect on slip ratio compared to heat flux. Both flow visualization and film thickness measurement provide detailed information on structure of the flow. The information is crucial for model development because of the insight of physics obtained and amount of latent heat transfer calculated.
For the study of heat transfer, the conventional HTC presents completely different behavior in TP and CSH region. In TP region, the HTC measurement agrees well with the physics insight from flow characterization that higher mass flux should give higher HTC while heat flux does not affect HTC. In CSH region, HTC behaves counterintuitively in that mass flux does not affect HTC while higher heat flux results in higher HTC. Also, a peak of HTC at quality one is always observed and there is no physical reason for that. The problem is then addressed by the proposal of film HTC, whose driving force is defined consistently as temperature difference between saturation and wall. By comparing HTC and film HTC, film HTC agrees better with flow characterization and is considered a strong tool in developing a unified model for the entire condensation process.

5.2 Future work

Since the physics of two-phase flow is better understood and the objective of model is determined to be film HTC, the most important future work is to develop a more physical and accurate heat transfer model that combines the CSH and TP region. Besides, pressure drop should be taken simultaneously with HTC to better understand how it behaves in CSH region. Currently, all of the work is done in smooth round tube with one inner diameter and one refrigerant. Therefore, it is highly recommended to do experiment on other refrigerants (R32, R1234zf etc.) to generalize the conclusions made in this study. Then, the model for pressure drop can be established and validated by experimental data. In addition, it will be interesting to look at other tube geometries and dimensions. Microchannel tubes would be one of the directions to go to because of the different characters in terms of flow regimes. Micro-grooved tube could also be
looked into. To dig deeper, mixture would potentially lead to much more detailed analysis due to the change in composition locally. Because of this reason, it is expected that the effect of condensation in CSH region should be even more when it comes to mixtures. The heat transfer and pressure drop in condensing subcooled (CSC) region might also be good to be studied once a situation where non-uniformity (high heat flux, low $h_{fg}$ etc.) is still important at the end of condensation.
REFERENCES


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Wujek, S., 2011, Doctoral dissertation, University of Illinois at Urbana-Champaign, Urbana, IL.

Appendix A: Screen shots of flow visualization videos

A.1 R134a condensing at P=1.319 MPa with Q=10 kW m\(^{-2}\) and G=50 kg m\(^{-2}\) s\(^{-1}\)

Figure 15: \(h=490 \text{ kJ kg}^{-1}, T_{\text{superheat}}=60 ^\circ\text{C}\)

Figure 16: \(h=480 \text{ kJ kg}^{-1}, T_{\text{superheat}}=51 ^\circ\text{C}\)

Figure 17: \(h=460 \text{ kJ kg}^{-1}, T_{\text{superheat}}=32 ^\circ\text{C}\)
Figure 18: $h=450 \text{kJ kg}^{-1}$, $T_{\text{superheat}} = 23 \degree\text{C}$

Figure 19: $h=440 \text{kJ kg}^{-1}$, $T_{\text{superheat}} = 14 \degree\text{C}$

Figure 20: $h=430 \text{kJ kg}^{-1}$, $T_{\text{superheat}} = 6 \degree\text{C}$
Figure 21: $h=423 \text{ kJ kg}^{-1}, x = 1$

Figure 22: $h=408 \text{ kJ kg}^{-1}, x = 0.9$

Figure 23: $h=393 \text{ kJ kg}^{-1}, x = 0.8$
Figure 24: $h=378 \text{ kJ kg}^{-1}, x = 0.7$

Figure 25: $h=363 \text{ kJ kg}^{-1}, x = 0.6$

Figure 26: $h=348 \text{ kJ kg}^{-1}, x = 0.5$
Figure 27: $h = 323 \text{ kJ kg}^{-1}, x = 0.4$

Figure 28: $h = 317 \text{ kJ kg}^{-1}, x = 0.3$

Figure 29: $h = 302 \text{ kJ kg}^{-1}, x = 0.2$
Figure 30: $h=287$ kJ kg$^{-1}$, $x = 0.1$

Figure 31: $h=272$ kJ kg$^{-1}$, $x = 0$

Figure 32: $h=250$ kJ kg$^{-1}$, $T_{\text{subcool}} = 14^\circ\text{C}$
A.2 R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$

Figure 33: $h=480$ kJ kg$^{-1}$, $T_{\text{superheat}} = 51$ °C

Figure 34: $h=460$ kJ kg$^{-1}$, $T_{\text{superheat}} = 32$ °C

Figure 35: $h=450$ kJ kg$^{-1}$, $T_{\text{superheat}} = 23$ °C
Figure 36: $h=440 \text{ kJ kg}^{-1}, T_{\text{superheat}} = 14 \degree \text{C}$

Figure 37: $h=430 \text{ kJ kg}^{-1}, T_{\text{superheat}} = 6 \degree \text{C}$

Figure 38: $h=423 \text{ kJ kg}^{-1}, x = 1$
Figure 39: $h=408 \text{ kJ kg}^{-1}$, $x = 0.9$

Figure 40: $h=393 \text{ kJ kg}^{-1}$, $x = 0.8$

Figure 41: $h=378 \text{ kJ kg}^{-1}$, $x = 0.7$
Figure 42: $h=363 \text{ kJ kg}^{-1}, x = 0.6$

Figure 43: $h=348 \text{ kJ kg}^{-1}, x = 0.5$

Figure 44: $h=323 \text{ kJ kg}^{-1}, x = 0.4$
Figure 45: $h=317 \text{ kJ kg}^{-1}, x = 0.3$

Figure 46: $h=302 \text{ kJ kg}^{-1}, x = 0.2$

Figure 47: $h=287 \text{ kJ kg}^{-1}, x = 0.1$
A.3 R134a condensing at $P=1.319$ MPa with $Q=10 \text{ kW m}^{-2}$ and $G=150 \text{ kg m}^{-2} \text{ s}^{-1}$
Figure 51: \(h=450\ \text{kJ kg}^{-1}, T_{\text{superheat}} = 23\ ^\circ\text{C}\)

Figure 52: \(h=440\ \text{kJ kg}^{-1}, T_{\text{superheat}} = 14\ ^\circ\text{C}\)

Figure 53: \(h=430\ \text{kJ kg}^{-1}, T_{\text{superheat}} = 6\ ^\circ\text{C}\)
Figure 54: $h=423 \text{ kJ kg}^{-1}$, $x = 1$

Figure 55: $h=408 \text{ kJ kg}^{-1}$, $x = 0.9$

Figure 56: $h=393 \text{ kJ kg}^{-1}$, $x = 0.8$
Figure 57: $h = 378 \text{ kJ kg}^{-1}, x = 0.7$

Figure 58: $h = 363 \text{ kJ kg}^{-1}, x = 0.6$

Figure 59: $h = 348 \text{ kJ kg}^{-1}, x = 0.5$
Figure 60: $h=323$ kJ kg$^{-1}$, $x = 0.4$

Figure 61: $h=317$ kJ kg$^{-1}$, $x = 0.3$

Figure 62: $h=302$ kJ kg$^{-1}$, $x = 0.2$
Figure 63: $h=287 \text{ kJ kg}^{-1}, x = 0.1$

Figure 64: $h=272 \text{ kJ kg}^{-1}, x = 0$

Figure 65: $h=250 \text{ kJ kg}^{-1}, T_{\text{subcoat}} = 14^\circ\text{C}$
A.4 R134a condensing at P=1.319 MPa with Q=10 kW m$^{-2}$ and G=200 kg m$^{-2}$ s$^{-1}$

**Figure 66**: $h=440$ kJ kg$^{-1}$, $T_{superheat} = 14$ °C

**Figure 67**: $h=430$ kJ kg$^{-1}$, $T_{superheat} = 6$ °C

**Figure 68**: $h=423$ kJ kg$^{-1}$, $x = 1$
Figure 69: $h=408$ kJ kg$^{-1}$, $x = 0.9$

Figure 70: $h=393$ kJ kg$^{-1}$, $x = 0.8$

Figure 71: $h=378$ kJ kg$^{-1}$, $x = 0.7$
Figure 72: $h=363 \text{ kJ kg}^{-1}, x = 0.6$

Figure 73: $h=348 \text{ kJ kg}^{-1}, x = 0.5$

Figure 74: $h=323 \text{ kJ kg}^{-1}, x = 0.4$
Figure 75: $h = 317 \text{ kJ kg}^{-1}, x = 0.3$

Figure 76: $h = 302 \text{ kJ kg}^{-1}, x = 0.2$

Figure 77: $h = 287 \text{ kJ kg}^{-1}, x = 0.1$
Figure 78: $h=272 \text{ kJ kg}^{-1}, x = 0$

Figure 79: $h=250 \text{ kJ kg}^{-1}, T_{	ext{subcool}} = 14^\circ\text{C}$

A.5 R134a condensing at $P=1.319 \text{ MPa}$ with $Q=5 \text{ kW m}^{-2}$ and $G=100 \text{ kg m}^{-2} \text{ s}^{-1}$

Figure 80: $h=450 \text{ kJ kg}^{-1}, T_{	ext{superheat}} = 23^\circ\text{C}$
Figure 81: $h=440 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 14 ^\circ C$

Figure 82: $h=430 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 6 ^\circ C$

Figure 83: $h=423 \text{ kJ kg}^{-1}$, $x = 1$
Figure 84: $h=408 \text{ kJ kg}^{-1}, x = 0.9$

Figure 85: $h=393 \text{ kJ kg}^{-1}, x = 0.8$

Figure 86: $h=378 \text{ kJ kg}^{-1}, x = 0.7$
Figure 87: $h=363$ kJ kg$^{-1}$, $x = 0.6$

Figure 88: $h=348$ kJ kg$^{-1}$, $x = 0.5$

Figure 89: $h=323$ kJ kg$^{-1}$, $x = 0.4$
Figure 90: \( h = 317 \text{ kJ kg}^{-1}, x = 0.3 \)

Figure 91: \( h = 302 \text{ kJ kg}^{-1}, x = 0.2 \)

Figure 92: \( h = 287 \text{ kJ kg}^{-1}, x = 0.1 \)
A.6 R134a condensing at $P=1.319$ MPa with $Q=15$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$
Figure 96: $h=460 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 32^\circ\text{C}$

Figure 97: $h=450 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 23^\circ\text{C}$

Figure 98: $h=440 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 14^\circ\text{C}$
Figure 99: $h = 430 \text{ kJ kg}^{-1}, T_{\text{superheat}} = 6 ^\circ\text{C}$

Figure 100: $h = 423 \text{ kJ kg}^{-1}, x = 1$

Figure 101: $h = 408 \text{ kJ kg}^{-1}, x = 0.9$
Figure 102: $h=393 \text{ kJ kg}^{-1}, x = 0.8$

Figure 103: $h=378 \text{ kJ kg}^{-1}, x = 0.7$

Figure 104: $h=363 \text{ kJ kg}^{-1}, x = 0.6$
Figure 105: h=348 kJ kg\(^{-1}\), \(x = 0.5\)

Figure 106: h=323 kJ kg\(^{-1}\), \(x = 0.4\)

Figure 107: h=317 kJ kg\(^{-1}\), \(x = 0.3\)
Figure 108: $h=302$ kJ $\text{kg}^{-1}$, $x = 0.2$

Figure 109: $h=287$ kJ $\text{kg}^{-1}$, $x = 0.1$

Figure 110: $h=272$ kJ $\text{kg}^{-1}$, $x = 0$
Figure 111: $h=250 \text{ kJ kg}^{-1}, T_{\text{subcool}} = 14^\circ\text{C}$
Appendix B: Film distribution along tube circumference

B.1 R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$

Figure 112: $h=480$ kJ kg$^{-1}$, $T_{\text{superheat}} = 51$ °C

Figure 113: $h=460$ kJ kg$^{-1}$, $T_{\text{superheat}} = 32$ °C
Figure 114: $h=450 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 23 \, ^\circ\text{C}$

Figure 115: $h=440 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 14 \, ^\circ\text{C}$
Figure 116: $h=430 \text{ kJ kg}^{-1}, T_{\text{superheat}} = 6 ^\circ\text{C}$

Figure 117: $h=423 \text{ kJ kg}^{-1}, x = 1$
Figure 118: $h=408 \text{ kJ kg}^{-1}$, $x = 0.9$

Figure 119: $h=393 \text{ kJ kg}^{-1}$, $x = 0.8$

$G=50 \text{ [kg m}^{-2} \text{s}^{-1}]$, $Q=10 \text{ [kW m}^{-2}]$
B.2 R134a condensing at P=1.319 MPa with Q=10 kW m$^{-2}$ and G=100 kg m$^{-2}$ s$^{-1}$

Figure 120: h=460 kJ kg$^{-1}$, $T_{superheat} = 32$ °C

Figure 121: h=450 kJ kg$^{-1}$, $T_{superheat} = 23$ °C
Figure 122: $h=440 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 14 ^\circ\text{C}$

Figure 123: $h=430 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 6 ^\circ\text{C}$
Figure 124: h=423 kJ kg\(^{-1}\), x = 1

Figure 125: h=408 kJ kg\(^{-1}\), x = 0.9
Figure 126: h=393 kJ kg\(^{-1}\), \(x = 0.8\)

Figure 127: h=378 kJ kg\(^{-1}\), \(x = 0.7\)
B.3 R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=150$ kg m$^{-2}$ s$^{-1}$

Figure 128: $h=450$ kJ kg$^{-1}$, $T_{\text{superheat}} = 23^\circ$C

Figure 129: $h=440$ kJ kg$^{-1}$, $T_{\text{superheat}} = 14^\circ$C
Figure 130: $h=430 \text{kJ kg}^{-1}, T_{\text{superheat}} = 6^\circ C$

Figure 131: $h=423 \text{kJ kg}^{-1}, x = 1$
Figure 132: $h=408 \text{ kJ kg}^{-1}$, $x = 0.9$

Figure 133: $h=393 \text{ kJ kg}^{-1}$, $x = 0.8$
Figure 134: $h=378 \text{ kJ kg}^{-1}$, $x = 0.7$

Figure 135: $h=363 \text{ kJ kg}^{-1}$, $x = 0.6$
B.4 R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=200$ kg m$^{-2}$ s$^{-1}$

Figure 136: $h=430$ kJ kg$^{-1}$, $T_{\text{superheat}} = 6$ $^\circ$C

Figure 137: $h=423$ kJ kg$^{-1}$, $x = 1$
Figure 138: $h = 408 \text{ kJ kg}^{-1}$, $x = 0.9$

Figure 139: $h = 393 \text{ kJ kg}^{-1}$, $x = 0.8$
Figure 140: $h=378 \text{ kJ kg}^{-1}$, $x = 0.7$

Figure 141: $h=363 \text{ kJ kg}^{-1}$, $x = 0.6$
B.5 R134a condensing at $P=1.319$ MPa with $Q=5$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$

Figure 142: $h=348$ kJ kg$^{-1}$, $x = 0.5$

Figure 143: $h=440$ kJ kg$^{-1}$, $T_{\text{superheat}} = 14$ °C
Figure 144: $h=430 \text{ kJ kg}^{-1}, T_{\text{superheat}} = 6 ^\circ \text{C}$

Figure 145: $h=423 \text{ kJ kg}^{-1}, x = 1$
Figure 146: $h=408 \text{ kJ kg}^{-1}, x = 0.9$

Figure 147: $h=393 \text{ kJ kg}^{-1}, x = 0.8$
Figure 148: $h=378 \text{ kJ kg}^{-1}, x = 0.7$

Figure 149: $h=363 \text{ kJ kg}^{-1}, x = 0.6$
Figure 150: $h=348$ kJ kg$^{-1}$, $x = 0.5$

B.6 R134a condensing at $P=1.319$ MPa with $Q=15$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$

Figure 151: $h=460$ kJ kg$^{-1}$, $T_{superheat} = 32$ °C
Figure 152: $h = 450 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 23^\circ \text{C}$

Figure 153: $h = 440 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 14^\circ \text{C}$
Figure 154: $h=430 \text{ kJ kg}^{-1}$, $T_{\text{superheat}} = 6^\circ\text{C}$

Figure 155: $h=423 \text{ kJ kg}^{-1}$, $x = 1$
Figure 156: $h=408 \text{ kJ kg}^{-1}$, $x = 0.9$

Figure 157: $h=393 \text{ kJ kg}^{-1}$, $x = 0.8$
Figure 158: $h=378$ kJ kg$^{-1}$, $x = 0.7$
Appendix C: Uncertainty

C.1 Measurement uncertainties

Table 1: Uncertainties of measurement

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<td>$m_{H2O,FC}$</td>
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C.2 HTC uncertainties

Figure 159: Uncertainties of R134a condensing at $P=1.319$ MPa with $Q=5$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$
Figure 160: Uncertainties of R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$

Figure 161: Uncertainties of R134a condensing at $P=1.319$ MPa with $Q=15$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$

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Figure 162: Uncertainties of R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=200$ kg m$^{-2}$ s$^{-1}$

Figure 163: Uncertainties of R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=150$ kg m$^{-2}$ s$^{-1}$
Figure 164: Uncertainties of R134a condensing at P=1.319 MPa with Q=10 kW m$^{-2}$ and G=50 kg m$^{-2}$ s$^{-1}$
Appendix D: Schematic drawing of film thickness measurement principle

Figure 165: Principle of film thickness measurement
Appendix E: Film thickness measurement result

Table 2: R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=50$ kg m$^{-2}$ s$^{-1}$

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Table 3: R134a condensing at $P=1.319$ MPa with $Q=10$ kW m$^{-2}$ and $G=100$ kg m$^{-2}$ s$^{-1}$

<table>
<thead>
<tr>
<th>Avg hout</th>
<th>0°</th>
<th>60°</th>
<th>90°</th>
<th>135°</th>
<th>180°</th>
<th>225°</th>
<th>270°</th>
<th>300°</th>
</tr>
</thead>
<tbody>
<tr>
<td>kJ/kg</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
</tr>
<tr>
<td>460</td>
<td>0.023</td>
<td>0.084</td>
<td>0.053</td>
<td>0.031</td>
<td>0.073</td>
<td>0.003</td>
<td>0.000</td>
<td>0.001</td>
</tr>
<tr>
<td>450</td>
<td>0.052</td>
<td>0.084</td>
<td>0.066</td>
<td>0.048</td>
<td>0.056</td>
<td>0.029</td>
<td>0.019</td>
<td>0.021</td>
</tr>
<tr>
<td>440</td>
<td>0.057</td>
<td>0.064</td>
<td>0.024</td>
<td>0.098</td>
<td>0.048</td>
<td>0.032</td>
<td>0.026</td>
<td>0.044</td>
</tr>
<tr>
<td>430</td>
<td>0.101</td>
<td>0.094</td>
<td>0.104</td>
<td>0.057</td>
<td>0.055</td>
<td>0.000</td>
<td>0.025</td>
<td>0.008</td>
</tr>
<tr>
<td>423</td>
<td>0.097</td>
<td>0.100</td>
<td>0.076</td>
<td>0.043</td>
<td>0.046</td>
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<td>0.029</td>
</tr>
<tr>
<td>408</td>
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<td>0.158</td>
<td>0.107</td>
<td>0.087</td>
<td>0.095</td>
<td>0.100</td>
<td>0.102</td>
</tr>
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<td>0.132</td>
<td>0.118</td>
<td>0.127</td>
<td>0.103</td>
<td>0.101</td>
<td>0.099</td>
</tr>
<tr>
<td>378</td>
<td>0.662</td>
<td>0.230</td>
<td>0.150</td>
<td>0.120</td>
<td>0.123</td>
<td>0.095</td>
<td>0.090</td>
<td>0.144</td>
</tr>
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Table 4: R134a condensing at P=1.319 MPa with Q=10 kW m\(^{-2}\) and G=150 kg m\(^{-2}\) s\(^{-1}\)

<table>
<thead>
<tr>
<th>Avg hout (kJ/kg)</th>
<th>0° (mm)</th>
<th>60° (mm)</th>
<th>90° (mm)</th>
<th>135° (mm)</th>
<th>180° (mm)</th>
<th>225° (mm)</th>
<th>270° (mm)</th>
<th>300° (mm)</th>
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</thead>
<tbody>
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<td>450</td>
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<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.002</td>
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<tr>
<td>440</td>
<td>0.081</td>
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<td>0.012</td>
<td>0.000</td>
<td>0.000</td>
<td>0.011</td>
<td>0.012</td>
<td>0.016</td>
</tr>
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<td>0.012</td>
<td>0.000</td>
<td>0.000</td>
<td>0.011</td>
<td>0.012</td>
<td>0.016</td>
</tr>
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<td>0.065</td>
<td>0.016</td>
<td>0.013</td>
<td>0.015</td>
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<td>0.065</td>
</tr>
<tr>
<td>408</td>
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<td>0.027</td>
<td>0.060</td>
<td>0.000</td>
<td>0.046</td>
<td>0.022</td>
<td>0.142</td>
</tr>
<tr>
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<td>0.428</td>
<td>0.242</td>
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<td>0.048</td>
<td>0.044</td>
<td>0.204</td>
<td>0.218</td>
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<td>0.043</td>
<td>0.068</td>
<td>0.128</td>
<td>0.179</td>
</tr>
<tr>
<td>363</td>
<td>0.667</td>
<td>0.462</td>
<td>0.258</td>
<td>0.129</td>
<td>0.039</td>
<td>0.025</td>
<td>0.057</td>
<td>0.347</td>
</tr>
</tbody>
</table>

Table 5: R134a condensing at P=1.319 MPa with Q=10 kW m\(^{-2}\) and G=200 kg m\(^{-2}\) s\(^{-1}\)

<table>
<thead>
<tr>
<th>Avg hout (kJ/kg)</th>
<th>0° (mm)</th>
<th>60° (mm)</th>
<th>90° (mm)</th>
<th>135° (mm)</th>
<th>180° (mm)</th>
<th>225° (mm)</th>
<th>270° (mm)</th>
<th>300° (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>430</td>
<td>0.046</td>
<td>0.037</td>
<td>0.002</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.029</td>
<td>0.014</td>
</tr>
<tr>
<td>423</td>
<td>0.040</td>
<td>0.037</td>
<td>0.018</td>
<td>0.000</td>
<td>0.000</td>
<td>0.012</td>
<td>0.054</td>
<td>0.020</td>
</tr>
<tr>
<td>408</td>
<td>0.063</td>
<td>0.034</td>
<td>0.011</td>
<td>0.000</td>
<td>0.000</td>
<td>0.011</td>
<td>0.022</td>
<td>0.018</td>
</tr>
<tr>
<td>393</td>
<td>0.127</td>
<td>0.098</td>
<td>0.049</td>
<td>0.014</td>
<td>0.008</td>
<td>0.059</td>
<td>0.073</td>
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</tr>
<tr>
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<td>0.242</td>
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<td>0.020</td>
<td>0.015</td>
<td>0.091</td>
<td>0.181</td>
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<tr>
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<td>0.383</td>
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<td>0.055</td>
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</tr>
<tr>
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<td>0.244</td>
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<td>0.055</td>
<td>0.053</td>
<td>0.221</td>
<td>0.296</td>
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</table>
Table 6: R134a condensing at P=1.319 MPa with Q=5 kW m$^{-2}$ and G=100 kg m$^{-2}$ s$^{-1}$

<table>
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<tr>
<th>Avg hout (kJ/kg)</th>
<th>0°</th>
<th>60°</th>
<th>90°</th>
<th>135°</th>
<th>180°</th>
<th>225°</th>
<th>270°</th>
<th>300°</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
</tr>
<tr>
<td>440</td>
<td>0.047</td>
<td>0.037</td>
<td>0.023</td>
<td>0.008</td>
<td>0.008</td>
<td>0.000</td>
<td>0.000</td>
<td>0.023</td>
</tr>
<tr>
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<td>0.023</td>
<td>0.030</td>
<td>0.030</td>
<td>0.024</td>
<td>0.025</td>
<td>0.012</td>
<td>0.015</td>
</tr>
<tr>
<td>423</td>
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<td>0.005</td>
<td>0.006</td>
<td>0.026</td>
<td>0.000</td>
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<tr>
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<td>0.026</td>
<td>0.042</td>
<td>0.000</td>
<td>0.015</td>
<td>0.013</td>
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<tr>
<td>393</td>
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<td>0.179</td>
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<td>0.026</td>
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<td>0.123</td>
<td>0.172</td>
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<td>0.164</td>
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<td>0.113</td>
<td>0.182</td>
<td>0.285</td>
</tr>
</tbody>
</table>

Table 7: R134a condensing at P=1.319 MPa with Q=15 kW m$^{-2}$ and G=100 kg m$^{-2}$ s$^{-1}$

<table>
<thead>
<tr>
<th>Avg hout (kJ/kg)</th>
<th>0°</th>
<th>60°</th>
<th>90°</th>
<th>135°</th>
<th>180°</th>
<th>225°</th>
<th>270°</th>
<th>300°</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
<td>mm</td>
</tr>
<tr>
<td>460</td>
<td>0.039</td>
<td>0.032</td>
<td>0.015</td>
<td>0.000</td>
<td>0.000</td>
<td>0.004</td>
<td>0.000</td>
<td>0.018</td>
</tr>
<tr>
<td>450</td>
<td>0.037</td>
<td>0.012</td>
<td>0.000</td>
<td>0.000</td>
<td>0.000</td>
<td>0.012</td>
<td>0.000</td>
<td>0.039</td>
</tr>
<tr>
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<td>0.034</td>
<td>0.003</td>
<td>0.002</td>
<td>0.012</td>
<td>0.025</td>
<td>0.076</td>
</tr>
<tr>
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<td>0.046</td>
<td>0.023</td>
<td>0.025</td>
<td>0.011</td>
<td>0.012</td>
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<tr>
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<td>0.113</td>
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<td>0.118</td>
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<td>0.193</td>
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<td>0.224</td>
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<td>0.070</td>
<td>0.107</td>
<td>0.171</td>
<td>0.219</td>
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