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Heat Transfer of Condensation in Smooth Round Tube from Superheated Vapor

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ABSTRACT

Heat transfer of R134a condensing in a horizontal smooth round tube with 6.1 mm inner diameter is investigated in this study. The paper presents heat transfer measurement with mass flux from 50 to 200 kg m^{-2} s^{-1} and heat flux from 5 to 15 kW m^{-2}, showing the effect of mass flux and heat flux on the heat transfer coefficient (HTC). All the measurement is taken at constant pressure of 1.319 MPa, which corresponds to a saturation temperature of 50 ºC. By connecting heat transfer result with flow characterization, the behavior of HTC is explained from the perspectives of both mathematics and physics. The result shows that for fixed heat flux and different mass flux, even though the liquid film for larger mass flux is much thinner in the condensing superheated (CSH) region, the HTC is not affected by mass flux. In the two-phase (TP) region, however, higher mass flux clearly yields higher HTC. Opposite behavior is found when heat flux is varied and mass flux is fixed. In the CSH region, the HTC increases with heat flux, while in the TP region, HTC does not change with heat flux. For both cases, a peak of HTC is presented at quality one, which seems to imply some counter-acting factors that always put heat transfer largest at quality one. The counter-acting factors, however, should not exist based on flow characterization. After mathematically explaining the behavior of HTC in the CSH region, a newly defined HTC named as film HTC (HTC_f) is proposed and ready to serve as a tool in a unified heat transfer model throughout the entire CSH and TP regions.

1. INTRODUCTION

Condensation in smooth round tubes occurs 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 mechanisms and acts as a baseline model for other geometries. Conventionally, the first droplet of condensation appears when the bulk quality of the flow is one and the condensation ends at quality zero. Therefore, almost all of the two-phase heat transfer models apply to quality ranging from one to zero. However, the implicit assumption of thermal equilibrium does not hold in real heat exchangers where a 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 HTC between two regions, and they define the new region which represents the transition to be the CSH region. However, because of the empirical nature of the correlation, the model proposed by Kondo and Hrnjak takes a form in which the heat is transferred in parallel from core vapor and liquid film, whereas thermal resistance between core vapor and liquid film should be in series. Agarwal and Hrnjak (2013) refined the model by trying to determine the ratio of latent heat against sensible heat. The essence of the models in the CSH region, however, remains to be an asymptotic approach between single-phase and two-phase correlations. To better comprehend the process of condensation in the CSH region, Xiao and Hrnjak (2016) studied the flow characteristics of R134a under different working conditions to be able to connect HTC directly to the liquid film.

2. EXPERIMENT DESCRIPTION

2.1 Overview of experimental facility

Figure 1 shows the schematic drawing of the facility where experimental data were taken from. The subcooled refrigerant is pumped into an electric heater where refrigerant is heated until it becomes superheated vapor. Mass flow rate of refrigerant is determined from the mass flow meter before the heater. In the mixing chamber, the refrigerant specific enthalpy is determined by measuring pressure and temperature. Then the state of the flow is adjusted in a pre-cooler and test section whose secondary fluid is water with known mass flow rate, inlet and outlet temperature. Through the energy balance from the water side, the condition of the test section outlet can be calculated. In the film thickness measurement section, film thickness is measured using the critical angle method. In the visualization section, the diabatic condensation process is recorded with a high speed camera. Eventually the refrigerant is subcooled again in the after-cooler and the cycle completes.

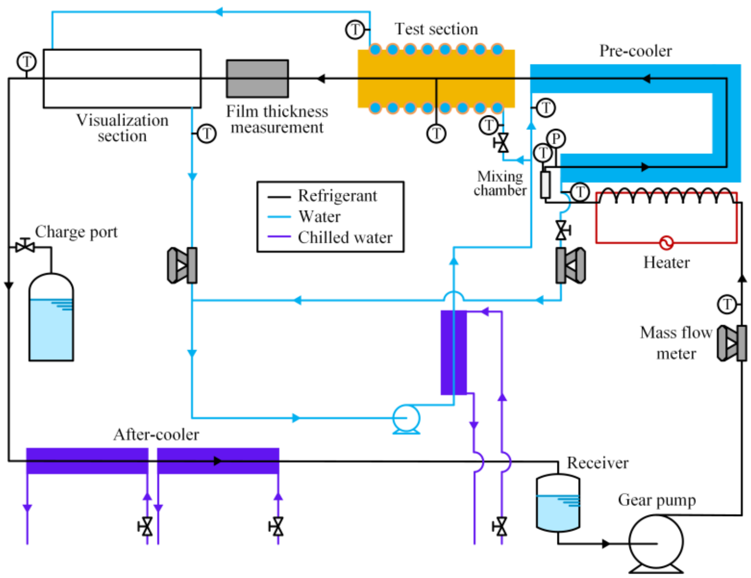

Figure 1: Schematic drawing of experimental facility (Meyer and Hrnjak, 2014)

2.2 Test section

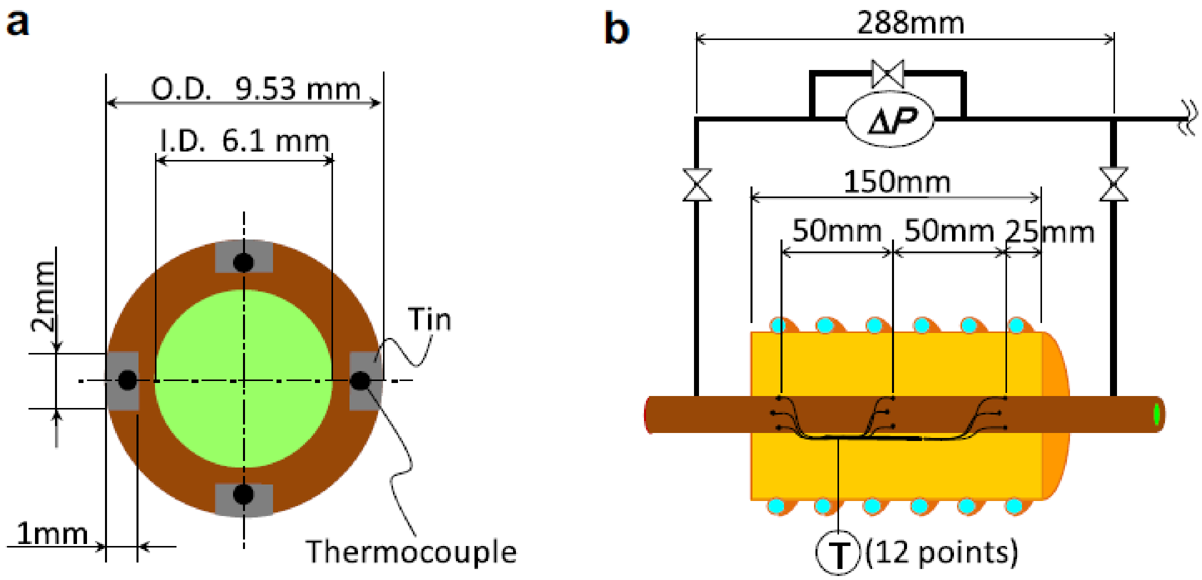

Figure 2: Test section dimensions and thermocouple positions (Kondo and Hrnjak, 2011a)

Figure 2 show the dimensions of the test section and positions of thermocouples. The refrigerant tube is a 150 mm long copper tube with 6.1 mm inner diameter and 9.53 mm outer diameter. The tube is placed horizontally with a water jacket covering it. Thermocouples are embedded at 3 different axial locations, and at each axial location, there are four at top, bottom and side of the tube. Heat flux is measured from the water side with known mass flow rate as well as inlet and outlet temperatures. Wall temperature is taken to be the average of the readings from twelve thermocouples.

3. RESULTS AND DISCUSSIONS

3.1 Heat transfer coefficient measurement

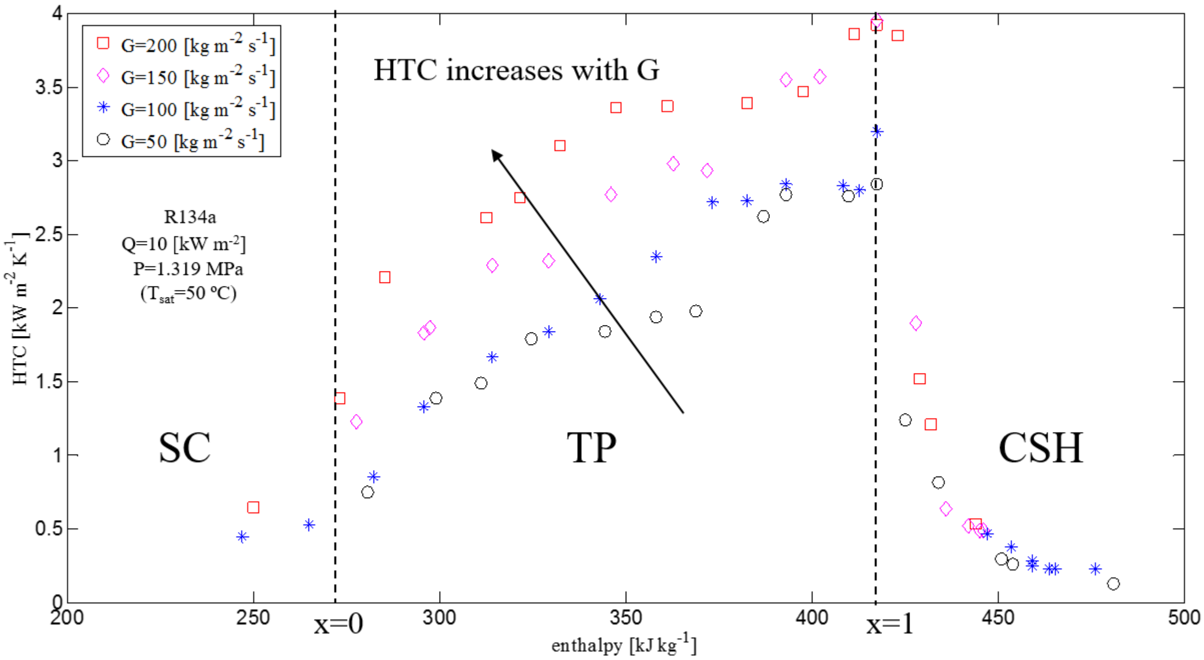

Figure 3: HTC measured under same heat flux and different mass flux

Figure3 shows HTC against enthalpy for R134a condensing at 1.319 MPa with same heat flux of 10 kW m^{-2} and different mass flux from 50 kg m^{-2} s^{-1} to 200 kg m^{-2} s^{-1}. In the TP region, it is obvious 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 heat in a faster manner. Secondly, the liquid-vapor interaction is stronger for higher mass flux, which creates more mixing. This can be directly observed from flow visualization by Xiao and Hrnjak (2016). The film thickness measurement from that work also shows that higher mass flux means thinner film, whose thermal resistance is lower. In the CSH region, however, the mass flux does not have any effect on HTC. Since the same logic applies also to the CSH region, the behavior seems to be counterintuitive. The reason will be discussed later in this paper.

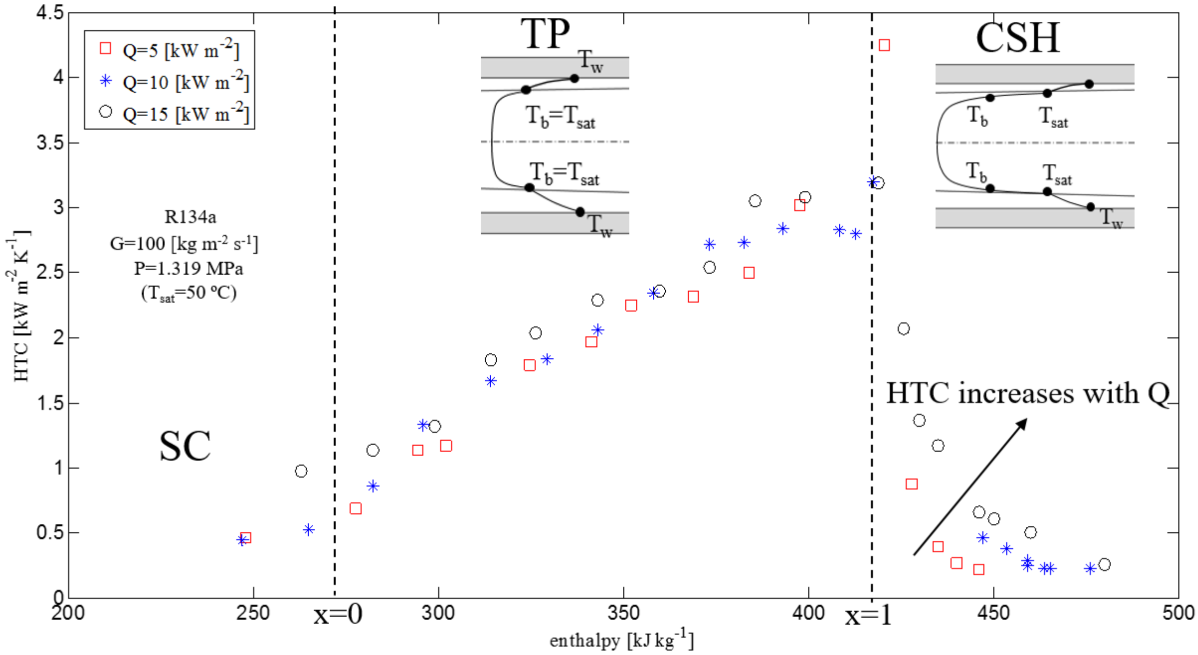

Figure 4: HTC measured under same mass flux and different heat flux

Figure 4 shows HTC against enthalpy for R134a condensing at 1.319 MPa with mass flux of 100 kg m^{-2} s^{-1} and heat flux from 5 kW m^{-2} to 15 kW m^{-2}. In the TP region, heat flux has very limited affected on HTC. This is reasonable because even though higher heat flux and the same mass flux implies higher condensation rate, thicker film and higher thermal resistance, it has been shown in flow characterization by Xiao and Hrnjak (2016) that flow regime is not dependent on heat flux. Since flow regime is one of the most important factors that alters the heat transfer behavior, especially after the early stages of condensation where HTC is extremely sensitive to film thickness, heat flux should not have much effect on HTC. In the CSH region, again, the heat transfer behaves oddly because it shows that with the higher heat flux, the HTC is higher. 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 peaks at quality one. There is, however, no physical explanation suggesting some counteracting parameters that could consistently give the highest HTC exactly at that quality.

3.2 Mathematical interpretation

The reason for the odd behavior of HTC actually lies in the mathematical definition of HTC itself. Conventionally, HTC is defined as heat flux divided by difference between bulk and wall temperature as is shown in the equation below. In the TP region, bulk temperature is automatically saturation temperature everywhere. In the CSH region, however, the bulk temperature decreases dramatically with enthalpy and the superheat of bulk flow is generally much higher than the subcooling of the wall. Therefore, in the CSH region, the subcooling is usually neglected, as is shown in Equation (1). 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 the CSH region, superheat of bulk flow consistently decreases as refrigerant condenses more and more. Hence the HTC increases dramatically as enthalpy decreases. In the TP region, since superheat is zero, the expression for HTC reduces to Equation (2) and HTC consistently decreases as condensation marches on because the subcooling of the wall becomes greater.

$$\mathrm{HTC} = \frac{\mathrm{Q}}{\mathrm{T_b} - \mathrm{T_w}} = \frac{\mathrm{Q}}{(\mathrm{T_b} - \mathrm{T_{sat}}) + (\mathrm{T_{sat}} - \mathrm{T_w})} = \frac{\mathrm{Q}}{\mathrm{T_{superheat}} + \mathrm{T_{subcool}}} \begin{cases} \approx \dfrac{\mathrm{Q}}{\mathrm{T_{superheat}}} \text{ (in CSH)} & (1) \\ = \dfrac{\mathrm{Q}}{\mathrm{T_{subcool}}} \text{ (in TP)} & (2) \end{cases}$$

3.3 Film heat transfer coefficient and its physical interpretation

Even though there is a mathematical explanation for the behavior of HTC in the CSH region, physically, the trend is still counterintuitive. From the visualization and film distribution demonstrated by Xiao and Hrnjak (2016), there is nothing particularly special at quality one, the beginning of the TP region. The film forms at the beginning of the CSH region and continues to grow until the end of the TP region. Treating the CSH and TP regions separately is physically unnecessary. One way to unify the definitions is by focusing on the heat transfer through the film instead of the whole tube since whatever heat that is removed from the refrigerant has to go through the liquid film into the wall. In other words, the wall only sees liquid film flowing along its surface, and vapor is there just to set the configuration of the liquid. If the HTC is redefined locally, the temperature difference will consistently be the difference between liquid-vapor interface temperature and wall temperature. Note that by focusing on the heat transfer across the film, the scenario becomes much more like a boundary layer problem, whose HTC at the very beginning approaches infinity. 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 according to Kondo and Hrnjak (2011a, 2011b, 2012), Agarwal and Hrnjak (2013), Meyer and Hrnjak (2014) as well as Xiao and Hrnjak (2016). Moreover, it is always heat transfer rate that matters instead of HTC. The conventional HTC seems to be low in the CSH region while the driving force is actually very high, meaning the heat transfer in this region could also be very high. In this sense, even though the discrepancy between the Kondo-Hrnjak correlation (Kondo and Hrnjak, 2012) and the Gnielinski correlation (Gnielinski, 1973) may not seem large at some conditions, the rather larger driving force will make a larger difference in heat transfer than HTC itself. In order to have a more physical representation of heat transfer in the CSH region, a newly defined HTC named as film heat transfer coefficient, is presented in Equation (3)

$$HTC_f = \frac{Q}{T_{sat} - T_w} \tag{3}$$

Note that the conversion between conventional HTC and film HTC is very simple due to the fact that the change is almost purely mathematical. It is suggested here to use film HTC as a tool for modeling but always to convert it back to conventional HTC for easier use by engineers.

3.4 Comparison between conventional HTC and film HTC

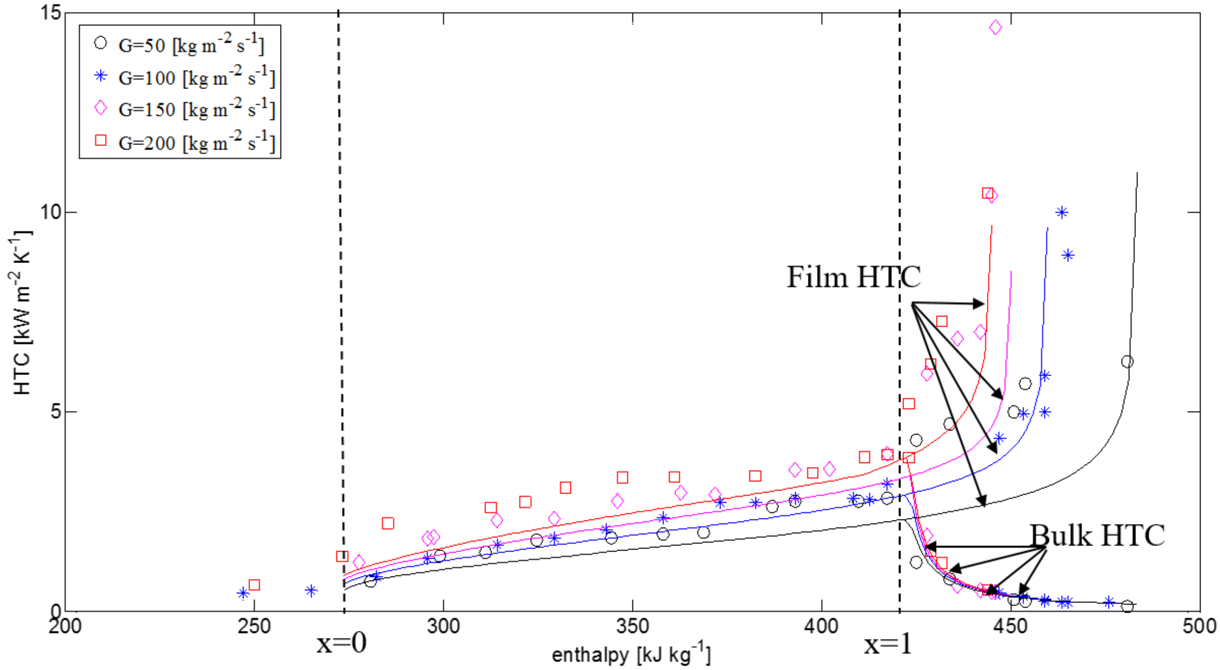

Figure 5: Comparison of conventional (bulk) HTC and film HTC under same heat flux and different mass flux

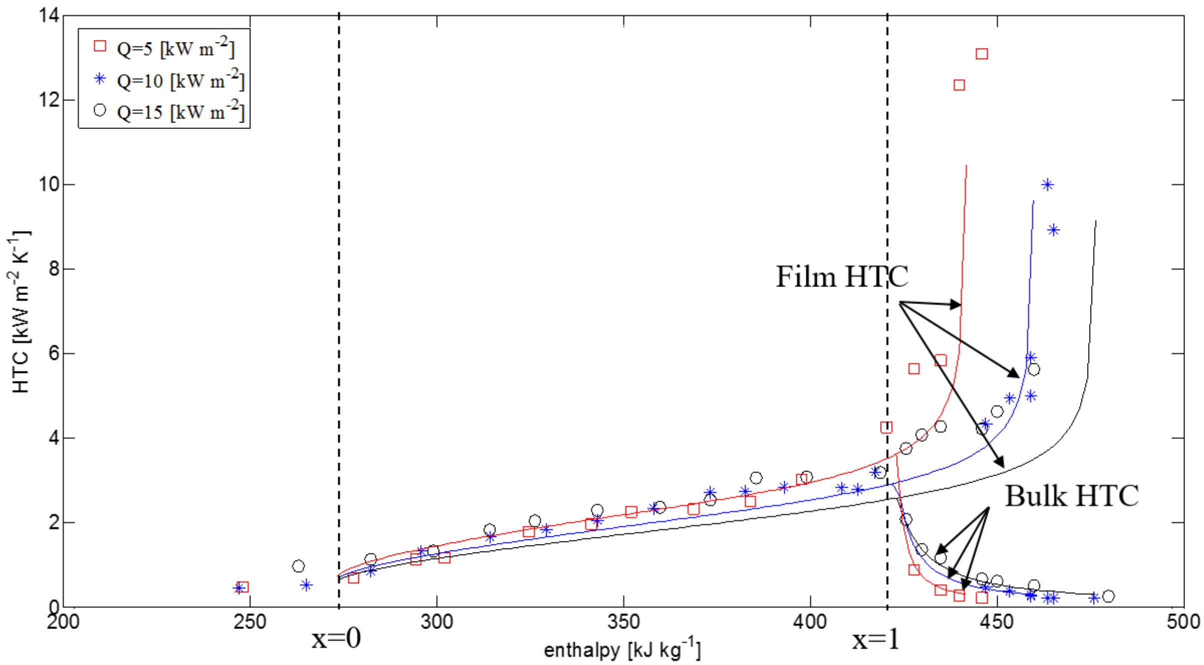

Figure 6: Comparison of conventional (bulk) HTC and film HTC under same mass flux and different heat flux

Figure 5 illustrates the comparison between conventional HTC and film HTC with heat flux of 10 kW m^{-2} and mass flus from 50 to 200 kg m^{-2} s^{-2}. Film HTC increases with mass flux and remains a smooth curve during the entire condensation process. Figure 6 illustrates the comparison between conventional HTC and film HTC with heat flux from 5 to 15 kW m^{-2} and mass flus of 100 kg m^{-2} s^{-2}. At the beginning where heat transfer is very sensitive to film thickness, larger heat flux yields smaller film HTC due to thicker film. As quality decreases, heat transfer becomes less sensitive to film thickness and more to flow regime. Since heat flux does not alter the flow regime for condensation, the difference of film HTC created by heat flux becomes smaller and smaller as the condensation proceeds.

Film HTC unifies the heat transfer performance in the 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 the CSH region and film HTC will sharply decrease due to the increase of film thickness. Meanwhile, stratification effects start to show at the bottom of the tube. The goal of the model would be to catch the mechanism of condensation and create a continuous curve from superheated vapor to subcooled liquid.

4. CONCLUSIONS

In this study, the conventional HTC presents completely different behavior in the TP and CSH region. In the 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 very much. In the 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 are no physical reasons 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 shows its potential to be used as a tool in developing a unified model for the entire condensation process.

NOMENCLATURE

SH	Superheated	
CSH	Condensing superheated	
TP	Two-phase	
HTC	Heat transfer coefficient	(W m^{-2} K^{-1})
T	Temperature	(ºC)
P	Pressure	(Pa)
G	Mass flux	(kg m^{-2})
Q	Heat flux	(kW m^{-2})
O.D	Outer diameter	mm
O.D	Inner diameter	mm
x	Quality	

Subscripts

b	Bulk
sat	Saturated
w	Wall
f	Film
superheat	Temperature difference between saturation and bulk flow
subcool	Temperature difference between saturation and wall

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